Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

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Chapter 1
INTRODUCTION AND BASIC CONCEPTS

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Thermodynamics

1-1C On a downhill road the potential energy of the bicyclist is being converted to kinetic energy, and thus the bicyclist picks up speed. There is no creation of energy, and thus no violation of the conservation of energy principle.

1-2C A car going uphill without the engine running would increase the energy of the car, and thus it would be a violation of the first law of thermodynamics. Therefore, this cannot happen. Using a level meter (a device with an air bubble between two marks of a horizontal water tube) it can shown that the road that looks uphill to the eye is actually downhill.

1-3C There is no truth to his claim. It violates the second law of thermodynamics.

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Mass, Force, and Units

1-4C The "pound" mentioned here must be "**lbf**" since thrust is a force, and the lbf is the force unit in the English system. You should get into the habit of *never* writing the unit "lb", but always use either "lbm" or "lbf" as appropriate since the two units have different dimensions.

1-5C In this unit, the word *light* refers to the speed of light. The light-year unit is then the product of a velocity and time. Hence, this product forms a distance dimension and unit.

1-6C There is no acceleration, thus the net force is zero in both cases.

1-7E The weight of a man on earth is given. His weight on the moon is to be determined.

Analysis Applying Newton's second law to the weight force gives

$$W = mg \longrightarrow m = \frac{W}{g} = \frac{210\ \text{lbf}}{32.10\ \text{ft/s}^2}\left(\frac{32.174\ \text{lbm}\cdot\text{ft/s}^2}{1\ \text{lbf}}\right) = 210.5\ \text{lbm}$$

Mass is invariant and the man will have the same mass on the moon. Then, his weight on the moon will be

$$W = mg = (210.5\ \text{lbm})(5.47\ \text{ft/s}^2)\left(\frac{1\ \text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/s}^2}\right) = \mathbf{35.8\ lbf}$$

1-8 The interior dimensions of a room are given. The mass and weight of the air in the room are to be determined.

Assumptions The density of air is constant throughout the room.

Properties The density of air is given to be $\rho = 1.16\ \text{kg/m}^3$.

Analysis The mass of the air in the room is

$$m = \rho V = (1.16\ \text{kg/m}^3)(6\times 6\times 8\ \text{m}^3) = \mathbf{334.1\ kg}$$

Thus,

$$W = mg = (334.1\ \text{kg})(9.81\ \text{m/s}^2)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{3277\ N}$$



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1-9 The variation of gravitational acceleration above the sea level is given as a function of altitude. The height at which the weight of a body will decrease by 0.5% is to be determined.



Analysis The weight of a body at the elevation z can be expressed as

$$W = mg = m(9.807 - 3.32 \times 10^{-6} z)$$

In our case,

$$W = 0.995 W_s = 0.995 mg_s = 0.995(m)(9.81)$$

Substituting,

$$0.995(9.81) = (9.81 - 3.32 \times 10^{-6} z) \longrightarrow z = 14{,}774 \text{ m} \cong \mathbf{14{,}770\ m}$$

1-10 The mass of an object is given. Its weight is to be determined.

Analysis Applying Newton's second law, the weight is determined to be

$$W = mg = (200 \text{ kg})(9.6 \text{ m/s}^2) = \mathbf{1920\ N}$$

1-11E The constant-pressure specific heat of air given in a specified unit is to be expressed in various units.

Analysis Applying Newton's second law, the weight is determined in various units to be

$$c_p = (1.005 \text{ kJ/kg} \cdot ^\circ\text{C})\left(\frac{1 \text{ kJ/kg} \cdot \text{K}}{1 \text{ kJ/kg} \cdot ^\circ\text{C}}\right) = \mathbf{1.005\ kJ/kg \cdot K}$$

$$c_p = (1.005 \text{ kJ/kg} \cdot ^\circ\text{C})\left(\frac{1000 \text{ J}}{1 \text{ kJ}}\right)\left(\frac{1 \text{ kg}}{1000 \text{ g}}\right) = \mathbf{1.005\ J/g \cdot ^\circ C}$$

$$c_p = (1.005 \text{ kJ/kg} \cdot ^\circ\text{C})\left(\frac{1 \text{ kcal}}{4.1868 \text{ kJ}}\right) = \mathbf{0.240\ kcal/kg \cdot ^\circ C}$$

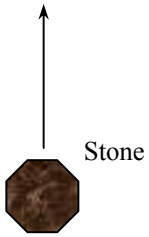
$$c_p = (1.005 \text{ kJ/kg} \cdot ^\circ\text{C})\left(\frac{1 \text{ Btu/lbm} \cdot ^\circ\text{F}}{4.1868 \text{ kJ/kg} \cdot ^\circ\text{C}}\right) = \mathbf{0.240\ Btu/lbm \cdot ^\circ F}$$

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1-12 A rock is thrown upward with a specified force. The acceleration of the rock is to be determined.

Analysis The weight of the rock is

$$W = mg = (3\ \text{kg})(9.79\ \text{m/s}^2)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 29.37\ \text{N}$$



Then the net force that acts on the rock is

$$F_{\text{net}} = F_{\text{up}} - F_{\text{down}} = 200 - 29.37 = 170.6\ \text{N}$$

From the Newton's second law, the acceleration of the rock becomes

$$a = \frac{F}{m} = \frac{170.6\ \text{N}}{3\ \text{kg}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = \mathbf{56.9\ m/s^2}$$

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1-13 Problem 1-12 is reconsidered. The entire EES solution is to be printed out, including the numerical results with proper units.

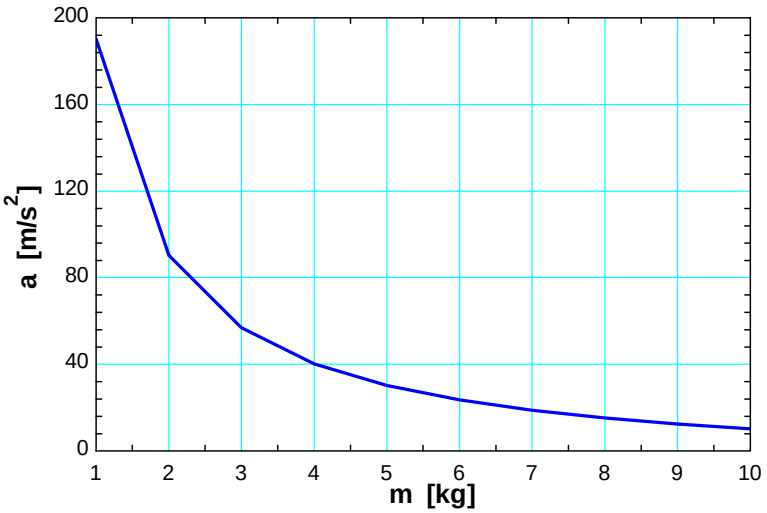
Analysis The problem is solved using EES, and the solution is given below.

```
"The weight of the rock is"
W=m*g
m=3 [kg]
g=9.79 [m/s2]
"The force balance on the rock yields the net force acting on the rock as"
F_up=200 [N]
F_net = F_up - F_down
F_down=W
"The acceleration of the rock is determined from Newton's second law."
F_net=m*a

"To Run the program, press F2 or select Solve from the Calculate menu."

SOLUTION
a=56.88 [m/s^2]
F_down=29.37 [N]
F_net=170.6 [N]
F_up=200 [N]
g=9.79 [m/s2]
m=3 [kg]
W=29.37 [N]
```

m [kg]	a [m/s^2]
1	190.2
2	90.21
3	56.88
4	40.21
5	30.21
6	23.54
7	18.78
8	15.21
9	12.43
10	10.21



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1-14 During an analysis, a relation with inconsistent units is obtained. A correction is to be found, and the probable cause of the error is to be determined.

Analysis The two terms on the right-hand side of the equation

$$E = 25 \text{ kJ} + 7 \text{ kJ/kg}$$

do not have the same units, and therefore they cannot be added to obtain the total energy. Multiplying the last term by mass will eliminate the kilograms in the denominator, and the whole equation will become dimensionally homogeneous; that is, every term in the equation will have the same unit.

Discussion Obviously this error was caused by forgetting to multiply the last term by mass at an earlier stage.

1-15 A resistance heater is used to heat water to desired temperature. The amount of electric energy used in kWh and kJ are to be determined.

Analysis The resistance heater consumes electric energy at a rate of 4 kW or 4 kJ/s. Then the total amount of electric energy used in 2 hours becomes

Total energy = (Energy per unit time)(Time interval)

= (4 kW)(2 h)

= **8 kWh**

Noting that 1 kWh = (1 kJ/s)(3600 s) = 3600 kJ,

Total energy = (8 kWh)(3600 kJ/kWh)

= **28,800 kJ**

Discussion Note kW is a unit for power whereas kWh is a unit for energy.

1-16 A gas tank is being filled with gasoline at a specified flow rate. Based on unit considerations alone, a relation is to be obtained for the filling time.

Assumptions Gasoline is an incompressible substance and the flow rate is constant.

Analysis The filling time depends on the volume of the tank and the discharge rate of gasoline. Also, we know that the unit of time is 'seconds'. Therefore, the independent quantities should be arranged such that we end up with the unit of seconds. Putting the given information into perspective, we have

$$t \text{ [s]} \quad \leftrightarrow \quad V \text{ [L]}, \text{ and } \dot{V} \text{ [L/s}\}$$

It is obvious that the only way to end up with the unit "s" for time is to divide the tank volume by the discharge rate. Therefore, the desired relation is

$$\boxed{t = \frac{V}{\dot{V}}}$$

Discussion Note that this approach may not work for cases that involve dimensionless (and thus unitless) quantities.

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1-17 A pool is to be filled with water using a hose. Based on unit considerations, a relation is to be obtained for the volume of the pool.

Assumptions Water is an incompressible substance and the average flow velocity is constant.

Analysis The pool volume depends on the filling time, the cross-sectional area which depends on hose diameter, and flow velocity. Also, we know that the unit of volume is m^3. Therefore, the independent quantities should be arranged such that we end up with the unit of seconds. Putting the given information into perspective, we have

V [m^3] is a function of t [s], D [m], and V [m/s}

It is obvious that the only way to end up with the unit "m^3" for volume is to multiply the quantities t and V with the square of D. Therefore, the desired relation is

$$V = CD^2Vt$$

where the constant of proportionality is obtained for a round hose, namely, $C = \pi/4$ so that $V = (\pi D^2/4)Vt$.

Discussion Note that the values of dimensionless constants of proportionality cannot be determined with this approach.

1-18 It is to be shown that the power needed to accelerate a car is proportional to the mass and the square of the velocity of the car, and inversely proportional to the time interval.

Assumptions The car is initially at rest.

Analysis The power needed for acceleration depends on the mass, velocity change, and time interval. Also, the unit of power $\dot{W}$ is watt, W, which is equivalent to

$$W = J/s = N\cdot m/s = (kg\cdot m/s^2)m/s = kg\cdot m^2/s^3$$

Therefore, the independent quantities should be arranged such that we end up with the unit $kg\cdot m^2/s^3$ for power. Putting the given information into perspective, we have

$\dot{W}$ [$kg\cdot m^2/s^3$] is a function of m [kg], V [m/s], and t [s]

It is obvious that the only way to end up with the unit "$kg\cdot m^2/s^3$" for power is to multiply mass with the square of the velocity and divide by time. Therefore, the desired relation is

$\dot{W}$ is proportional to mV^2/t

or,

$$\boxed{\dot{W} = CmV^2/t}$$

where C is the dimensionless constant of proportionality (whose value is ½ in this case).

Discussion Note that this approach cannot determine the numerical value of the dimensionless numbers involved.

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Systems, Properties, State, and Processes

1-19C This system is a region of space or open system in that mass such as air and food can cross its control boundary. The system can also interact with the surroundings by exchanging heat and work across its control boundary. By tracking these interactions, we can determine the energy conversion characteristics of this system.

1-20C The system is taken as the air contained in the piston-cylinder device. This system is a closed or fixed mass system since no mass enters or leaves it.

1-21C Any portion of the atmosphere which contains the ozone layer will work as an open system to study this problem. Once a portion of the atmosphere is selected, we must solve the practical problem of determining the interactions that occur at the control surfaces which surround the system's control volume.

1-22C Intensive properties do not depend on the size (extent) of the system but extensive properties do.

1-23C If we were to divide the system into smaller portions, the weight of each portion would also be smaller. Hence, the weight is an *extensive property*.

1-24C If we were to divide this system in half, both the volume and the number of moles contained in each half would be one-half that of the original system. The molar specific volume of the original system is

$$\bar{v} = \frac{V}{N}$$

and the molar specific volume of one of the smaller systems is

$$\bar{v} = \frac{V/2}{N/2} = \frac{V}{N}$$

which is the same as that of the original system. The molar specific volume is then an *intensive property*.

1-25C For a system to be in thermodynamic equilibrium, the temperature has to be the same throughout but the pressure does not. However, there should be no unbalanced pressure forces present. The increasing pressure with depth in a fluid, for example, should be balanced by increasing weight.

1-26C A process during which a system remains almost in equilibrium at all times is called a quasi-equilibrium process. Many engineering processes can be approximated as being quasi-equilibrium. The work output of a device is maximum and the work input to a device is minimum when quasi-equilibrium processes are used instead of nonquasi-equilibrium processes.

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1-27C A process during which the temperature remains constant is called isothermal; a process during which the pressure remains constant is called isobaric; and a process during which the volume remains constant is called isochoric.

1-28C The state of a simple compressible system is completely specified by two independent, intensive properties.

1-29C The pressure and temperature of the water are normally used to describe the state. Chemical composition, surface tension coefficient, and other properties may be required in some cases.

As the water cools, its pressure remains fixed. This cooling process is then an isobaric process.

1- 30C When analyzing the acceleration of gases as they flow through a nozzle, the proper choice for the system is the volume within the nozzle, bounded by the entire inner surface of the nozzle and the inlet and outlet cross-sections. This is a control volume since mass crosses the boundary.

1-31C A process is said to be steady-flow if it involves no changes with time anywhere within the system or at the system boundaries.

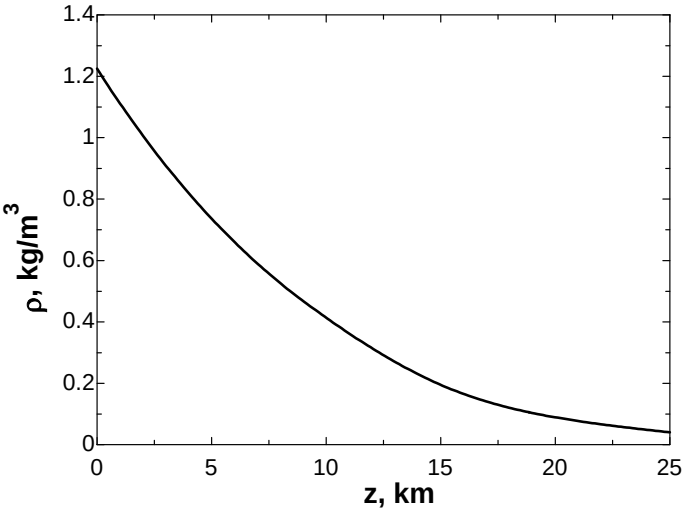
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1-32 The variation of density of atmospheric air with elevation is given in tabular form. A relation for the variation of density with elevation is to be obtained, the density at 7 km elevation is to be calculated, and the mass of the atmosphere using the correlation is to be estimated.

Assumptions **1** Atmospheric air behaves as an ideal gas. **2** The earth is perfectly sphere with a radius of 6377 km, and the thickness of the atmosphere is 25 km.

Properties The density data are given in tabular form as

r, km	z, km	ρ, kg/m^3
6377	0	1.225
6378	1	1.112
6379	2	1.007
6380	3	0.9093
6381	4	0.8194
6382	5	0.7364
6383	6	0.6601
6385	8	0.5258
6387	10	0.4135
6392	15	0.1948
6397	20	0.08891
6402	25	0.04008



Analysis Using EES, (1) Define a trivial function rho= a+z in equation window, (2) select new parametric table from Tables, and type the data in a two-column table, (3) select Plot and plot the data, and (4) select plot and click on "curve fit" to get curve fit window. Then specify 2nd order polynomial and enter/edit equation. The results are:

$\rho(z) = a + bz + cz^2 = 1.20252 - 0.101674z + 0.0022375z^2$ for the unit of kg/m^3,

(or, $\rho(z) = (1.20252 - 0.101674z + 0.0022375z^2)\times 10^9$ for the unit of kg/km^3)

where z is the vertical distance from the earth surface at sea level. At z = 7 km, the equation would give ρ = 0.60 kg/m^3.

(*b*) The mass of atmosphere can be evaluated by integration to be

$$m = \int_V \rho dV = \int_{z=0}^{h} (a + bz + cz^2)4\pi(r_0 + z)^2 dz = 4\pi \int_{z=0}^{h} (a + bz + cz^2)(r_0^2 + 2r_0 z + z^2)dz$$

$$= 4\pi\left[ar_0^2 h + r_0(2a + br_0)h^2/2 + (a + 2br_0 + cr_0^2)h^3/3 + (b + 2cr_0)h^4/4 + ch^5/5\right]$$

where r_0 = 6377 km is the radius of the earth, h = 25 km is the thickness of the atmosphere, and a = 1.20252, b = -0.101674, and c = 0.0022375 are the constants in the density function. Substituting and multiplying by the factor 10^9 for the density unity kg/km^3, the mass of the atmosphere is determined to be

m = **5.092×10^{18} kg**

Discussion Performing the analysis with excel would yield exactly the same results.

EES Solution for final result:

```
a=1.2025166;          b=-0.10167
c=0.0022375;          r=6377;          h=25
m=4*pi*(a*r^2*h+r*(2*a+b*r)*h^2/2+(a+2*b*r+c*r^2)*h^3/3+(b+2*c*r)*h^4/4+c*h^5/5)*1E+9
```

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Temperature

1-33C The zeroth law of thermodynamics states that two bodies are in thermal equilibrium if both have the same temperature reading, even if they are not in contact.

1-34C They are Celsius (°C) and kelvin (K) in the SI, and fahrenheit (°F) and rankine (R) in the English system.

1-35C Probably, but not necessarily. The operation of these two thermometers is based on the thermal expansion of a fluid. If the thermal expansion coefficients of both fluids vary linearly with temperature, then both fluids will expand at the same rate with temperature, and both thermometers will always give identical readings. Otherwise, the two readings may deviate.

1-36 A temperature is given in °C. It is to be expressed in K.

Analysis The Kelvin scale is related to Celsius scale by

$$T(\text{K}) = T(°\text{C}) + 273$$

Thus,

$$T(\text{K}) = 37°\text{C} + 273 = \mathbf{310\ K}$$

1-37E The temperature of air given in °C unit is to be converted to °F and R unit.

Analysis Using the conversion relations between the various temperature scales,

$$T(°\text{F}) = 1.8T(°\text{C}) + 32 = (1.8)(150) + 32 = \mathbf{302°F}$$
$$T(\text{R}) = T(°\text{F}) + 460 = 302 + 460 = \mathbf{762\ R}$$

1-38 A temperature change is given in °C. It is to be expressed in K.

Analysis This problem deals with temperature changes, which are identical in Kelvin and Celsius scales. Thus,

$$\Delta T(\text{K}) = \Delta T(°\text{C}) = \mathbf{45\ K}$$

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1-39E The flash point temperature of engine oil given in °F unit is to be converted to K and R units.

Analysis Using the conversion relations between the various temperature scales,

$$T(\mathrm{R}) = T(°\mathrm{F}) + 460 = 363 + 460 = \mathbf{823\ R}$$

$$T(\mathrm{K}) = \frac{T(\mathrm{R})}{1.8} = \frac{823}{1.8} = \mathbf{457\ K}$$

1-40E The temperature of ambient air given in °C unit is to be converted to °F, K and R units.

Analysis Using the conversion relations between the various temperature scales,

$$T = -40°\mathrm{C} = (-40)(1.8) + 32 = \mathbf{-40°C}$$

$$T = -40 + 273.15 = \mathbf{233.15\ K}$$

$$T = -40 + 459.67 = \mathbf{419.67\ R}$$

1-41E The change in water temperature given in °F unit is to be converted to °C, K and R units.

Analysis Using the conversion relations between the various temperature scales,

$$\Delta T = 10 / 1.8 = \mathbf{5.6°C}$$

$$\Delta T = 10 / 1.8 = \mathbf{5.6\ K}$$

$$\Delta T = 10°\mathrm{F} = \mathbf{10\ R}$$

1-42E A temperature range given in °F unit is to be converted to °C unit and the temperature difference in °F is to be expressed in K, °C, and R.

Analysis The lower and upper limits of comfort range in °C are

$$T(°\mathrm{C}) = \frac{T(°\mathrm{F}) - 32}{1.8} = \frac{65 - 32}{1.8} = \mathbf{18.3°C}$$

$$T(°\mathrm{C}) = \frac{T(°\mathrm{F}) - 32}{1.8} = \frac{75 - 32}{1.8} = \mathbf{23.9°C}$$

A temperature change of 10°F in various units are

$$\Delta T(\mathrm{R}) = \Delta T(°\mathrm{F}) = \mathbf{10\ R}$$

$$\Delta T(°\mathrm{C}) = \frac{\Delta T(°\mathrm{F})}{1.8} = \frac{10}{1.8} = \mathbf{5.6°C}$$

$$\Delta T(\mathrm{K}) = \Delta T(°\mathrm{C}) = \mathbf{5.6\ K}$$

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Pressure, Manometer, and Barometer

1-43C The pressure relative to the atmospheric pressure is called the *gage pressure*, and the pressure relative to an absolute vacuum is called *absolute pressure*.

1-44C The blood vessels are more restricted when the arm is parallel to the body than when the arm is perpendicular to the body. For a constant volume of blood to be discharged by the heart, the blood pressure must increase to overcome the increased resistance to flow.

1-45C No, the absolute pressure in a liquid of constant density does not double when the depth is doubled. It is the *gage pressure* that doubles when the depth is doubled.

1-46C If the lengths of the sides of the tiny cube suspended in water by a string are very small, the magnitudes of the pressures on all sides of the cube will be the same.

1-47C *Pascal's principle* states that *the pressure applied to a confined fluid increases the pressure throughout by the same amount*. This is a consequence of the pressure in a fluid remaining constant in the horizontal direction. An example of Pascal's principle is the operation of the hydraulic car jack.

1-48E The pressure given in psia unit is to be converted to kPa.

Analysis Using the psia to kPa units conversion factor,

$$P = (150\ \text{psia})\left(\frac{6.895\ \text{kPa}}{1\ \text{psia}}\right) = \mathbf{1034\ kPa}$$

1-49 The pressure in a tank is given. The tank's pressure in various units are to be determined.

Analysis Using appropriate conversion factors, we obtain

(*a*) $$P = (1500\ \text{kPa})\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right) = \mathbf{1500\ kN/m^2}$$

(*b*) $$P = (1500\ \text{kPa})\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right)\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right) = \mathbf{1{,}500{,}000\ kg/m\cdot s^2}$$

(*c*) $$P = (1500\ \text{kPa})\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right)\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)\left(\frac{1000\ \text{m}}{1\ \text{km}}\right) = \mathbf{1{,}500{,}000{,}000\ kg/km\cdot s^2}$$

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1-50E The pressure in a tank in SI unit is given. The tank's pressure in various English units are to be determined.

Analysis Using appropriate conversion factors, we obtain

(*a*) $$P = (1500\ \text{kPa})\left(\frac{20.886\ \text{lbf/ft}^2}{1\ \text{kPa}}\right) = \mathbf{31{,}330\ lbf/ft^2}$$

(*b*) $$P = (1500\ \text{kPa})\left(\frac{20.886\ \text{lbf/ft}^2}{1\ \text{kPa}}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right)\left(\frac{1\ \text{psia}}{1\ \text{lbf/in}^2}\right) = \mathbf{217.6\ psia}$$

1-51E The pressure given in mm Hg unit is to be converted to psia.

Analysis Using the mm Hg to kPa and kPa to psia units conversion factors,

$$P = (1500\ \text{mm Hg})\left(\frac{0.1333\ \text{kPa}}{1\ \text{mm Hg}}\right)\left(\frac{1\ \text{psia}}{6.895\ \text{kPa}}\right) = \mathbf{29.0\ psia}$$

1-52 The pressure given in mm Hg unit is to be converted to kPa.

Analysis Using the mm Hg to kPa units conversion factor,

$$P = (1250\ \text{mm Hg})\left(\frac{0.1333\ \text{kPa}}{1\ \text{mm Hg}}\right) = \mathbf{166.6\ kPa}$$

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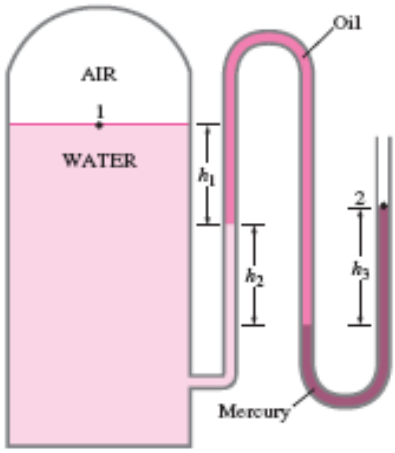
1-53 The pressure in a pressurized water tank is measured by a multi-fluid manometer. The gage pressure of air in the tank is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus we can determine the pressure at the air-water interface.

Properties The densities of mercury, water, and oil are given to be 13,600, 1000, and 850 kg/m^3, respectively.

Analysis Starting with the pressure at point 1 at the air-water interface, and moving along the tube by adding (as we go down) or subtracting (as we go up) th e ρgh terms until we reach point 2, and setting the result equal to P_{atm} since the tube is open to the atmosphere gives

$$P_1 + \rho_{water} g h_1 + \rho_{oil} g h_2 - \rho_{mercury} g h_3 = P_{atm}$$



Solving for P_1,

$$P_1 = P_{atm} - \rho_{water} g h_1 - \rho_{oil} g h_2 + \rho_{mercury} g h_3$$

or,

$$P_1 - P_{atm} = g(\rho_{mercury} h_3 - \rho_{water} h_1 - \rho_{oil} h_2)$$

Noting that $P_{1,gage} = P_1 - P_{atm}$ and substituting,

$$P_{1,gage} = (9.81\,\text{m/s}^2)[(13{,}600\,\text{kg/m}^3)(0.46\,\text{m}) - (1000\,\text{kg/m}^3)(0.2\,\text{m}) - (850\,\text{kg/m}^3)(0.3\,\text{m})]\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right)$$

$$= \mathbf{56.9\,kPa}$$

Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

1-54 The barometric reading at a location is given in height of mercury column. The atmospheric pressure is to be determined.

Properties The density of mercury is given to be 13,600 kg/m^3.

Analysis The atmospheric pressure is determined directly from

$$P_{atm} = \rho g h$$

$$= (13{,}600\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(0.750\,\text{m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right)$$

$$= \mathbf{100.1\ kPa}$$

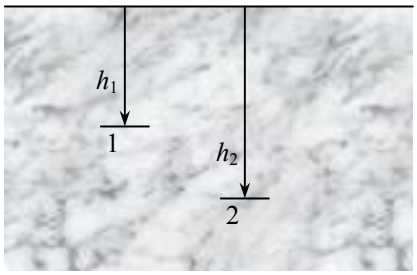
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1-55 The gage pressure in a liquid at a certain depth is given. The gage pressure in the same liquid at a different depth is to be determined.

Assumptions The variation of the density of the liquid with depth is negligible.

Analysis The gage pressure at two different depths of a liquid can be expressed as

$$P_1 = \rho g h_1 \qquad \text{and} \qquad P_2 = \rho g h_2$$



Taking their ratio,

$$\frac{P_2}{P_1} = \frac{\rho g h_2}{\rho g h_1} = \frac{h_2}{h_1}$$

Solving for P_2 and substituting gives

$$P_2 = \frac{h_2}{h_1} P_1 = \frac{9\ \text{m}}{3\ \text{m}}(42\ \text{kPa}) = \mathbf{126\ kPa}$$

Discussion Note that the gage pressure in a given fluid is proportional to depth.

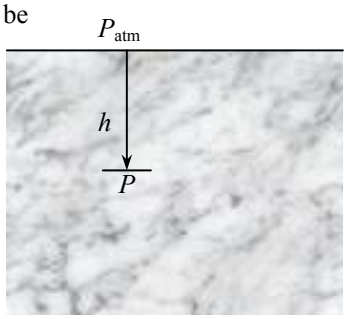
1-56 The absolute pressure in water at a specified depth is given. The local atmospheric pressure and the absolute pressure at the same depth in a different liquid are to be determined.

Assumptions The liquid and water are incompressible.

Properties The specific gravity of the fluid is given to be SG = 0.85. We take the density of water to be 1000 kg/m^3. Then density of the liquid is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{H_2O} = (0.85)(1000\ \text{kg/m}^3) = 850\ \text{kg/m}^3$$

Analysis (*a*) Knowing the absolute pressure, the atmospheric pressure can be determined from



$$\begin{aligned} P_{\text{atm}} &= P - \rho g h \\ &= (145\ \text{kPa}) - (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(5\ \text{m})\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{96.0\ kPa} \end{aligned}$$

(*b*) The absolute pressure at a depth of 5 m in the other liquid is

$$\begin{aligned} P &= P_{\text{atm}} + \rho g h \\ &= (96.0\ \text{kPa}) + (850\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(5\ \text{m})\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{137.7\ kPa} \end{aligned}$$

Discussion Note that at a given depth, the pressure in the lighter fluid is lower, as expected.

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1-57E It is to be shown that 1 kgf/cm^2 = 14.223 psi .

Analysis Noting that 1 kgf = 9.80665 N, 1 N = 0.22481 lbf, and 1 in = 2.54 cm, we have

$$1\text{ kgf} = 9.80665\text{ N} = (9.80665\text{ N})\left(\frac{0.22481\text{ lbf}}{1\text{ N}}\right) = 2.20463\text{ lbf}$$

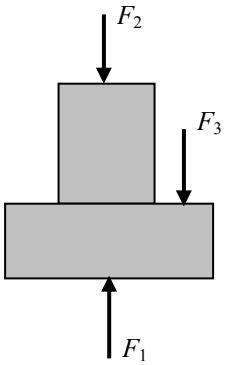
and

$$1\text{ kgf/cm}^2 = 2.20463\text{ lbf/cm}^2 = (2.20463\text{ lbf/cm}^2)\left(\frac{2.54\text{ cm}}{1\text{ in}}\right)^2 = 14.223\text{ lbf/in}^2 = \mathbf{14.223\text{ psi}}$$

1-58E The pressure in chamber 3 of the two-piston cylinder shown in the figure is to be determined.

Analysis The area upon which pressure 1 acts is

$$A_1 = \pi\frac{D_1^2}{4} = \pi\frac{(3\text{ in})^2}{4} = 7.069\text{ in}^2$$



and the area upon which pressure 2 acts is

$$A_2 = \pi\frac{D_2^2}{4} = \pi\frac{(1.5\text{ in})^2}{4} = 1.767\text{ in}^2$$

The area upon which pressure 3 acts is given by

$$A_3 = A_1 - A_2 = 7.069 - 1.767 = 5.302\text{ in}^2$$

The force produced by pressure 1 on the piston is then

$$F_1 = P_1 A_1 = (150\text{ psia})\left(\frac{1\text{ lbf/in}^2}{1\text{ psia}}\right)(7.069\text{ in}^2) = 1060\text{ lbf}$$

while that produced by pressure 2 is

$$F_1 = P_2 A_2 = (250\text{ psia})(1.767\text{ in}^2) = 441.8\text{ lbf}$$

According to the vertical force balance on the piston free body diagram

$$F_3 = F_1 - F_2 = 1060 - 441.8 = 618.3\text{ lbf}$$

Pressure 3 is then

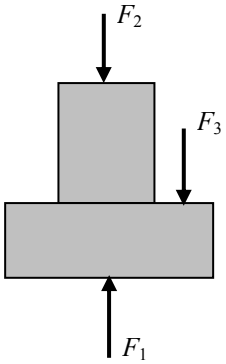
$$P_3 = \frac{F_3}{A_3} = \frac{618.3\text{ lbf}}{5.302\text{ in}^2} = \mathbf{117\text{ psia}}$$

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1-59 The pressure in chamber 1 of the two-piston cylinder shown in the figure is to be determined.

Analysis Summing the forces acting on the piston in the vertical direction gives

$$F_2 + F_3 = F_1$$
$$P_2 A_2 + P_3 (A_1 - A_2) = P_1 A_1$$



which when solved for P_1 gives

$$P_1 = P_2 \frac{A_2}{A_1} + P_3 \left(1 - \frac{A_2}{A_1}\right)$$

since the areas of the piston faces are given by $A = \pi D^2 / 4$ the above equation becomes

$$P_1 = P_2 \left(\frac{D_2}{D_1}\right)^2 + P_3 \left[1 - \left(\frac{D_2}{D_1}\right)^2\right]$$
$$= (2000\ \text{kPa})\left(\frac{4}{10}\right)^2 + (700\ \text{kPa})\left[1 - \left(\frac{4}{10}\right)^2\right]$$
$$= \mathbf{908\ kPa}$$

1-60 The mass of a woman is given. The minimum imprint area per shoe needed to enable her to walk on the snow without sinking is to be determined.

Assumptions **1** The weight of the person is distributed uniformly on the imprint area of the shoes. **2** One foot carries the entire weight of a person during walking, and the shoe is sized for walking conditions (rather than standing). **3** The weight of the shoes is negligible.

Analysis The mass of the woman is given to be 70 kg. For a pressure of 0.5 kPa on the snow, the imprint area of one shoe must be



$$A = \frac{W}{P} = \frac{mg}{P}$$
$$= \frac{(70\ \text{kg})(9.81\ \text{m/s}^2)}{0.5\ \text{kPa}} \left(\frac{1\ \text{N}}{1\ \text{kg} \cdot \text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) = \mathbf{1.37\ m^2}$$

Discussion This is a very large area for a shoe, and such shoes would be impractical to use. Therefore, some sinking of the snow should be allowed to have shoes of reasonable size.

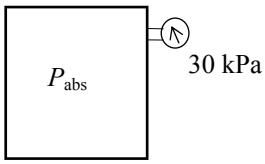
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1-61 The vacuum pressure reading of a tank is given. The absolute pressure in the tank is to be determined.

Properties The density of mercury is given to be $\rho = 13{,}590$ kg/m^3.

Analysis The atmospheric (or barometric) pressure can be expressed as

$$\begin{aligned} P_{\text{atm}} &= \rho g h \\ &= (13{,}590 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.750 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= 100.0 \text{ kPa} \end{aligned}$$



$P_{\text{atm}} = 750$ mmHg

Then the absolute pressure in the tank becomes

$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}} = 100.0 - 30 = \mathbf{70.0 \text{ kPa}}$$

1-62E The vacuum pressure given in kPa unit is to be converted to various units.

Analysis Using the definition of vacuum pressure,

$$P_{\text{gage}} = \text{not applicable for pressures below atmospheric pressure}$$
$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}} = 98 - 80 = \mathbf{18 \text{ kPa}}$$

Then using the conversion factors,

$$P_{\text{abs}} = (18 \text{ kPa})\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right) = \mathbf{18 \text{ kN/m}^2}$$

$$P_{\text{abs}} = (18 \text{ kPa})\left(\frac{1 \text{ lbf/in}^2}{6.895 \text{ kPa}}\right) = \mathbf{2.61 \text{ lbf/in}^2}$$

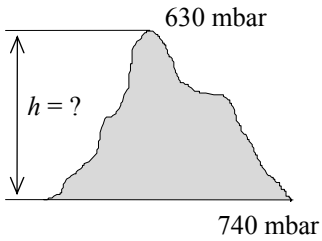
$$P_{\text{abs}} = (18 \text{ kPa})\left(\frac{1 \text{ psi}}{6.895 \text{ kPa}}\right) = \mathbf{2.61 \text{ psi}}$$

$$P_{\text{abs}} = (18 \text{ kPa})\left(\frac{1 \text{ mm Hg}}{0.1333 \text{ kPa}}\right) = \mathbf{135 \text{ mm Hg}}$$

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1-63 A mountain hiker records the barometric reading before and after a hiking trip. The vertical distance climbed is to be determined.

Assumptions The variation of air density and the gravitational acceleration with altitude is negligible.



Properties The density of air is given to be ρ = 1.20 kg/m^3.

Analysis Taking an air column between the top and the bottom of the mountain and writing a force balance per unit base area, we obtain

$$W_{\text{air}} / A = P_{\text{bottom}} - P_{\text{top}}$$
$$(\rho g h)_{\text{air}} = P_{\text{bottom}} - P_{\text{top}}$$

$$(1.20 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(h)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ bar}}{100{,}000 \text{ N/m}^2}\right) = (0.740 - 0.630) \text{ bar}$$

It yields

h = **934 m**

which is also the distance climbed.

1-64 A barometer is used to measure the height of a building by recording reading at the bottom and at the top of the building. The height of the building is to be determined.

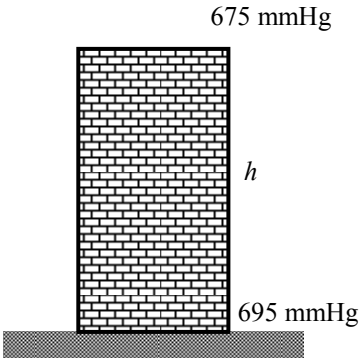
Assumptions The variation of air density with altitude is negligible.

Properties The density of air is given to be ρ = 1.18 kg/m^3. The density of mercury is 13,600 kg/m^3.

Analysis Atmospheric pressures at the top and at the bottom of the building are

$$\begin{aligned} P_{\text{top}} &= (\rho g h)_{\text{top}} \\ &= (13{,}600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.675 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= 90.06 \text{ kPa} \end{aligned}$$

$$\begin{aligned} P_{\text{bottom}} &= (\rho g h)_{\text{bottom}} \\ &= (13{,}600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.695 \text{ m})\left(\frac{1\text{N}}{1\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= 92.72 \text{ kPa} \end{aligned}$$



Taking an air column between the top and the bottom of the building and writing a force balance per unit base area, we obtain

$$W_{\text{air}} / A = P_{\text{bottom}} - P_{\text{top}}$$
$$(\rho g h)_{\text{air}} = P_{\text{bottom}} - P_{\text{top}}$$

$$(1.18 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(h)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) = (92.72 - 90.06) \text{ kPa}$$

It yields

h = **231 m**

which is also the height of the building.

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1-65 Problem 1-64 is reconsidered. The entire EES solution is to be printed out, including the numerical results with proper units.

Analysis The problem is solved using EES, and the solution is given below.

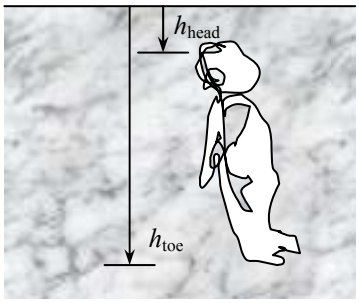
```
P_bottom=695 [mmHg]
P_top=675 [mmHg]
g=9.81 [m/s^2]  "local acceleration of gravity at sea level"
rho=1.18  [kg/m^3]
DELTAP_abs=(P_bottom-P_top)*CONVERT(mmHg, kPa) "[kPa]"     "Delta P reading from the barometers, converted from mmHg to kPa."
DELTAP_h =rho*g*h*Convert(Pa, kPa) "Delta P due to the air fluid column height, h, between the top and bottom of the building."
DELTAP_abs=DELTAP_h

SOLUTION
DELTAP_abs=2.666 [kPa]
DELTAP_h=2.666 [kPa]
g=9.81 [m/s^2]
h=230.3 [m]
P_bottom=695 [mmHg]
P_top=675 [mmHg]
rho=1.18 [kg/m^3]
```

1-66 A man is standing in water vertically while being completely submerged. The difference between the pressures acting on the head and on the toes is to be determined.

Assumptions Water is an incompressible substance, and thus the density does not change with depth.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.



Analysis The pressures at the head and toes of the person can be expressed as

$$P_{head} = P_{atm} + \rho g h_{head} \quad \text{and} \quad P_{toe} = P_{atm} + \rho g h_{toe}$$

where h is the vertical distance of the location in water from the free surface. The pressure difference between the toes and the head is determined by subtracting the first relation above from the second,

$$P_{toe} - P_{head} = \rho g h_{toe} - \rho g h_{head} = \rho g (h_{toe} - h_{head})$$

Substituting,

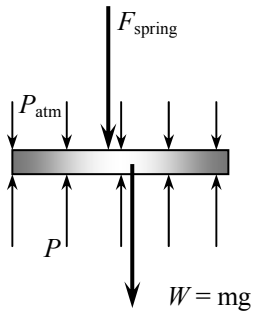
$$P_{toe} - P_{head} = (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(1.75 \text{ m} - 0)\left(\frac{1\text{N}}{1\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{kPa}}{1000\text{N/m}^2}\right) = \mathbf{17.2\ kPa}$$

Discussion This problem can also be solved by noting that the atmospheric pressure (1 atm = 101.325 kPa) is equivalent to 10.3-m of water height, and finding the pressure that corresponds to a water height of 1.75 m.

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1-67 A gas contained in a vertical piston-cylinder device is pressurized by a spring and by the weight of the piston. The pressure of the gas is to be determined.

Analysis Drawing the free body diagram of the piston and balancing the vertical forces yield



$$PA = P_{atm}A + W + F_{spring}$$

Thus,

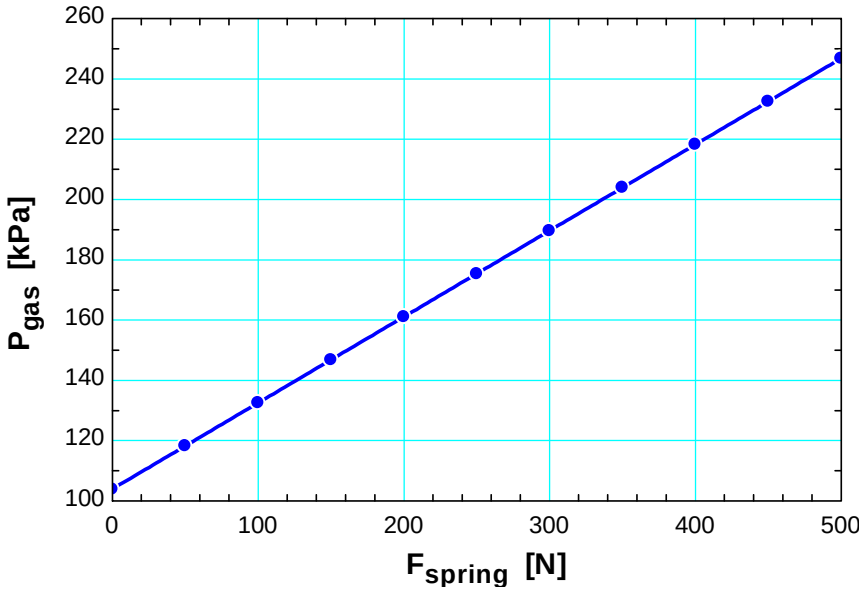
$$P = P_{atm} + \frac{mg + F_{spring}}{A}$$
$$= (95\ \text{kPa}) + \frac{(3.2\ \text{kg})(9.81\ \text{m/s}^2) + 150\ \text{N}}{35 \times 10^{-4}\ \text{m}^2}\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right)$$
$$= \mathbf{147\ kPa}$$

1-68 Problem 1-67 is reconsidered. The effect of the spring force in the range of 0 to 500 N on the pressure inside the cylinder is to be investigated. The pressure against the spring force is to be plotted, and results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
g=9.81 [m/s^2]
P_atm= 95 [kPa]
m_piston=3.2 [kg]
{F_spring=150 [N]}
A=35*CONVERT(cm^2, m^2)
W_piston=m_piston*g
F_atm=P_atm*A*CONVERT(kPa, N/m^2)
"From the free body diagram of the piston, the balancing vertical forces yield:"
F_gas= F_atm+F_spring+W_piston
P_gas=F_gas/A*CONVERT(N/m^2, kPa)
```

F_{spring} [N]	P_{gas} [kPa]
0	104
50	118.3
100	132.5
150	146.8
200	161.1
250	175.4
300	189.7
350	204
400	218.3
450	232.5
500	246.8

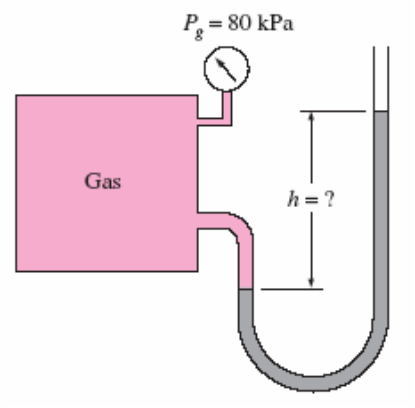


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1-69 Both a gage and a manometer are attached to a gas to measure its pressure. For a specified reading of gage pressure, the difference between the fluid levels of the two arms of the manometer is to be determined for mercury and water.

Properties The densities of water and mercury are given to be $\rho_{water} = 1000\ kg/m^3$ and be $\rho_{Hg} = 13{,}600\ kg/m^3$.

Analysis The gage pressure is related to the vertical distance h between the two fluid levels by



$$P_{\text{gage}} = \rho g h \longrightarrow h = \frac{P_{\text{gage}}}{\rho g}$$

(*a*) For mercury,

$$h = \frac{P_{\text{gage}}}{\rho_{Hg} g}$$

$$= \frac{80\ \text{kPa}}{(13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right)\left(\frac{1000\ \text{kg/m}\cdot\text{s}^2}{1\ \text{kN}}\right) = \mathbf{0.60\ m}$$

(*b*) For water,

$$h = \frac{P_{\text{gage}}}{\rho_{\text{H}_2\text{O}} g} = \frac{80\ \text{kPa}}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right)\left(\frac{1000\ \text{kg/m}\cdot\text{s}^2}{1\ \text{kN}}\right) = \mathbf{8.16\ m}$$

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1-70 Problem 1-69 is reconsidered. The effect of the manometer fluid density in the range of 800 to 13,000 kg/m^3 on the differential fluid height of the manometer is to be investigated. Differential fluid height against the density is to be plotted, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
"Let's modify this problem to also calculate the absolute pressure in the tank by supplying the atmospheric pressure.
Use the relationship between the pressure gage reading and the manometer fluid column height. "

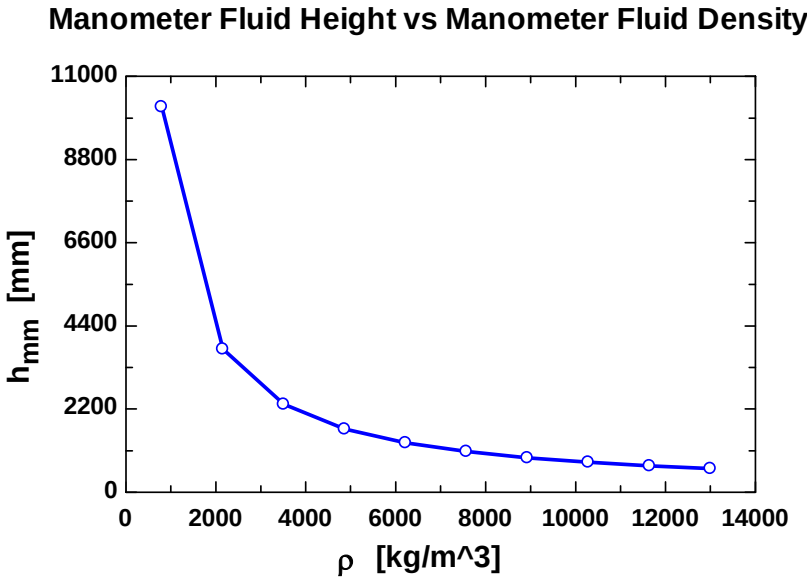
Function fluid_density(Fluid$)
"This function is needed since if-then-else logic can only be used in functions or procedures.
The underscore displays whatever follows as subscripts in the Formatted Equations Window."
    If fluid$='Mercury' then  fluid_density=13600 else fluid_density=1000
end

{Input from the diagram window.  If the diagram window is hidden, then all of the input must come from the equations window.  Also note that brackets can also denote comments - but these comments do not appear in the formatted equations window.}
{Fluid$='Mercury'
P_atm = 101.325 [kPa]
DELTAP=80  [kPa] "Note how DELTAP is displayed on the Formatted Equations Window."}

g=9.807  [m/s^2] "local acceleration of gravity at sea level"
rho=Fluid_density(Fluid$)   "Get the fluid density, either Hg or H2O, from the function"
"To plot fluid height against density place {} around the above equation.  Then set up the parametric table and solve."
DELTAP = RHO*g*h/1000
"Instead of dividiing by 1000 Pa/kPa we could have multiplied by the EES function, CONVERT(Pa,kPa)"
h_mm=h*convert(m, mm)   "The fluid height in mm is found using the built-in CONVERT function."
P_abs= P_atm + DELTAP

"To make the graph, hide the diagram window and remove the {}brackets from Fluid$ and from P_atm.  Select New Parametric Table from the Tables menu.  Choose P_abs, DELTAP and h to be in the table.  Choose Alter Values from the Tables menu.  Set values of h to range from 0 to 1 in steps of 0.2.  Choose Solve Table (or press F3) from the Calculate menu.  Choose New Plot Window from the Plot menu.  Choose to plot P_abs vs h and then choose Overlay Plot from the Plot menu and plot DELTAP on the same scale."
```

ρ [kg/m^3]	h_{mm} [mm]
800	10197
2156	3784
3511	2323
4867	1676
6222	1311
7578	1076
8933	913.1
10289	792.8
11644	700.5
13000	627.5



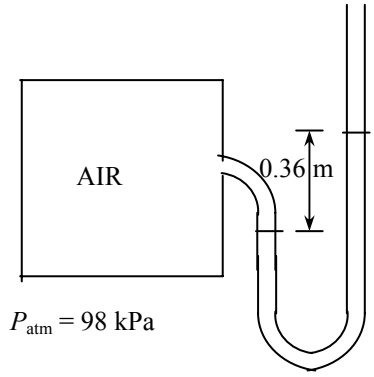
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1-71 The air pressure in a tank is measured by an oil manometer. For a given oil-level difference between the two columns, the absolute pressure in the tank is to be determined.

Properties The density of oil is given to be $\rho = 850$ kg/m^3.

Analysis The absolute pressure in the tank is determined from

$$\begin{aligned} P &= P_{\text{atm}} + \rho g h \\ &= (98\ \text{kPa}) + (850\ \text{kg/m}^3)(9.81\text{m/s}^2)(0.36\ \text{m})\left(\frac{1\text{kPa}}{1000\,\text{N/m}^2}\right) \\ &= \mathbf{101.0\ kPa} \end{aligned}$$



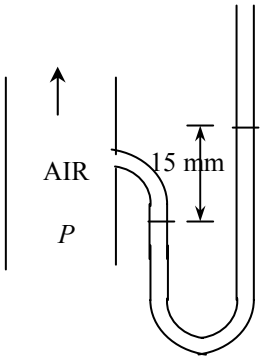
1-72 The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.

Properties The density of mercury is given to be ρ = 13,600 kg/m^3.

Analysis (*a*) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(*b*) The absolute pressure in the duct is determined from

$$\begin{aligned} P &= P_{\text{atm}} + \rho g h \\ &= (100\ \text{kPa}) + (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.015\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{102\ \ kPa} \end{aligned}$$



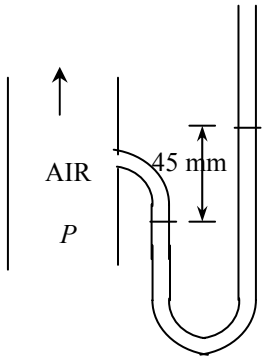
1-73 The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.

Properties The density of mercury is given to be ρ = 13,600 kg/m^3.

Analysis (*a*) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(*b*) The absolute pressure in the duct is determined from

$$\begin{aligned} P &= P_{\text{atm}} + \rho g h \\ &= (100\ \text{kPa}) + (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.045\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{106\ \ kPa} \end{aligned}$$



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1-74E The systolic and diastolic pressures of a healthy person are given in mmHg. These pressures are to be expressed in kPa, psi, and meter water column.

Assumptions Both mercury and water are incompressible substances.

Properties We take the densities of water and mercury to be 1000 kg/m^3 and 13,600 kg/m^3, respectively.

Analysis Using the relation $P = \rho gh$ for gage pressure, the high and low pressures are expressed as

$$P_{\text{high}} = \rho g h_{\text{high}} = (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.12\ \text{m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{16.0\ kPa}$$

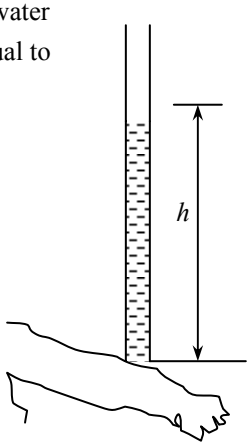
$$P_{\text{low}} = \rho g h_{\text{low}} = (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.08\ \text{m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{10.7\ kPa}$$

Noting that 1 psi = 6.895 kPa,

$$P_{\text{high}} = (16.0\ \text{Pa})\left(\frac{1\ \text{psi}}{6.895\,\text{kPa}}\right) = \mathbf{2.32\ psi} \qquad \text{and} \qquad P_{\text{low}} = (10.7\ \text{Pa})\left(\frac{1\ \text{psi}}{6.895\,\text{kPa}}\right) = \mathbf{1.55\ psi}$$

For a given pressure, the relation $P = \rho gh$ can be expressed for mercury and water as $P = \rho_{\text{water}} g h_{\text{water}}$ and $P = \rho_{\text{mercury}} g h_{\text{mercury}}$. Setting these two relations equal to each other and solving for water height gives

$$P = \rho_{\text{water}} g h_{\text{water}} = \rho_{\text{mercury}} g h_{\text{mercury}} \ \rightarrow\ h_{\text{water}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury}}$$



Therefore,

$$h_{\text{water, high}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, high}} = \frac{13{,}600\ \text{kg/m}^3}{1000\ \text{kg/m}^3}(0.12\ \text{m}) = \mathbf{1.63\ m}$$

$$h_{\text{water, low}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, low}} = \frac{13{,}600\ \text{kg/m}^3}{1000\ \text{kg/m}^3}(0.08\ \text{m}) = \mathbf{1.09\ m}$$

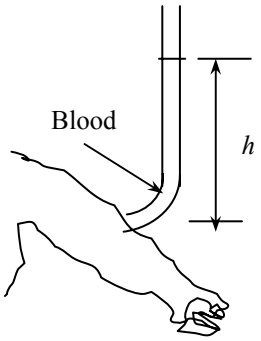
Discussion Note that measuring blood pressure with a "water" monometer would involve differential fluid heights higher than the person, and thus it is impractical. This problem shows why mercury is a suitable fluid for blood pressure measurement devices.

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1-75 A vertical tube open to the atmosphere is connected to the vein in the arm of a person. The height that the blood will rise in the tube is to be determined.

Assumptions **1** The density of blood is constant. **2** The gage pressure of blood is 120 mmHg.

Properties The density of blood is given to be ρ = 1050 kg/m^3.



Analysis For a given gage pressure, the relation $P = \rho gh$ can be expressed for mercury and blood as $P = \rho_{\text{blood}} g h_{\text{blood}}$ and $P = \rho_{\text{mercury}} g h_{\text{mercury}}$. Setting these two relations equal to each other we get

$$P = \rho_{\text{blood}} g h_{\text{blood}} = \rho_{\text{mercury}} g h_{\text{mercury}}$$

Solving for blood height and substituting gives

$$h_{\text{blood}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{blood}}} h_{\text{mercury}} = \frac{13{,}600 \text{ kg/m}^3}{1050 \text{ kg/m}^3}(0.12 \text{ m}) = \mathbf{1.55 \text{ m}}$$

Discussion Note that the blood can rise about one and a half meters in a tube connected to the vein. This explains why IV tubes must be placed high to force a fluid into the vein of a patient.

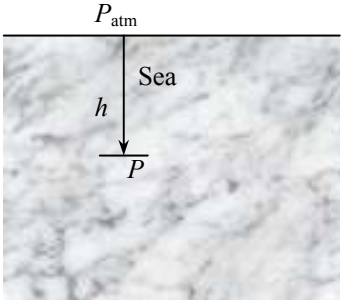
1-76 A diver is moving at a specified depth from the water surface. The pressure exerted on the surface of the diver by water is to be determined.

Assumptions The variation of the density of water with depth is negligible.

Properties The specific gravity of seawater is given to be SG = 1.03. We take the density of water to be 1000 kg/m^3.

Analysis The density of the seawater is obtained by multiplying its specific gravity by the density of water which is taken to be 1000 kg/m^3:

$$\rho = \text{SG} \times \rho_{H_2O} = (1.03)(1000 \text{ kg/m}^3) = 1030 \text{ kg/m}^3$$



The pressure exerted on a diver at 30 m below the free surface of the sea is the absolute pressure at that location:

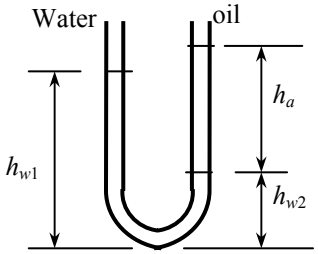
$$\begin{aligned} P &= P_{\text{atm}} + \rho gh \\ &= (101 \text{ kPa}) + (1030 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(30 \text{ m})\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= \mathbf{404 \text{ kPa}} \end{aligned}$$

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1-77 Water is poured into the U-tube from one arm and oil from the other arm. The water column height in one arm and the ratio of the heights of the two fluids in the other arm are given. The height of each fluid in that arm is to be determined.

Assumptions Both water and oil are incompressible substances.

Properties The density of oil is given to be ρ = 790 kg/m^3. We take the density of water to be ρ =1000 kg/m^3.



Analysis The height of water column in the left arm of the monometer is given to be h_{w1} = 0.70 m. We let the height of water and oil in the right arm to be h_{w2} and h_a, respectively. Then, $h_a = 4h_{w2}$. Noting that both arms are open to the atmosphere, the pressure at the bottom of the U-tube can be expressed as

$$P_{bottom} = P_{atm} + \rho_w g h_{w1} \quad \text{and} \quad P_{bottom} = P_{atm} + \rho_w g h_{w2} + \rho_a g h_a$$

Setting them equal to each other and simplifying,

$$\rho_w g h_{w1} = \rho_w g h_{w2} + \rho_a g h_a \quad \rightarrow \quad \rho_w h_{w1} = \rho_w h_{w2} + \rho_a h_a \quad \rightarrow \quad h_{w1} = h_{w2} + (\rho_a / \rho_w) h_a$$

Noting that $h_a = 4h_{w2}$, the water and oil column heights in the second arm are determined to be

$$0.7 \text{ m} = h_{w2} + (790/1000)\, 4h_{w2} \quad \rightarrow \quad h_{w2} = \mathbf{0.168\ m}$$

$$0.7 \text{ m} = 0.168 \text{ m} + (790/1000) h_a \quad \rightarrow \quad h_a = \mathbf{0.673\ m}$$

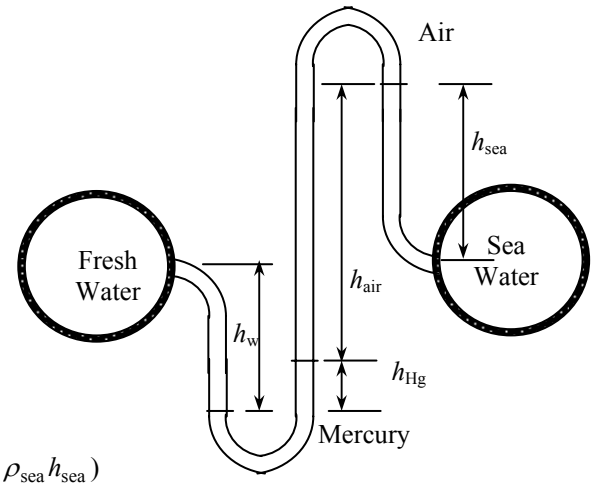
Discussion Note that the fluid height in the arm that contains oil is higher. This is expected since oil is lighter than water.

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1-78 Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

Properties The densities of seawater and mercury are given to be $\rho_{sea} = 1035$ kg/m^3 and $\rho_{Hg} = 13{,}600$ kg/m^3. We take the density of water to be $\rho_w = 1000$ kg/m^3.



Analysis Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the sea water pipe (point 2), and setting the result equal to P_2 gives

$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{air} g h_{air} + \rho_{sea} g h_{sea} = P_2$$

Rearranging and neglecting the effect of air column on pressure,

$$P_1 - P_2 = -\rho_w g h_w + \rho_{Hg} g h_{Hg} - \rho_{sea} g h_{sea} = g(\rho_{Hg} h_{Hg} - \rho_w h_w - \rho_{sea} h_{sea})$$

Substituting,

$$\begin{aligned} P_1 - P_2 &= (9.81\ \text{m/s}^2)[(13600\ \text{kg/m}^3)(0.1\ \text{m}) \\ &\quad - (1000\ \text{kg/m}^3)(0.6\ \text{m}) - (1035\ \text{kg/m}^3)(0.4\ \text{m})]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) \\ &= 3.39\ \text{kN/m}^2 = \mathbf{3.39\ kPa} \end{aligned}$$

Therefore, the pressure in the fresh water pipe is 3.39 kPa higher than the pressure in the sea water pipe.

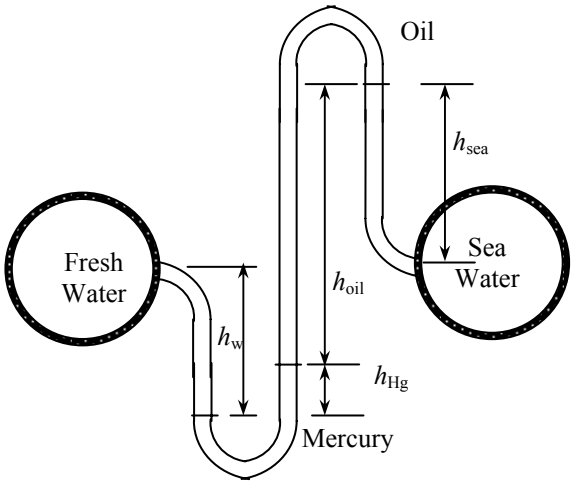
Discussion A 0.70-m high air column with a density of 1.2 kg/m^3 corresponds to a pressure difference of 0.008 kPa. Therefore, its effect on the pressure difference between the two pipes is negligible.

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1-79 Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.

Assumptions All the liquids are incompressible.

Properties The densities of seawater and mercury are given to be $\rho_{sea} = 1035$ kg/m^3 and $\rho_{Hg} = 13{,}600$ kg/m^3. We take the density of water to be $\rho_w = 1000$ kg/m^3. The specific gravity of oil is given to be 0.72, and thus its density is 720 kg/m^3.



Analysis Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the sea water pipe (point 2), and setting the result equal to P_2 gives

$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} + \rho_{sea} g h_{sea} = P_2$$

Rearranging,

$$\begin{aligned} P_1 - P_2 &= -\rho_w g h_w + \rho_{Hg} g h_{Hg} + \rho_{oil} g h_{oil} - \rho_{sea} g h_{sea} \\ &= g(\rho_{Hg} h_{Hg} + \rho_{oil} h_{oil} - \rho_w h_w - \rho_{sea} h_{sea}) \end{aligned}$$

Substituting,

$$\begin{aligned} P_1 - P_2 &= (9.81\ \text{m/s}^2)[(13600\ \text{kg/m}^3)(0.1\ \text{m}) + (720\ \text{kg/m}^3)(0.7\ \text{m}) - (1000\ \text{kg/m}^3)(0.6\ \text{m}) \\ &\quad - (1035\ \text{kg/m}^3)(0.4\ \text{m})]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) \\ &= 8.34\ \text{kN/m}^2 = \mathbf{8.34\ kPa} \end{aligned}$$

Therefore, the pressure in the fresh water pipe is 8.34 kPa higher than the pressure in the sea water pipe.

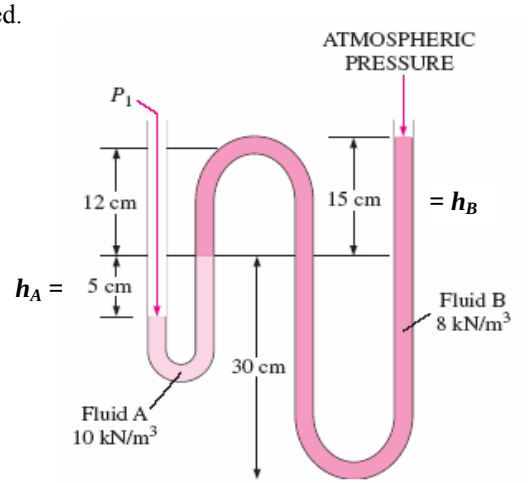
1-80 The pressure indicated by a manometer is to be determined.

Properties The specific weights of fluid A and fluid B are given to be 10 kN/m^3 and 8 kN/m^3, respectively.

Analysis The absolute pressure P_1 is determined from

$$\begin{aligned} P_1 &= P_{atm} + (\rho gh)_A + (\rho gh)_B \\ &= P_{atm} + \gamma_A h_A + \gamma_B h_B \\ &= (758\ \text{mm Hg})\left(\frac{0.1333\ \text{kPa}}{1\ \text{mm Hg}}\right) \\ &\quad + (10\ \text{kN/m}^3)(0.05\ \text{m}) + (8\ \text{kN/m}^3)(0.15\ \text{m}) \\ &= \mathbf{102.7\ kPa} \end{aligned}$$

Note that 1 kPa = 1 kN/m^2.



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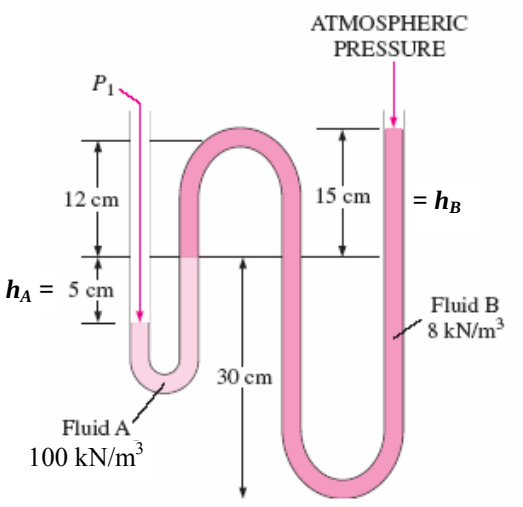
1-81 The pressure indicated by a manometer is to be determined.

Properties The specific weights of fluid A and fluid B are given to be 100 kN/m^3 and 8 kN/m^3, respectively.

Analysis The absolute pressure P_1 is determined from

$$\begin{aligned} P_1 &= P_{\text{atm}} + (\rho gh)_A + (\rho gh)_B \\ &= P_{\text{atm}} + \gamma_A h_A + \gamma_B h_B \\ &= 90\ \text{kPa} + (100\ \text{kN/m}^3)(0.05\ \text{m}) + (8\ \text{kN/m}^3)(0.15\ \text{m}) \\ &= \mathbf{96.2\ kPa} \end{aligned}$$

Note that 1 kPa = 1 kN/m^2.



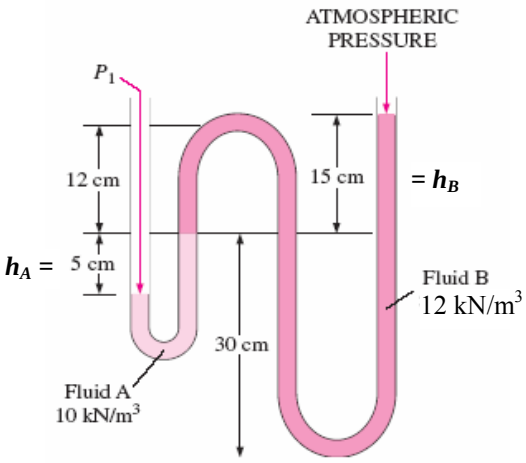
1-82 The pressure indicated by a manometer is to be determined.

Properties The specific weights of fluid A and fluid B are given to be 10 kN/m^3 and 12 kN/m^3, respectively.

Analysis The absolute pressure P_1 is determined from

$$\begin{aligned} P_1 &= P_{\text{atm}} + (\rho gh)_A + (\rho gh)_B \\ &= P_{\text{atm}} + \gamma_A h_A + \gamma_B h_B \\ &= (720\ \text{mm Hg})\left(\frac{0.1333\ \text{kPa}}{1\ \text{mm Hg}}\right) \\ &\quad + (10\ \text{kN/m}^3)(0.05\ \text{m}) + (12\ \text{kN/m}^3)(0.15\ \text{m}) \\ &= \mathbf{98.3\ kPa} \end{aligned}$$

Note that 1 kPa = 1 kN/m^2.



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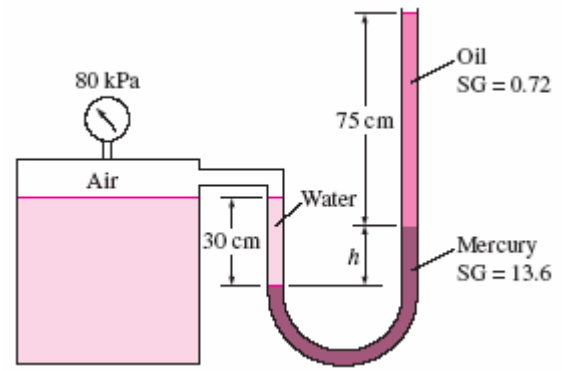
1-83 The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

Properties We take the density of water to be ρ_w=1000 kg/m^3. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

Analysis Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go u p) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

$$P_1 + \rho_{\text{w}} g h_w - \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{oil}} g h_{\text{oil}} = P_{atm}$$



Rearranging,

$$P_1 - P_{\text{atm}} = \rho_{\text{oil}} g h_{\text{oil}} + \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{w}} g h_w$$

or,

$$\frac{P_{1,\text{gage}}}{\rho_{\text{w}} g} = \text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_{\text{Hg}} h_{\text{Hg}} - h_w$$

Substituting,

$$\left(\frac{80\,\text{kPa}}{(1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)}\right)\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\,\text{kPa}.\cdot\text{m}^2}\right) = 0.72\times(0.75\,\text{m}) + 13.6\times h_{\text{Hg}} - 0.3\,\text{m}$$

Solving for h_{Hg} gives h_{Hg} = **0.582 m**. Therefore, the differential height of the mercury column must be 58.2 cm.

Discussion Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.

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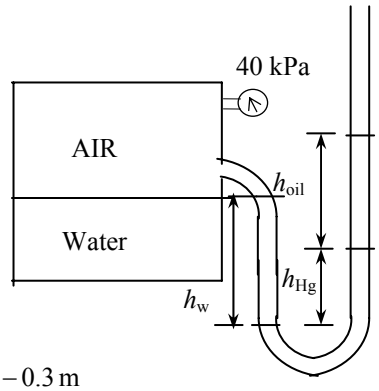
1-84 The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

Properties We take the density of water to be $\rho_w = 1000$ kg/m^3. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

Analysis Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} = P_{atm}$$



Rearranging,

$$P_1 - P_{atm} = \rho_{oil} g h_{oil} + \rho_{Hg} g h_{Hg} - \rho_w g h_w$$

or,
$$\frac{P_{1,gage}}{\rho_w g} = SG_{oil} h_{oil} + SG_{Hg} h_{Hg} - h_w$$

Substituting,

$$\left[\frac{40\text{ kPa}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)}\right]\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kPa}\cdot\text{m}^2}\right) = 0.72\times(0.75\text{ m}) + 13.6\times h_{Hg} - 0.3\text{ m}$$

Solving for h_{Hg} gives $h_{Hg} =$ **0.282 m**. Therefore, the differential height of the mercury column must be 28.2 cm.

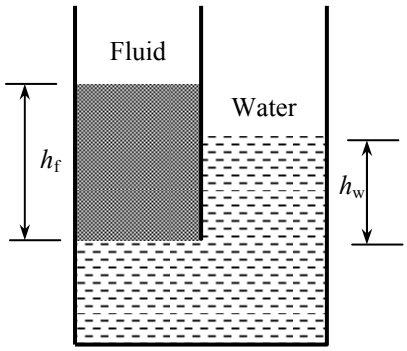
Discussion Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.

1-85 The top part of a water tank is divided into two compartments, and a fluid with an unknown density is poured into one side. The levels of the water and the liquid are measured. The density of the fluid is to be determined.

Assumptions **1** Both water and the added liquid are incompressible substances. **2** The added liquid does not mix with water.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis Both fluids are open to the atmosphere. Noting that the pressure of both water and the added fluid is the same at the contact surface, the pressure at this surface can be expressed as



$$P_{contact} = P_{atm} + \rho_f g h_f = P_{atm} + \rho_w g h_w$$

Simplifying and solving for ρ_f gives

$$\rho_f g h_f = \rho_w g h_w \quad \rightarrow \quad \rho_f = \frac{h_w}{h_f}\rho_w = \frac{55\text{ cm}}{65\text{ cm}}(1000\text{ kg/m}^3) = \mathbf{846\ kg/m^3}$$

Discussion Note that the added fluid is lighter than water as expected (a heavier fluid would sink in water).

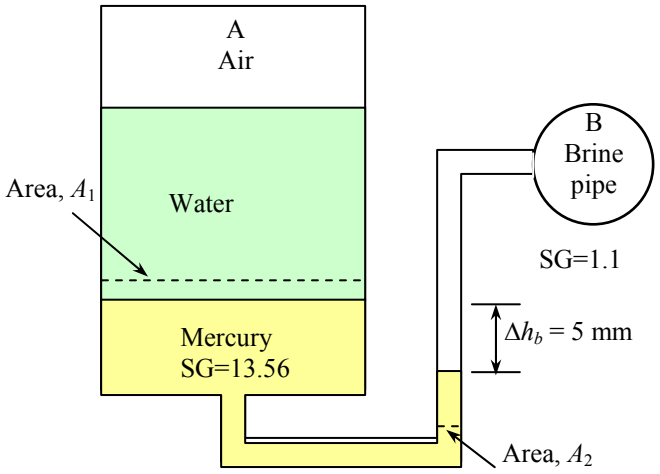
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1-86 The fluid levels in a multi-fluid U-tube manometer change as a result of a pressure drop in the trapped air space. For a given pressure drop and brine level change, the area ratio is to be determined.

Assumptions **1** All the liquids are incompressible. **2** Pressure in the brine pipe remains constant. **3** The variation of pressure in the trapped air space is negligible.

Properties The specific gravities are given to be 13.56 for mercury and 1.1 for brine. We take the standard density of water to be ρ_w=1000 kg/m^3.

Analysis It is clear from the problem statement and the figure that the brine pressure is much higher than the air pressure, and when the air pressure drops by 0.7 kPa, the pressure difference between the brine and the air space increases also by the same amount.



Starting with the air pressure (point A) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the brine pipe (point B), and setting the result equal to P_B before and after the pressure change of air give

Before: $$P_{A1} + \rho_{\text{w}} g h_w + \rho_{\text{Hg}} g h_{\text{Hg},1} - \rho_{\text{br}} g h_{\text{br},1} = P_B$$

After: $$P_{A2} + \rho_{\text{w}} g h_w + \rho_{\text{Hg}} g h_{\text{Hg},2} - \rho_{\text{br}} g h_{\text{br},2} = P_B$$

Subtracting,

$$P_{A2} - P_{A1} + \rho_{\text{Hg}} g \Delta h_{\text{Hg}} - \rho_{\text{br}} g \Delta h_{\text{br}} = 0 \quad \rightarrow \quad \frac{P_{A1} - P_{A2}}{\rho_w g} = SG_{\text{Hg}} \Delta h_{\text{Hg}} - SG_{\text{br}} \Delta h_{\text{br}} = 0 \qquad (1)$$

where Δh_{Hg} and Δh_{br} are the changes in the differential mercury and brine column heights, respectively, due to the drop in air pressure. Both of these are positive quantities since as the mercury-brine interface drops, the differential fluid heights for both mercury and brine increase. Noting also that the volume of mercury is constant, we have $A_1 \Delta h_{\text{Hg,left}} = A_2 \Delta h_{\text{Hg,right}}$ and

$$P_{A2} - P_{A1} = -0.7 \text{ kPa} = -700 \text{ N/m}^2 = -700 \text{ kg/m} \cdot \text{s}^2$$

$$\Delta h_{\text{br}} = 0.005 \text{ m}$$

$$\Delta h_{\text{Hg}} = \Delta h_{\text{Hg,right}} + \Delta h_{\text{Hg,left}} = \Delta h_{\text{br}} + \Delta h_{\text{br}} A_2 / \text{A}_1 = \Delta h_{\text{br}} (1 + A_2 / \text{A}_1)$$

Substituting,

$$\frac{700 \text{ kg/m} \cdot \text{s}^2}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)} = [13.56 \times 0.005(1 + A_2/A_1) - (1.1 \times 0.005)] \text{ m}$$

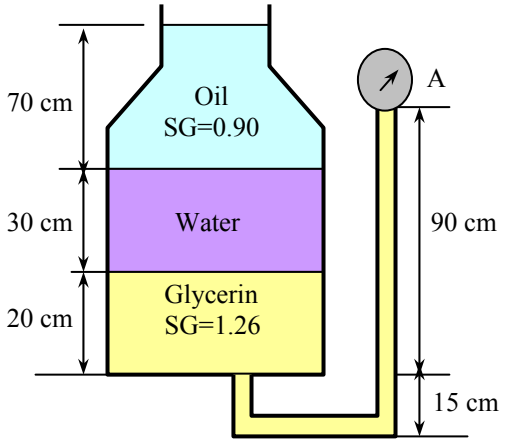
It gives

A_2/A_1 = **0.134**

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1-87 A multi-fluid container is connected to a U-tube. For the given specific gravities and fluid column heights, the gage pressure at A and the height of a mercury column that would create the same pressure at A are to be determined.

Assumptions **1** All the liquids are incompressible. **2** The multi-fluid container is open to the atmosphere.



Properties The specific gravities are given to be 1.26 for glycerin and 0.90 for oil. We take the standard density of water to be ρ_w =1000 kg/m^3, and the specific gravity of mercury to be 13.6.

Analysis Starting with the atmospheric pressure on the top surface of the container and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach point *A*, and setting the result equal to P_A give

$$P_{atm} + \rho_{oil} g h_{oil} + \rho_w g h_w - \rho_{gly} g h_{gly} = P_A$$

Rearranging and using the definition of specific gravity,

$$P_A - P_{atm} = SG_{oil}\rho_w g h_{oil} + SG_w \rho_w g h_w - SG_{gly}\rho_w g h_{gly}$$

or

$$P_{A,gage} = g\rho_w (SG_{oil} h_{oil} + SG_w h_w - SG_{gly} h_{gly})$$

Substituting,

$$P_{A,gage} = (9.81\,\text{m/s}^2)(1000\,\text{kg/m}^3)[0.90(0.70\,\text{m}) + 1(0.3\,\text{m}) - 1.26(0.70\,\text{m})]\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)$$

$$= 0.471\,\text{kN/m}^2 = \mathbf{0.471\,kPa}$$

The equivalent mercury column height is

$$h_{Hg} = \frac{P_{A,gage}}{\rho_{Hg} g} = \frac{0.471\,\text{kN/m}^2}{(13{,}600\,\text{kg/m}^3)(1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)}\left(\frac{1000\,\text{kg}\cdot\text{m/s}^2}{1\,\text{kN}}\right) = 0.00353\,\text{m} = \mathbf{0.353\,cm}$$

Discussion Note that the high density of mercury makes it a very suitable fluid for measuring high pressures in manometers.

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Solving Engineering Problems and EES

1-88C Despite the convenience and capability the engineering software packages offer, they are still just tools, and they will not replace the traditional engineering courses. They will simply cause a shift in emphasis in the course material from mathematics to physics. They are of great value in engineering practice, however, as engineers today rely on software packages for solving large and complex problems in a short time, and perform optimization studies efficiently.



1-89 Determine a positive real root of the following equation using EES:

$2x^3 - 10x^{0.5} - 3x = -3$

Solution by EES Software (Copy the following line and paste on a blank EES screen to verify solution):

```
2*x^3-10*x^0.5-3*x = -3
```

Answer: x = 2.063 (using an initial guess of x=2)



1-90 Solve the following system of 2 equations with 2 unknowns using EES:

$x^3 - y^2 = 7.75$

$3xy + y = 3.5$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
x^3-y^2=7.75
3*x*y+y=3.5
```

Answer x=2 y=0.5



1-91 Solve the following system of 3 equations with 3 unknowns using EES:

$2x - y + z = 7$

$3x^2 + 2y = z + 3$

$xy + 2z = 4$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
2*x-y+z=7
3*x^2+2*y=z+3
x*y+2*z=4
```

Answer x=1.609, y=-0.9872, z=2.794

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1-92 Solve the following system of 3 equations with 3 unknowns using EES:

$x^2y - z = 1$

$x - 3y^{0.5} + xz = -2$

$x + y - z = 2$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
x^2*y-z=1
x-3*y^0.5+x*z=-2
x+y-z=2
```

Answer x=1, y=1, z=0

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1-93E Specific heat of water is to be expressed at various units using unit conversion capability of EES.

Analysis The problem is solved using EES, and the solution is given below.

EQUATION WINDOW

```
"GIVEN"
C_p=4.18 [kJ/kg-C]

"ANALYSIS"
C_p_1=C_p*Convert(kJ/kg-C, kJ/kg-K)
C_p_2=C_p*Convert(kJ/kg-C, Btu/lbm-F)
C_p_3=C_p*Convert(kJ/kg-C, Btu/lbm-R)
C_p_4=C_p*Convert(kJ/kg-C, kCal/kg-C)
```

FORMATTED EQUATIONS WINDOW

GIVEN

$C_p = 4.18 \quad [kJ/kg\text{-}C]$

ANALYSIS

$$C_{p,1} = C_p \cdot \left| 1 \cdot \frac{kJ/kg-K}{kJ/kg-C} \right|$$

$$C_{p,2} = C_p \cdot \left| 0.238846 \cdot \frac{Btu/lbm-F}{kJ/kg-C} \right|$$

$$C_{p,3} = C_p \cdot \left| 0.238846 \cdot \frac{Btu/lbm-R}{kJ/kg-C} \right|$$

$$C_{p,4} = C_p \cdot \left| 0.238846 \cdot \frac{kCal/kg-C}{kJ/kg-C} \right|$$

SOLUTION

```
C_p=4.18 [kJ/kg-C]
C_p_1=4.18 [kJ/kg-K]
C_p_2=0.9984 [Btu/lbm-F]
C_p_3=0.9984 [Btu/lbm-R]
C_p_4=0.9984 [kCal/kg-C]
```

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Review Problems

1-94 The weight of a lunar exploration module on the moon is to be determined.

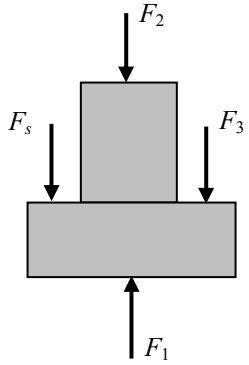
Analysis Applying Newton's second law, the weight of the module on the moon can be determined from

$$W_{\text{moon}} = mg_{\text{moon}} = \frac{W_{\text{earth}}}{g_{\text{earth}}} g_{\text{moon}} = \frac{2800\ \text{N}}{9.8\ \text{m/s}^2}(1.64\ \text{m/s}^2) = \mathbf{469\ N}$$

1-95 The deflection of the spring of the two-piston cylinder with a spring shown in the figure is to be determined.

Analysis Summing the forces acting on the piston in the vertical direction gives

$$F_s + F_2 + F_3 = F_1$$
$$kx + P_2 A_2 + P_3(A_1 - A_2) = P_1 A_1$$



which when solved for the deflection of the spring and substituting $A = \pi D^2/4$ gives

$$x = \frac{\pi}{4k}\left[P_1 D_1^2 - P_2 D_2^2 - P_3(D_1^2 - D_2^2)\right]$$
$$= \frac{\pi}{4\times 800}\left[5000\times 0.08^2 - 10{,}000\times 0.03^2 - 1000(0.08^2 - 0.03^2)\right]$$
$$= 0.0172\ \text{m}$$
$$= \mathbf{1.72\ cm}$$

We expressed the spring constant k in kN/m, the pressures in kPa (i.e., kN/m^2) and the diameters in m units.

1-96 An airplane is flying over a city. The local atmospheric pressure in that city is to be determined.

Assumptions The gravitational acceleration does not change with altitude.

Properties The densities of air and mercury are given to be 1.15 kg/m^3 and 13,600 kg/m^3.

Analysis The local atmospheric pressure is determined from

$$P_{\text{atm}} = P_{\text{plane}} + \rho g h$$
$$= 25\ \text{kPa} + (1.15\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(9000\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 126.5\ \text{kN/m}^2 \cong \mathbf{127\ kPa}$$

The atmospheric pressure may be expressed in mmHg as

$$h_{\text{Hg}} = \frac{P_{\text{atm}}}{\rho g} = \frac{126.5\ \text{kPa}}{(13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1000\ \text{Pa}}{1\ \text{kPa}}\right)\left(\frac{1000\ \text{mm}}{1\ \text{m}}\right) = \mathbf{948\ mmHg}$$

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1-97 The gravitational acceleration changes with altitude. Accounting for this variation, the weights of a body at different locations are to be determined.

Analysis The weight of an 80-kg man at various locations is obtained by substituting the altitude z (values in m) into the relation

$$W = mg = (80\text{kg})(9.807 - 3.32\times10^{-6}\,z\,\text{m/s}^2)\left(\frac{1\text{N}}{1\text{kg}\cdot\text{m/s}^2}\right)$$

Sea level: (z = 0 m): W = 80×(9.807-3.32x10^{-6}×0) = 80×9.807 = **784.6 N**

Denver: (z = 1610 m): W = 80×(9.807-3.32x10^{-6}×1610) = 80×9.802 = **784.2 N**

Mt. Ev.: (z = 8848 m): W = 80×(9.807-3.32x10^{-6}×8848) = 80×9.778 = **782.2 N**

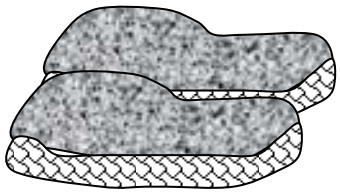
1-98 A man is considering buying a 12-oz steak for $3.15, or a 300-g steak for $2.95. The steak that is a better buy is to be determined.

Assumptions The steaks are of identical quality.

Analysis To make a comparison possible, we need to express the cost of each steak on a common basis. Let us choose 1 kg as the basis for comparison. Using proper conversion factors, the unit cost of each steak is determined to be

12 ounce steak:

$$\text{Unit Cost} = \left(\frac{\$3.15}{12\text{ oz}}\right)\left(\frac{16\text{ oz}}{1\text{ lbm}}\right)\left(\frac{1\text{ lbm}}{0.45359\text{ kg}}\right) = \mathbf{\$9.26/kg}$$



300 gram steak:

$$\text{Unit Cost} = \left(\frac{\$2.95}{300\text{ g}}\right)\left(\frac{1000\text{ g}}{1\text{ kg}}\right) = \mathbf{\$9.83/kg}$$

Therefore, the steak at the traditional market is a better buy.

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1.99E The mass of a substance is given. Its weight is to be determined in various units.

Analysis Applying Newton's second law, the weight is determined in various units to be

$$W = mg = (1\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{9.81\,N}$$

$$W = mg = (1\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{0.00981\,kN}$$

$$W = mg = (1\text{ kg})(9.81\text{ m/s}^2) = \mathbf{1\,kg\cdot m/s^2}$$

$$W = mg = (1\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kgf}}{9.81\text{ N}}\right) = \mathbf{1\,kgf}$$

$$W = mg = (1\text{ kg})\left(\frac{2.205\text{ lbm}}{1\text{ kg}}\right)(32.2\text{ ft/s}^2) = \mathbf{71\,lbm\cdot ft/s^2}$$

$$W = mg = (1\text{ kg})\left(\frac{2.205\text{ lbm}}{1\text{ kg}}\right)(32.2\text{ ft/s}^2)\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{2.21\,lbf}$$

1-100E The efficiency of a refrigerator increases by 3% per °C rise in the minimum temperature. This increase is to be expressed per °F, K, and R rise in the minimum temperature.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the increase in efficiency is

(a) **3%** for each K rise in temperature, and

(b), (c) 3/1.8 = **1.67%** for each R or °F rise in temperature.

1-101E The boiling temperature of water decreases by 3°C for each 1000 m rise in altitude. This decrease in temperature is to be expressed in °F, K, and R.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the decrease in the boiling temperature is

(a) **3 K** for each 1000 m rise in altitude, and

(b), (c) 3×1.8 = **5.4°F = 5.4 R** for each 1000 m rise in altitude.

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1-102E Hyperthermia of 5°C is considered fatal. This fatal level temperature change of body temperature is to be expressed in °F, K, and R.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the fatal level of hypothermia is

(a) **5 K**

(b) 5×1.8 = **9°F**

(c) 5×1.8 = **9 R**

1-103E A house is losing heat at a rate of 2700 kJ/h per °C temperature difference between the indoor and the outdoor temperatures. The rate of heat loss is to be expressed per °F, K, and R of temperature difference between the indoor and the outdoor temperatures.

Analysis The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the rate of heat loss from the house is

(a) **2700 kJ/h** per K difference in temperature, and

(b), (c) 2700/1.8 = **1500 kJ/h** per R or °F rise in temperature.

1-104 The average temperature of the atmosphere is expressed as $T_{atm} = 288.15 - 6.5z$ where z is altitude in km. The temperature outside an airplane cruising at 12,000 m is to be determined.

Analysis Using the relation given, the average temperature of the atmosphere at an altitude of 12,000 m is determined to be

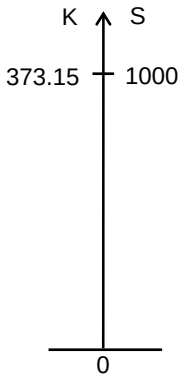
$$T_{atm} = 288.15 - 6.5z$$
$$= 288.15 - 6.5\times 12$$
$$= \mathbf{210.15\ K} = \mathbf{-\ 63°C}$$

Discussion This is the "average" temperature. The actual temperature at different times can be different.

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1-105 A new "Smith" absolute temperature scale is proposed, and a value of 1000 S is assigned to the boiling point of water. The ice point on this scale, and its relation to the Kelvin scale are to be determined.

Analysis All linear absolute temperature scales read zero at absolute zero pressure, and are constant multiples of each other. For example, T(R) = 1.8 T(K). That is, multiplying a temperature value in K by 1.8 will give the same temperature in R.



The proposed temperature scale is an acceptable absolute temperature scale since it differs from the other absolute temperature scales by a constant only. The boiling temperature of water in the Kelvin and the Smith scales are 315.15 K and 1000 K, respectively. Therefore, these two temperature scales are related to each other by

$$T(S) = \frac{1000}{373.15} T(\mathrm{K}) = \mathbf{2.6799T(K)}$$

The ice point of water on the Smith scale is

$$T(\mathrm{S})_{\mathrm{ice}} = 2.6799\ T(\mathrm{K})_{\mathrm{ice}} = 2.6799 \times 273.15 = \mathbf{732.0\ S}$$

1-106E An expression for the equivalent wind chill temperature is given in English units. It is to be converted to SI units.

Analysis The required conversion relations are 1 mph = 1.609 km/h and $T(°\mathrm{F}) = 1.8T(°\mathrm{C}) + 32$. The first thought that comes to mind is to replace $T(°\mathrm{F})$ in the equation by its equivalent $1.8T(°\mathrm{C}) + 32$, and V in mph by 1.609 km/h, which is the "regular" way of converting units. However, the equation we have is not a regular dimensionally homogeneous equation, and thus the regular rules do not apply. The V in the equation is a constant whose value is equal to the numerical value of the velocity in mph. Therefore, if V is given in km/h, we should divide it by 1.609 to convert it to the desired unit of mph. That is,

$$T_{\mathrm{equiv}}(°\mathrm{F}) = 91.4 - [91.4 - T_{\mathrm{ambient}}(°\mathrm{F})][0.475 - 0.0203(V / 1.609) + 0.304\sqrt{V / 1.609}]$$

or

$$T_{\mathrm{equiv}}(°\mathrm{F}) = 91.4 - [91.4 - T_{\mathrm{ambient}}(°\mathrm{F})][0.475 - 0.0126V + 0.240\sqrt{V}]$$

where V is in km/h. Now the problem reduces to converting a temperature in °F to a temperature in °C, using the proper convection relation:

$$1.8T_{\mathrm{equiv}}(°\mathrm{C}) + 32 = 91.4 - [91.4 - (1.8T_{\mathrm{ambient}}(°\mathrm{C}) + 32)][0.475 - 0.0126V + 0.240\sqrt{V}]$$

which simplifies to

$$T_{\mathrm{equiv}}(°\mathrm{C}) = 33.0 - (33.0 - T_{\mathrm{ambient}})(0.475 - 0.0126V + 0.240\sqrt{V})$$

where the ambient air temperature is in °C.

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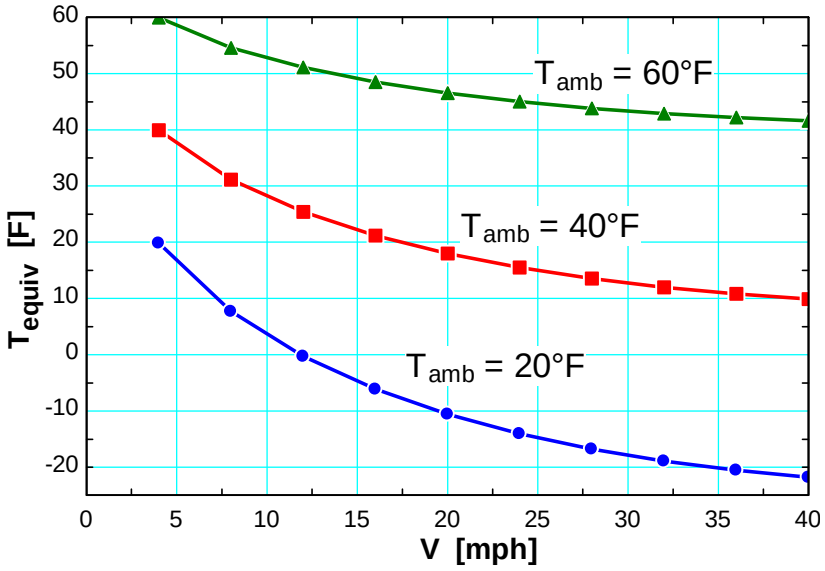
1-107E Problem 1-106E is reconsidered. The equivalent wind-chill temperatures in °F as a function of wind velocity in the range of 4 mph to 40 mph for the ambient temperatures of 20, 40, and 60°F are to be plotted, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
T_ambient=20
"V=20"
T_equiv=91.4-(91.4-T_ambient)*(0.475 - 0.0203*V + 0.304*sqrt(V))
```

V [mph]	T_{equiv} [F]
4	59.94
8	54.59
12	51.07
16	48.5
20	46.54
24	45.02
28	43.82
32	42.88
36	42.16
40	41.61

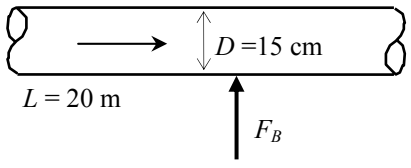
The table is for $T_{ambient}$=60°F



1-108 One section of the duct of an air-conditioning system is laid underwater. The upward force the water will exert on the duct is to be determined.

Assumptions **1** The diameter given is the outer diameter of the duct (or, the thickness of the duct material is negligible). **2** The weight of the duct and the air in is negligible.

Properties The density of air is given to be ρ = 1.30 kg/m^3. We take the density of water to be 1000 kg/m^3.



Analysis Noting that the weight of the duct and the air in it is negligible, the net upward force acting on the duct is the buoyancy force exerted by water. The volume of the underground section of the duct is

$$V = AL = (\pi D^2/4)L = [\pi(0.15\,\text{m})^2/4](20\,\text{m}) = 0.353\ \text{m}^3$$

Then the buoyancy force becomes

$$F_B = \rho g V = (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.353\ \text{m}^3)\left(\frac{1\,\text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{3.46\ kN}$$

Discussion The upward force exerted by water on the duct is 3.46 kN, which is equivalent to the weight of a mass of 353 kg. Therefore, this force must be treated seriously.

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1-109 A helium balloon tied to the ground carries 2 people. The acceleration of the balloon when it is first released is to be determined.

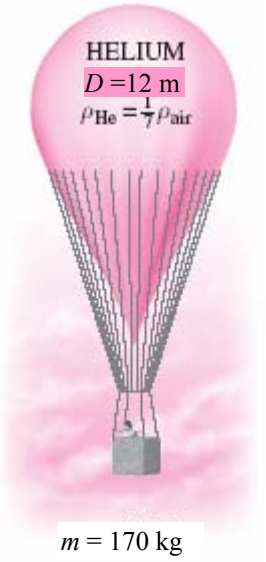
Assumptions The weight of the cage and the ropes of the balloon is negligible.

Properties The density of air is given to be $\rho = 1.16$ kg/m^3. The density of helium gas is 1/7th of this.

Analysis The buoyancy force acting on the balloon is

$$\boldsymbol{V}_{balloon} = 4\pi r^3/3 = 4\pi(6\text{ m})^3/3 = 904.8\text{ m}^3$$

$$F_B = \rho_{air} g \boldsymbol{V}_{balloon}$$
$$= (1.16\text{ kg/m}^3)(9.81\text{m/s}^2)(904.8\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 10{,}296\text{ N}$$



The total mass is

$$m_{\text{He}} = \rho_{\text{He}} \boldsymbol{V} = \left(\frac{1.16}{7}\text{kg/m}^3\right)(904.8\text{ m}^3) = 149.9\text{ kg}$$

$$m_{\text{total}} = m_{\text{He}} + m_{\text{people}} = 149.9 + 2\times 85 = 319.9\text{ kg}$$

The total weight is

$$W = m_{\text{total}} g = (319.9\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 3138\text{ N}$$

Thus the net force acting on the balloon is

$$F_{\text{net}} = F_B - W = 10{,}296 - 3138 = 7157\text{ N}$$

Then the acceleration becomes

$$a = \frac{F_{\text{net}}}{m_{\text{total}}} = \frac{7157\text{ N}}{319.9\text{ kg}}\left(\frac{1\text{kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{22.4\ m/s^2}$$

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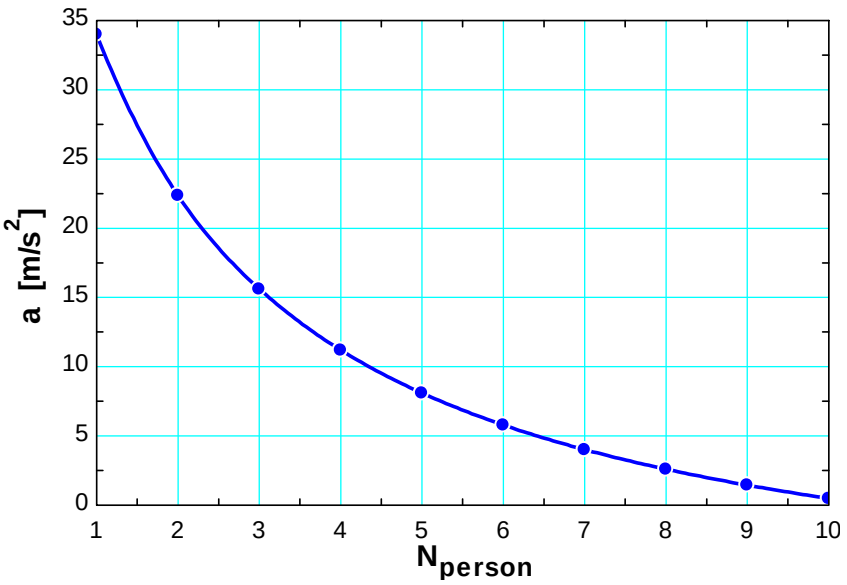
1-110 Problem 1-109 is reconsidered. The effect of the number of people carried in the balloon on acceleration is to be investigated. Acceleration is to be plotted against the number of people, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
D=12 [m]
N_person=2
m_person=85 [kg]
rho_air=1.16 [kg/m^3]
rho_He=rho_air/7

"Analysis"
g=9.81 [m/s^2]
V_ballon=pi*D^3/6
F_B=rho_air*g*V_ballon
m_He=rho_He*V_ballon
m_people=N_person*m_person
m_total=m_He+m_people
W=m_total*g
F_net=F_B-W
a=F_net/m_total
```

N_{person}	a $[m/s^2]$
1	34
2	22.36
3	15.61
4	11.2
5	8.096
6	5.79
7	4.01
8	2.595
9	1.443
10	0.4865



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1-111 A balloon is filled with helium gas. The maximum amount of load the balloon can carry is to be determined.

Assumptions The weight of the cage and the ropes of the balloon is negligible.

Properties The density of air is given to be $\rho = 1.16$ kg/m^3. The density of helium gas is 1/7th of this.



Analysis

The buoyancy force acting on the balloon is

$$\boldsymbol{V}_{balloon} = 4\pi r^3/3 = 4\pi(6\text{ m})^3/3 = 904.8\text{ m}^3$$

$$F_B = \rho_{\text{air}}\, g\, \boldsymbol{V}_{\text{balloon}}$$
$$= (1.16\text{ kg/m}^3)(9.81\text{m/s}^2)(904.8\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 10{,}296\text{ N}$$

The mass of helium is

$$m_{\text{He}} = \rho_{\text{He}}\boldsymbol{V} = \left(\frac{1.16}{7}\text{kg/m}^3\right)(904.8\text{ m}^3) = 149.9\text{ kg}$$

In the limiting case, the net force acting on the balloon will be zero. That is, the buoyancy force and the weight will balance each other:

$$W = mg = F_B$$

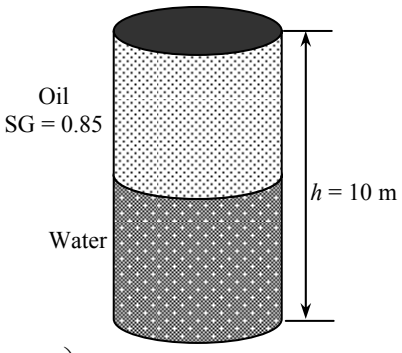
$$m_{\text{total}} = \frac{F_B}{g} = \frac{10{,}296\text{ N}}{9.81\text{ m/s}^2} = 1050\text{ kg}$$

Thus,

$$m_{\text{people}} = m_{\text{total}} - m_{\text{He}} = 1050 - 149.9 = \mathbf{900\text{ kg}}$$

1-112 A 10-m high cylindrical container is filled with equal volumes of water and oil. The pressure difference between the top and the bottom of the container is to be determined.

Properties The density of water is given to be $\rho = 1000$ kg/m^3. The specific gravity of oil is given to be 0.85.



Analysis The density of the oil is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{\text{H}_2\text{O}} = (0.85)(1000\text{ kg/m}^3) = 850\text{ kg/m}^3$$

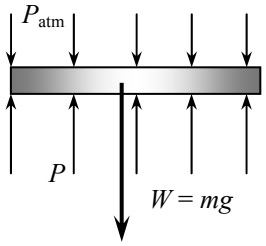
The pressure difference between the top and the bottom of the cylinder is the sum of the pressure differences across the two fluids,

$$\Delta P_{\text{total}} = \Delta P_{\text{oil}} + \Delta P_{\text{water}} = (\rho g h)_{\text{oil}} + (\rho g h)_{\text{water}}$$
$$= \left[(850\text{ kg/m}^3)(9.81\text{ m/s}^2)(5\text{ m}) + (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(5\text{ m})\right]\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right)$$
$$= \mathbf{90.7\text{ kPa}}$$

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1-113 The pressure of a gas contained in a vertical piston-cylinder device is measured to be 180 kPa. The mass of the piston is to be determined.

Assumptions There is no friction between the piston and the cylinder.



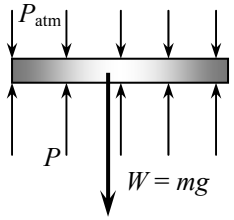
Analysis Drawing the free body diagram of the piston and balancing the vertical forces yield

$$W = PA - P_{atm}A$$
$$mg = (P - P_{atm})A$$
$$(m)(9.81\ \text{m/s}^2) = (180 - 100\ \text{kPa})(25\times10^{-4}\ \text{m}^2)\left(\frac{1000\ \text{kg/m}\cdot\text{s}^2}{1\text{kPa}}\right)$$

It yields $m =$ **20.4 kg**

1-114 The gage pressure in a pressure cooker is maintained constant at 100 kPa by a petcock. The mass of the petcock is to be determined.

Assumptions There is no blockage of the pressure release valve.

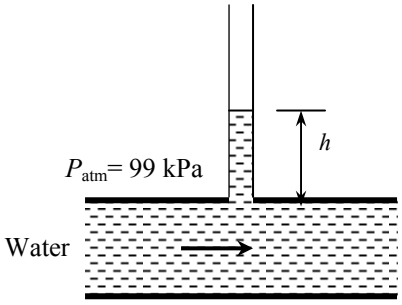


Analysis Atmospheric pressure is acting on all surfaces of the petcock, which balances itself out. Therefore, it can be disregarded in calculations if we use the gage pressure as the cooker pressure. A force balance on the petcock ($\Sigma F_y = 0$) yields

$$W = P_{gage}A$$
$$m = \frac{P_{gage}A}{g} = \frac{(100\ \text{kPa})(4\times10^{-6}\ \text{m}^2)}{9.81\ \text{m/s}^2}\left(\frac{1000\ \text{kg/m}\cdot\text{s}^2}{1\ \text{kPa}}\right)$$
$$= \mathbf{0.0408\ kg}$$

1-115 A glass tube open to the atmosphere is attached to a water pipe, and the pressure at the bottom of the tube is measured. It is to be determined how high the water will rise in the tube.

Properties The density of water is given to be $\rho = 1000\ \text{kg/m}^3$.



Analysis The pressure at the bottom of the tube can be expressed as

$$P = P_{atm} + (\rho g h)_{tube}$$

Solving for h,

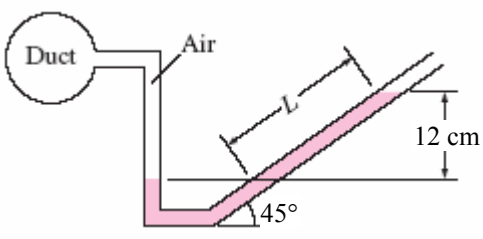
$$h = \frac{P - P_{atm}}{\rho g}$$
$$= \frac{(120 - 99)\ \text{kPa}}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)\left(\frac{1000\ \text{N/m}^2}{1\ \text{kPa}}\right)$$
$$= \mathbf{2.14\ m}$$

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1-116 The air pressure in a duct is measured by an inclined manometer. For a given vertical level difference, the gage pressure in the duct and the length of the differential fluid column are to be determined.

Assumptions The manometer fluid is an incompressible substance.

Properties The density of the liquid is given to be $\rho = 0.81$ kg/L = 810 kg/m^3.



Analysis The gage pressure in the duct is determined from

$$\begin{aligned} P_{\text{gage}} &= P_{\text{abs}} - P_{\text{atm}} = \rho g h \\ &= (810\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.12\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{Pa}}{1\ \text{N/m}^2}\right) \\ &= \mathbf{954\ Pa} \end{aligned}$$

The length of the differential fluid column is

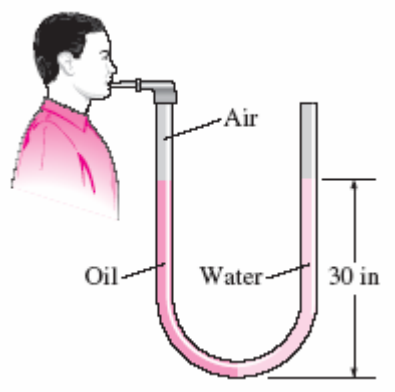
$$L = h / \sin\theta = (12\ \text{cm}) / \sin 45° = \mathbf{17.0\ cm}$$

Discussion Note that the length of the differential fluid column is extended considerably by inclining the manometer arm for better readability.

1-117E Equal volumes of water and oil are poured into a U-tube from different arms, and the oil side is pressurized until the contact surface of the two fluids moves to the bottom and the liquid levels in both arms become the same. The excess pressure applied on the oil side is to be determined.

Assumptions **1** Both water and oil are incompressible substances. **2** Oil does not mix with water. **3** The cross-sectional area of the U-tube is constant.

Properties The density of oil is given to be $\rho_{\text{oil}} = 49.3$ lbm/ft^3. We take the density of water to be $\rho_{\text{w}} = 62.4$ lbm/ft^3.



Analysis Noting that the pressure of both the water and the oil is the same at the contact surface, the pressure at this surface can be expressed as

$$P_{\text{contact}} = P_{\text{blow}} + \rho_{\text{a}} g h_{\text{a}} = P_{\text{atm}} + \rho_{\text{w}} g h_{\text{w}}$$

Noting that $h_a = h_w$ and rearranging,

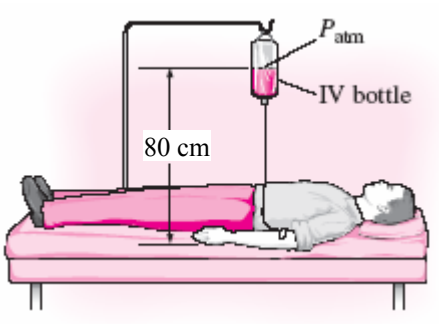
$$\begin{aligned} P_{\text{gage,blow}} &= P_{\text{blow}} - P_{\text{atm}} = (\rho_w - \rho_{oil}) g h \\ &= (62.4 - 49.3\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(30/12\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right) \\ &= \mathbf{0.227\ psi} \end{aligned}$$

Discussion When the person stops blowing, the oil will rise and some water will flow into the right arm. It can be shown that when the curvature effects of the tube are disregarded, the differential height of water will be 23.7 in to balance 30-in of oil.

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1-118 It is given that an IV fluid and the blood pressures balance each other when the bottle is at a certain height, and a certain gage pressure at the arm level is needed for sufficient flow rate. The gage pressure of the blood and elevation of the bottle required to maintain flow at the desired rate are to be determined.

Assumptions **1** The IV fluid is incompressible. **2** The IV bottle is open to the atmosphere.



Properties The density of the IV fluid is given to be $\rho = 1020$ kg/m^3.

Analysis (*a*) Noting that the IV fluid and the blood pressures balance each other when the bottle is 0.8 m above the arm level, the gage pressure of the blood in the arm is simply equal to the gage pressure of the IV fluid at a depth of 0.8 m,

$$\begin{aligned} P_{\text{gage, arm}} &= P_{\text{abs}} - P_{\text{atm}} = \rho g h_{\text{arm-bottle}} \\ &= (1020\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.8\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) \\ &= \mathbf{8.0\ kPa} \end{aligned}$$

(*b*) To provide a gage pressure of 15 kPa at the arm level, the height of the bottle from the arm level is again determined from $P_{\text{gage, arm}} = \rho g h_{\text{arm-bottle}}$ to be

$$\begin{aligned} h_{\text{arm-bottle}} &= \frac{P_{\text{gage, arm}}}{\rho g} \\ &= \frac{15\ \text{kPa}}{(1020\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right) = \mathbf{1.5\ m} \end{aligned}$$

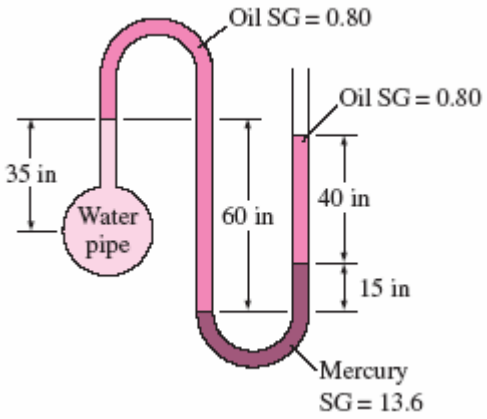
Discussion Note that the height of the reservoir can be used to control flow rates in gravity driven flows. When there is flow, the pressure drop in the tube due to friction should also be considered. This will result in raising the bottle a little higher to overcome pressure drop.

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1-119E A water pipe is connected to a double-U manometer whose free arm is open to the atmosphere. The absolute pressure at the center of the pipe is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The solubility of the liquids in each other is negligible.

Properties The specific gravities of mercury and oil are given to be 13.6 and 0.80, respectively. We take the density of water to be $\rho_w = 62.4$ lbm/ft^3.



Analysis Starting with the pressure at the center of the water pipe, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

$$P_{\text{water pipe}} - \rho_{\text{water}} g h_{\text{water}} + \rho_{\text{oil}} g h_{\text{oil}} - \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{oil}} g h_{\text{oil}} = P_{atm}$$

Solving for $P_{\text{water pipe}}$,

$$P_{\text{water pipe}} = P_{atm} + \rho_{\text{water}} g (h_{\text{water}} - SG_{\text{oil}} h_{\text{oil}} + SG_{\text{Hg}} h_{\text{Hg}} + SG_{\text{oil}} h_{\text{oil}})$$

Substituting,

$$\begin{aligned} P_{\text{water pipe}} &= 14.2\,\text{psia} + (62.4\,\text{lbm/ft}^3)(32.2\,\text{ft/s}^2)[(35/12\,\text{ft}) - 0.8(60/12\,\text{ft}) + 13.6(15/12\,\text{ft}) \\ &\quad + 0.8(40/12\,\text{ft})] \times \left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right) \\ &= \mathbf{22.3\ psia} \end{aligned}$$

Therefore, the absolute pressure in the water pipe is 22.3 psia.

Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

1-120 The average atmospheric pressure is given as $P_{\text{atm}} = 101.325(1 - 0.02256z)^{5.256}$ where z is the altitude in km. The atmospheric pressures at various locations are to be determined.

Analysis The atmospheric pressures at various locations are obtained by substituting the altitude z values in km into the relation

$$P_{atm} = 101.325(1 - 0.02256z)^{5.256}$$

Atlanta: (z = 0.306 km): $P_{\text{atm}} = 101.325(1 - 0.02256\times 0.306)^{5.256}$ = **97.7 kPa**

Denver: (z = 1.610 km): $P_{\text{atm}} = 101.325(1 - 0.02256\times 1.610)^{5.256}$ = **83.4 kPa**

M. City: (z = 2.309 km): $P_{\text{atm}} = 101.325(1 - 0.02256\times 2.309)^{5.256}$ = **76.5 kPa**

Mt. Ev.: (z = 8.848 km): $P_{\text{atm}} = 101.325(1 - 0.02256\times 8.848)^{5.256}$ = **31.4 kPa**

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1-121 The temperature of the atmosphere varies with altitude z as $T = T_0 - \beta z$, while the gravitational acceleration varies by $g(z) = g_0/(1+z/6{,}370{,}320)^2$. Relations for the variation of pressure in atmosphere are to be obtained (*a*) by ignoring and (*b*) by considering the variation of g with altitude.

Assumptions The air in the troposphere behaves as an ideal gas.

Analysis (*a*) Pressure change across a differential fluid layer of thickness dz in the vertical z direction is

$$dP = -\rho g dz$$

From the ideal gas relation, the air density can be expressed as $\rho = \frac{P}{RT} = \frac{P}{R(T_0 - \beta z)}$. Then,

$$dP = -\frac{P}{R(T_0 - \beta z)} g dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} \frac{dP}{P} = -\int_0^z \frac{g dz}{R(T_0 - \beta z)}$$

Performing the integrations.

$$\ln \frac{P}{P_0} = \frac{g}{R\beta} \ln \frac{T_0 - \beta z}{T_0}$$

Rearranging, the desired relation for atmospheric pressure for the case of constant g becomes

$$P = P_0 \left(1 - \frac{\beta z}{T_0}\right)^{\frac{g}{\beta R}}$$

(*b*) When the variation of g with altitude is considered, the procedure remains the same but the expressions become more complicated,

$$dP = -\frac{P}{R(T_0 - \beta z)} \frac{g_0}{(1+z/6{,}370{,}320)^2} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} \frac{dP}{P} = -\int_0^z \frac{g_0 dz}{R(T_0 - \beta z)(1+z/6{,}370{,}320)^2}$$

Performing the integrations,

$$\ln P \Big|_{P_0}^{P} = \frac{g_0}{R\beta} \left| \frac{1}{(1+kT_0/\beta)(1+kz)} - \frac{1}{(1+kT_0/\beta)^2} \ln \frac{1+kz}{T_0 - \beta z} \right|_0^z$$

where R = 287 J/kg·K = 287 m^2/s^2·K is the gas constant of air. After some manipulations, we obtain

$$P = P_0 \exp\left[-\frac{g_0}{R(\beta + kT_0)} \left(\frac{1}{1+1/kz} + \frac{1}{1+kT_0/\beta} \ln \frac{1+kz}{1-\beta z/T_0} \right) \right]$$

where T_0 = 288.15 K, β = 0.0065 K/m, g_0 = 9.807 m/s^2, k = 1/6,370,320 m^{-1}, and z is the elevation in m.

Discussion When performing the integration in part (*b*), the following expression from integral tables is used, together with a transformation of variable $x = T_0 - \beta z$,

$$\int \frac{dx}{x(a+bx)^2} = \frac{1}{a(a+bx)} - \frac{1}{a^2} \ln \frac{a+bx}{x}$$

Also, for z = 11,000 m, for example, the relations in (*a*) and (*b*) give 22.62 and 22.69 kPa, respectively.

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1-122 The variation of pressure with density in a thick gas layer is given. A relation is to be obtained for pressure as a function of elevation z.

Assumptions The property relation $P = C\rho^n$ is valid over the entire region considered.

Analysis The pressure change across a differential fluid layer of thickness dz in the vertical z direction is given as,

$$dP = -\rho g dz$$

Also, the relation $P = C\rho^n$ can be expressed as $C = P/\rho^n = P_0/\rho_0^n$, and thus

$$\rho = \rho_0 (P/P_0)^{1/n}$$

Substituting,

$$dP = -g\rho_0 (P/P_0)^{1/n} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0 = C\rho_0^n$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} (P/P_0)^{-1/n} dP = -\rho_0 g \int_0^z dz$$

Performing the integrations.

$$P_0 \frac{(P/P_0)^{-1/n+1}}{-1/n+1}\bigg|_{P_0}^{P} = -\rho_0 gz \quad \rightarrow \quad \left(\frac{P}{P_0}\right)^{(n-1)/n} - 1 = -\frac{n-1}{n}\frac{\rho_0 gz}{P_0}$$

Solving for P,

$$P = P_0\left(1 - \frac{n-1}{n}\frac{\rho_0 gz}{P_0}\right)^{n/(n-1)}$$

which is the desired relation.

Discussion The final result could be expressed in various forms. The form given is very convenient for calculations as it facilitates unit cancellations and reduces the chance of error.

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1-123 A pressure transducers is used to measure pressure by generating analogue signals, and it is to be calibrated by measuring both the pressure and the electric current simultaneously for various settings, and the results are tabulated. A calibration curve in the form of $P = aI + b$ is to be obtained, and the pressure corresponding to a signal of 10 mA is to be calculated.

Assumptions Mercury is an incompressible liquid.

Properties The specific gravity of mercury is given to be 13.56, and thus its density is 13,560 kg/m^3.

Analysis For a given differential height, the pressure can be calculated from

$$P = \rho g \Delta h$$

For Δh = 28.0 mm = 0.0280 m, for example,

$$P = 13.56(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.0280\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) = 3.75\ \text{kPa}$$

Repeating the calculations and tabulating, we have

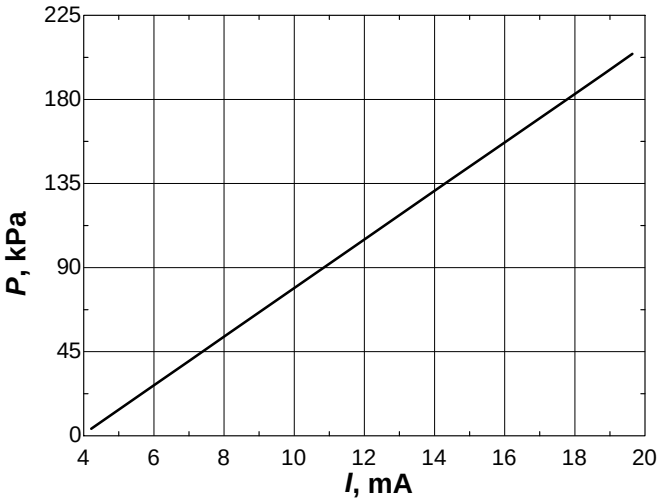
Δh(mm)	28.0	181.5	297.8	413.1	765.9	1027	1149	1362	1458	1536
***P*(kPa)**	**3.73**	**24.14**	**39.61**	**54.95**	**101.9**	**136.6**	**152.8**	**181.2**	**193.9**	**204.3**
I (mA)	4.21	5.78	6.97	8.15	11.76	14.43	15.68	17.86	18.84	19.64

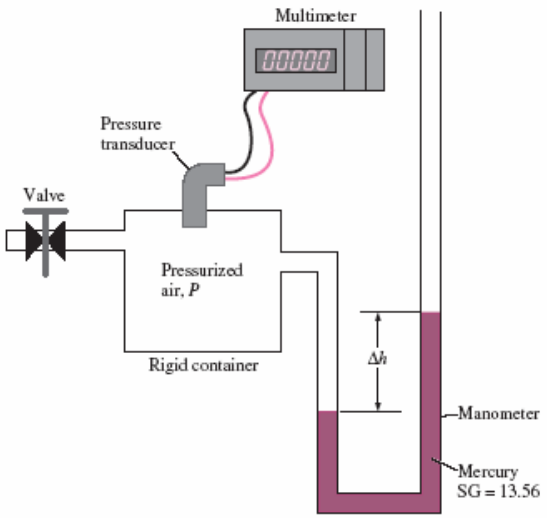
A plot of P versus I is given below. It is clear that the pressure varies linearly with the current, and using EES, the best curve fit obtained to be

$P = 13.00I - 51.00$ (kPa) for $4.21 \le I \le 19.64$.

For I = 10 mA, for example, we would get

$P =$ **79.0 kPa**





Discussion Note that the calibration relation is valid in the specified range of currents or pressures.

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1-124 The flow of air through a wind turbine is considered. Based on unit considerations, a proportionality relation is to be obtained for the mass flow rate of air through the blades.

Assumptions Wind approaches the turbine blades with a uniform velocity.

Analysis The mass flow rate depends on the air density, average wind velocity, and the cross-sectional area which depends on hose diameter. Also, the unit of mass flow rate $\dot{m}$ is kg/s. Therefore, the independent quantities should be arranged such that we end up with the proper unit. Putting the given information into perspective, we have

$\dot{m}$ [kg/s] is a function of ρ [kg/m^3], D [m], and V [m/s}

It is obvious that the only way to end up with the unit "kg/s" for mass flow rate is to multiply the quantities ρ and V with the square of D. Therefore, the desired proportionality relation is

$\dot{m}$ is proportional to $\rho D^2 V$

or,

$$\dot{m} = C\rho D^2 V$$

where the constant of proportionality is $C = \pi/4$ so that $\dot{m} = \rho(\pi D^2 / 4)V$

Discussion Note that the dimensionless constants of proportionality cannot be determined with this approach.

1-125 A relation for the air drag exerted on a car is to be obtained in terms of on the drag coefficient, the air density, the car velocity, and the frontal area of the car.

Analysis The drag force depends on a dimensionless drag coefficient, the air density, the car velocity, and the frontal area. Also, the unit of force F is newton N, which is equivalent to kg·m/s^2. Therefore, the independent quantities should be arranged such that we end up with the unit kg·m/s^2 for the drag force. Putting the given information into perspective, we have

F_D [kg·m/s^2] $\leftrightarrow$ C_{Drag} [], A_{front} [m^2], ρ [kg/m^3], and V [m/s]

It is obvious that the only way to end up with the unit "kg·m/s^2" for drag force is to multiply mass with the square of the velocity and the fontal area, with the drag coefficient serving as the constant of proportionality. Therefore, the desired relation is

$$F_D = C_{\text{Drag}} \rho A_{\text{front}} V^2$$

Discussion Note that this approach is not sensitive to dimensionless quantities, and thus a strong reasoning is required.

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Fundamentals of Engineering (FE) Exam Problems

1-126 Consider a fish swimming 5 m below the free surface of water. The increase in the pressure exerted on the fish when it dives to a depth of 25 m below the free surface is

(a) 196 Pa (b) 5400 Pa (c) 30,000 Pa (d) 196,000 Pa (e) 294,000 Pa

Answer (*d*) 196,000 Pa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1000 "kg/m3"
g=9.81 "m/s2"
z1=5 "m"
z2=25 "m"
DELTAP=rho*g*(z2-z1) "Pa"

"Some Wrong Solutions with Common Mistakes:"
W1_P=rho*g*(z2-z1)/1000    "dividing by 1000"
W2_P=rho*g*(z1+z2)     "adding depts instead of subtracting"
W3_P=rho*(z1+z2)     "not using g"
W4_P=rho*g*(0+z2)      "ignoring z1"
```

1-127 The atmospheric pressures at the top and the bottom of a building are read by a barometer to be 96.0 and 98.0 kPa. If the density of air is 1.0 kg/m^3, the height of the building is

(a) 17 m (b) 20 m (c) 170 m (d) 204 m (e) 252 m

Answer (d) 204 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1.0 "kg/m3"
g=9.81 "m/s2"
P1=96 "kPa"
P2=98 "kPa"
DELTAP=P2-P1 "kPa"
DELTAP=rho*g*h/1000 "kPa"

"Some Wrong Solutions with Common Mistakes:"
DELTAP=rho*W1_h/1000        "not using g"
DELTAP=g*W2_h/1000          "not using rho"
P2=rho*g*W3_h/1000           "ignoring P1"
P1=rho*g*W4_h/1000           "ignoring P2"
```

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1-128 An apple loses 4.5 kJ of heat as it cools per °C drop in its temperature. The amount of heat loss from the apple per °F drop in its temperature is

(a) 1.25 kJ (b) 2.50 kJ (c) 5.0 kJ (d) 8.1 kJ (e) 4.1 kJ

Answer (b) 2.50 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q_perC=4.5 "kJ"
Q_perF=Q_perC/1.8 "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_Q=Q_perC*1.8 "multiplying instead of dividing"
W2_Q=Q_perC "setting them equal to each other"
```

1-129 Consider a 2-m deep swimming pool. The pressure difference between the top and bottom of the pool is

(a) 12.0 kPa (b) 19.6 kPa (c) 38.1 kPa (d) 50.8 kPa (e) 200 kPa

Answer (b) 19.6 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1000 "kg/m^3"
g=9.81 "m/s2"
z1=0 "m"
z2=2 "m"
DELTAP=rho*g*(z2-z1)/1000 "kPa"

"Some Wrong Solutions with Common Mistakes:"
W1_P=rho*(z1+z2)/1000     "not using g"
W2_P=rho*g*(z2-z1)/2000       "taking half of z"
W3_P=rho*g*(z2-z1) "not dividing by 1000"
```

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1-130 At sea level, the weight of 1 kg mass in SI units is 9.81 N. The weight of 1 lbm mass in English units is

(a) 1 lbf (b) 9.81 lbf (c) 32.2 lbf (d) 0.1 lbf (e) 0.031 lbf

Answer (a) 1 lbf

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=1 "lbm"
g=32.2 "ft/s2"
W=m*g/32.2 "lbf"

"Some Wrong Solutions with Common Mistakes:"
gSI=9.81 "m/s2"
W1_W= m*gSI "Using wrong conversion"
W2_W= m*g "Using wrong conversion"
W3_W= m/gSI "Using wrong conversion"
W4_W= m/g "Using wrong conversion"
```

1-131 During a heating process, the temperature of an object rises by 10°C. This temperature rise is equivalent to a temperature rise of

(a) 10°F (b) 42°F (c) 18 K (d) 18 R (e) 283 K

Answer (d) 18 R

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T_inC=10 "C"
T_inR=T_inC*1.8 "R"

"Some Wrong Solutions with Common Mistakes:"
W1_TinF=T_inC "F, setting C and F equal to each other"
W2_TinF=T_inC*1.8+32 "F, converting to F "
W3_TinK=1.8*T_inC "K, wrong conversion from C to K"
W4_TinK=T_inC+273 "K, converting to K"
```

1-132, 1-133 Design and Essay Problems



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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 2
ENERGY, ENERGY TRANSFER, AND GENERAL ENERGY ANALYSIS

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Forms of Energy

2-1C The *macroscopic* forms of energy are those a system possesses as a whole with respect to some outside reference frame. The *microscopic* forms of energy, on the other hand, are those related to the molecular structure of a system and the degree of the molecular activity, and are independent of outside reference frames.

2-2C The sum of all forms of the energy a system possesses is called *total energy*. In the absence of magnetic, electrical and surface tension effects, the total energy of a system consists of the kinetic, potential, and internal energies.

2-3C Thermal energy is the sensible and latent forms of internal energy, and it is referred to as heat in daily life.

2-4C The *mechanical energy* is the form of energy that can be converted to mechanical work completely and directly by a mechanical device such as a propeller. It differs from thermal energy in that thermal energy cannot be converted to work directly and completely. The forms of mechanical energy of a fluid stream are kinetic, potential, and flow energies.

2-5C Hydrogen is also a fuel, since it can be burned, but it is not an energy source since there are no hydrogen reserves in the world. Hydrogen can be obtained from water by using another energy source, such as solar or nuclear energy, and then the hydrogen obtained can used as a fuel to power cars or generators. Therefore, it is more proper to view hydrogen is an energy carrier than an energy source.

2-6E The total kinetic energy of an object is given is to be determined.

Analysis The total kinetic energy of the object is given by

$$\text{KE} = m\frac{V^2}{2} = (15\,\text{lbm})\frac{(100\,\text{ft/s})^2}{2}\left(\frac{1\,\text{Btu/lbm}}{25{,}037\,\text{ft}^2/\text{s}^2}\right) = \mathbf{3.00\ Btu}$$

2-7 The total kinetic energy of an object is given is to be determined.

Analysis The total kinetic energy of the object is given by

$$\text{KE} = m\frac{V^2}{2} = (100\,\text{kg})\frac{(20\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{20.0\ kJ}$$

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2-8E The specific potential energy of an object is to be determined.

Analysis In the English unit system, the specific potential energy in Btu is given by

$$\text{pe} = gz = (32.1\,\text{ft/s}^2)(100\,\text{ft})\left(\frac{1\,\text{Btu/lbm}}{25{,}037\,\text{ft}^2/\text{s}^2}\right) = \mathbf{0.128\,Btu/lbm}$$

2-9E The total potential energy of an object is to be determined.

Analysis Substituting the given data into the potential energy expression gives

$$\text{PE} = mgz = (200\,\text{lbm})(32.2\,\text{ft/s}^2)(10\,\text{ft})\left(\frac{1\,\text{Btu/lbm}}{25{,}037\,\text{ft}^2/\text{s}^2}\right) = \mathbf{2.57\,Btu}$$

2-10 The total potential energy of an object that is below a reference level is to be determined.

Analysis Substituting the given data into the potential energy expression gives

$$\text{PE} = mgz = (20\,\text{kg})(9.5\,\text{m/s}^2)(-20\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{-3.8\,kJ}$$

2-11 A person with his suitcase goes up to the 10$^{\text{th}}$ floor in an elevator. The part of the energy of the elevator stored in the suitcase is to be determined.

Assumptions **1** The vibrational effects in the elevator are negligible.

Analysis The energy stored in the suitcase is stored in the form of potential energy, which is *mgz*. Therefore,

$$\Delta E_{\text{suitcase}} = \Delta PE = mg\Delta z = (30\,\text{kg})(9.81\,\text{m/s}^2)(35\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{10.3\,kJ}$$

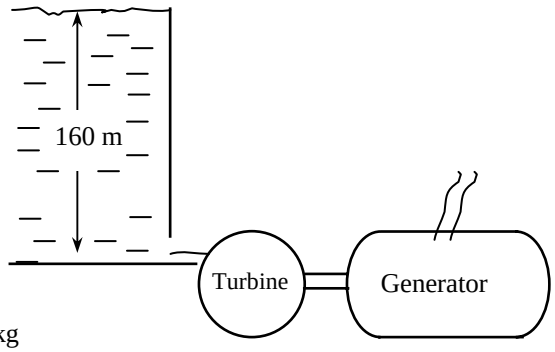
Therefore, the suitcase on 10$^{\text{th}}$ floor has 10.3 kJ more energy compared to an identical suitcase on the lobby level.

Discussion Noting that 1 kWh = 3600 kJ, the energy transferred to the suitcase is 10.3/3600 = 0.0029 kWh, which is very small.

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2-12 A hydraulic turbine-generator is to generate electricity from the water of a large reservoir. The power generation potential is to be determined.

Assumptions **1** The elevation of the reservoir remains constant. **2** The mechanical energy of water at the turbine exit is negligible.



Analysis The total mechanical energy water in a reservoir possesses is equivalent to the potential energy of water at the free surface, and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is *gz* per unit mass, and $\dot{m}gz$ for a given mass flow rate.

$$e_{\text{mech}} = pe = gz = (9.81\,\text{m/s}^2)(160\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 1.574\,\text{kJ/kg}$$

Then the power generation potential becomes

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (3500\,\text{kg/s})(1.574\,\text{kJ/kg})\left(\frac{1\,\text{kW}}{1\,\text{kJ/s}}\right) = \mathbf{5509\,kW}$$

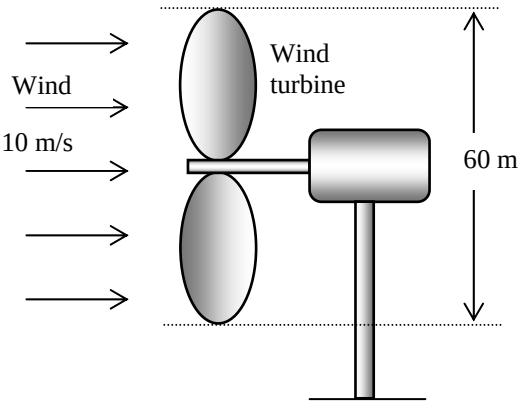
Therefore, the reservoir has the potential to generate 1766 kW of power.

Discussion This problem can also be solved by considering a point at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

2-13 Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass and the power generation potential are to be determined.

Assumptions The wind is blowing steadily at a constant uniform velocity.

Properties The density of air is given to be $\rho = 1.25\,\text{kg/m}^3$.



Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(10\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 0.050\,\text{kJ/kg}$$

$$\dot{m} = \rho VA = \rho V\frac{\pi D^2}{4} = (1.25\,\text{kg/m}^3)(10\,\text{m/s})\frac{\pi(60\,\text{m})^2}{4} = 35{,}340\,\text{kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (35{,}340\,\text{kg/s})(0.050\,\text{kJ/kg}) = \mathbf{1770\ kW}$$

Therefore, 1770 kW of actual power can be generated by this wind turbine at the stated conditions.

Discussion The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.

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2-14 A water jet strikes the buckets located on the perimeter of a wheel at a specified velocity and flow rate. The power generation potential of this system is to be determined.

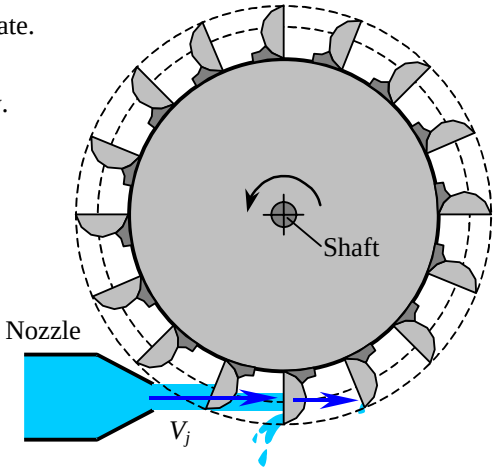
Assumptions Water jet flows steadily at the specified speed and flow rate.

Analysis Kinetic energy is the only form of harvestable mechanical energy the water jet possesses, and it can be converted to work entirely. Therefore, the power potential of the water jet is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(60\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 1.8\,\text{kJ/kg}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}}$$

$$= (120\,\text{kg/s})(1.8\,\text{kJ/kg})\left(\frac{1\,\text{kW}}{1\,\text{kJ/s}}\right) = \mathbf{216\ kW}$$



Therefore, 216 kW of power can be generated by this water jet at the stated conditions.

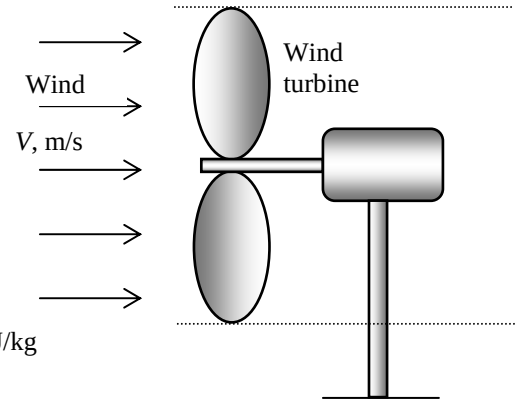
Discussion An actual hydroelectric turbine (such as the Pelton wheel) can convert over 90% of this potential to actual electric power.

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2-15 Two sites with specified wind data are being considered for wind power generation. The site better suited for wind power generation is to be determined.

Assumptions **1**The wind is blowing steadily at specified velocity during specified times. **2** The wind power generation is negligible during other times.

Properties We take the density of air to be $\rho = 1.25$ kg/m^3 (it does not affect the final answer).



Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate. Considering a unit flow area ($A = 1$ m^2), the maximum wind power and power generation becomes

$$e_{\text{mech},1} = ke_1 = \frac{V_1^2}{2} = \frac{(7\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.0245\text{ kJ/kg}$$

$$e_{\text{mech},2} = ke_2 = \frac{V_2^2}{2} = \frac{(10\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.050\text{ kJ/kg}$$

$$\dot{W}_{\max,1} = \dot{E}_{\text{mech},1} = \dot{m}_1 e_{\text{mech},1} = \rho V_1 A k e_1 = (1.25\text{ kg/m}^3)(7\text{ m/s})(1\text{ m}^2)(0.0245\text{ kJ/kg}) = 0.2144\text{ kW}$$

$$\dot{W}_{\max,2} = \dot{E}_{\text{mech},2} = \dot{m}_2 e_{\text{mech},2} = \rho V_2 A k e_2 = (1.25\text{ kg/m}^3)(10\text{ m/s})(1\text{ m}^2)(0.050\text{ kJ/kg}) = 0.625\text{ kW}$$

since 1 kW = 1 kJ/s. Then the maximum electric power generations per year become

$$E_{\max,1} = \dot{W}_{\max,1}\Delta t_1 = (0.2144\text{ kW})(3000\text{ h/yr}) = \mathbf{643\ kWh/yr}\ (\text{per m}^2\ \text{flow area})$$

$$E_{\max,2} = \dot{W}_{\max,2}\Delta t_2 = (0.625\text{ kW})(2000\text{ h/yr}) = \mathbf{1250\ kWh/yr}\ (\text{per m}^2\ \text{flow area})$$

Therefore, **second site** is a better one for wind generation.

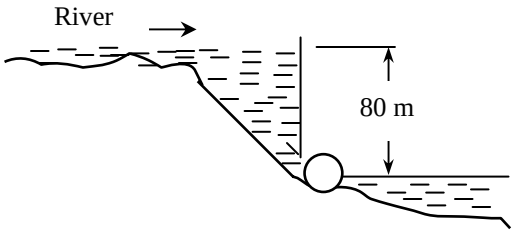
Discussion Note the power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the average wind velocity is the primary consideration in wind power generation decisions.

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2-16 A river flowing steadily at a specified flow rate is considered for hydroelectric power generation by collecting the water in a dam. For a specified water height, the power generation potential is to be determined.

Assumptions **1** The elevation given is the elevation of the free surface of the river. **2** The mechanical energy of water at the turbine exit is negligible.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.



Analysis The total mechanical energy the water in a dam possesses is equivalent to the potential energy of water at the free surface of the dam (relative to free surface of discharge water), and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is *gz* per unit mass, and $\dot{m}gz$ for a given mass flow rate.

$$e_{\text{mech}} = pe = gz = (9.81\,\text{m/s}^2)(80\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 0.7848\,\text{kJ/kg}$$

The mass flow rate is

$$\dot{m} = \rho \dot{V} = (1000\,\text{kg/m}^3)(175\,\text{m}^3/\text{s}) = 175{,}000\,\text{kg/s}$$

Then the power generation potential becomes

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m} e_{\text{mech}} = (175{,}000\,\text{kg/s})(0.7848\,\text{kJ/kg})\left(\frac{1\,\text{MW}}{1000\,\text{kJ/s}}\right) = \mathbf{137\ MW}$$

Therefore, 137 MW of power can be generated from this river if its power potential can be recovered completely.

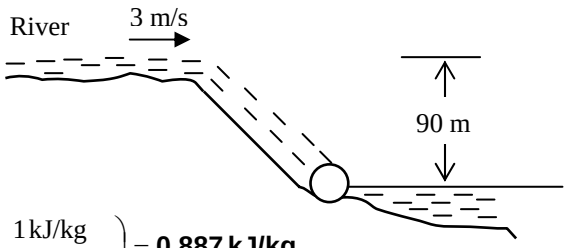
Discussion Note that the power output of an actual turbine will be less than 137 MW because of losses and inefficiencies.

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2-17 A river is flowing at a specified velocity, flow rate, and elevation. The total mechanical energy of the river water per unit mass, and the power generation potential of the entire river are to be determined.

Assumptions **1** The elevation given is the elevation of the free surface of the river. **2** The velocity given is the average velocity. **3** The mechanical energy of water at the turbine exit is negligible.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.



Analysis Noting that the sum of the flow energy and the potential energy is constant for a given fluid body, we can take the elevation of the entire river water to be the elevation of the free surface, and ignore the flow energy. Then the total mechanical energy of the river water per unit mass becomes

$$e_{\text{mech}} = pe + ke = gh + \frac{V^2}{2} = \left((9.81\,\text{m/s}^2)(90\,\text{m}) + \frac{(3\,\text{m/s})^2}{2} \right) \left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2} \right) = \mathbf{0.887\ kJ/kg}$$

The power generation potential of the river water is obtained by multiplying the total mechanical energy by the mass flow rate,

$$\dot{m} = \rho \dot{V} = (1000\,\text{kg/m}^3)(500\,\text{m}^3/\text{s}) = 500{,}000\,\text{kg/s}$$

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m} e_{\text{mech}} = (500{,}000\,\text{kg/s})(0.887\ \text{kJ/kg}) = 444{,}000\,\text{kW} = \mathbf{444\ MW}$$

Therefore, 444 MW of power can be generated from this river as it discharges into the lake if its power potential can be recovered completely.

Discussion Note that the kinetic energy of water is negligible compared to the potential energy, and it can be ignored in the analysis. Also, the power output of an actual turbine will be less than 444 MW because of losses and inefficiencies.

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Energy Transfer by Heat and Work

2-18C Energy can cross the boundaries of a closed system in two forms: heat and work.

2-19C The form of energy that crosses the boundary of a closed system because of a temperature difference is heat; all other forms are work.

2-20C An adiabatic process is a process during which there is no heat transfer. A system that does not exchange any heat with its surroundings is an adiabatic system.

2-21C Point functions depend on the state only whereas the path functions depend on the path followed during a process. Properties of substances are point functions, heat and work are path functions.

2-22C (*a*) The car's radiator transfers heat from the hot engine cooling fluid to the cooler air. No work interaction occurs in the radiator.

(*b*) The hot engine transfers heat to cooling fluid and ambient air while delivering work to the transmission.

(*c*) The warm tires transfer heat to the cooler air and to some degree to the cooler road while no work is produced. No work is produced since there is no motion of the forces acting at the interface between the tire and road.

(*d*) There is minor amount of heat transfer between the tires and road. Presuming that the tires are hotter than the road, the heat transfer is from the tires to the road. There is no work exchange associated with the road since it cannot move.

(*e*) Heat is being added to the atmospheric air by the hotter components of the car. Work is being done on the air as it passes over and through the car.

2-23C When the length of the spring is changed by applying a force to it, the interaction is a work interaction since it involves a force acting through a displacement. A heat interaction is required to change the temperature (and, hence, length) of the spring.

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2-24C (*a*) From the perspective of the contents, heat must be removed in order to reduce and maintain the content's temperature. Heat is also being added to the contents from the room air since the room air is hotter than the contents.

(*b*) Considering the system formed by the refrigerator box when the doors are closed, there are three interactions, electrical work and two heat transfers. There is a transfer of heat from the room air to the refrigerator through its walls. There is also a transfer of heat from the hot portions of the refrigerator (i.e., back of the compressor where condenser is placed) system to the room air. Finally, electrical work is being added to the refrigerator through the refrigeration system.

(*c*) Heat is transferred through the walls of the room from the warm room air to the cold winter air. Electrical work is being done on the room through the electrical wiring leading into the room.

2-25C (*a*) As one types on the keyboard, electrical signals are produced and transmitted to the processing unit. Simultaneously, the temperature of the electrical parts is increased slightly. The work done on the keys when they are depressed is work done on the system (i.e., keyboard). The flow of electrical current (with its voltage drop) does work on the keyboard. Since the temperature of the electrical parts of the keyboard is somewhat higher than that of the surrounding air, there is a transfer of heat from the keyboard to the surrounding air.

(*b*) The monitor is powered by the electrical current supplied to it. This current (and voltage drop) is work done on the system (i.e., monitor). The temperatures of the electrical parts of the monitor are higher than that of the surrounding air. Hence there is a heat transfer to the surroundings.

(*c*) The processing unit is like the monitor in that electrical work is done on it while it transfers heat to the surroundings.

(*d*) The entire unit then has electrical work done on it, and mechanical work done on it to depress the keys. It also transfers heat from all its electrical parts to the surroundings.

2-26 The power produced by an electrical motor is to be expressed in different units.

Analysis Using appropriate conversion factors, we obtain

(*a*) $$\dot{W} = (5\ \text{W})\left(\frac{1\ \text{J/s}}{1\ \text{W}}\right)\left(\frac{1\ \text{N}\cdot\text{m}}{1\ \text{J}}\right) = \mathbf{5\ N\cdot m/s}$$

(*b*) $$\dot{W} = (5\ \text{W})\left(\frac{1\ \text{J/s}}{1\ \text{W}}\right)\left(\frac{1\ \text{N}\cdot\text{m}}{1\ \text{J}}\right)\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = \mathbf{5\ kg\cdot m^2/s^3}$$

2-27E The power produced by a model aircraft engine is to be expressed in different units.

Analysis Using appropriate conversion factors, we obtain

(*a*) $$\dot{W} = (10\ \text{W})\left(\frac{1\ \text{Btu/s}}{1055.056\ \text{W}}\right)\left(\frac{778.169\ \text{lbf}\cdot\text{ft/s}}{1\ \text{Btu/s}}\right) = \mathbf{7.38\ lbf\cdot ft/s}$$

(*b*) $$\dot{W} = (10\ \text{W})\left(\frac{1\ \text{hp}}{745.7\ \text{W}}\right) = \mathbf{0.0134\ hp}$$

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Mechanical Forms of Work

2-28C The work done is the same, but the power is different.

2-29 A car is accelerated from rest to 100 km/h. The work needed to achieve this is to be determined.

Analysis The work needed to accelerate a body the change in kinetic energy of the body,

$$W_a = \frac{1}{2}m(V_2^2 - V_1^2) = \frac{1}{2}(800\ \text{kg})\left(\left(\frac{100{,}000\ \text{m}}{3600\ \text{s}}\right)^2 - 0\right)\left(\frac{1\ \text{kJ}}{1000\ \text{kg}\cdot\text{m}^2/\text{s}^2}\right) = \mathbf{309\ kJ}$$

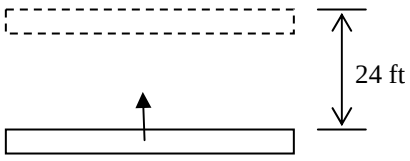
2-30E A construction crane lifting a concrete beam is considered. The amount of work is to be determined considering (a) the beam and (b) the crane as the system.

Analysis (*a*) The work is done on the beam and it is determined from

$$W = mg\Delta z = (2\times 3000\ \text{lbm})(32.174\ \text{ft/s}^2)\left(\frac{1\ \text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/s}^2}\right)(24\ \text{ft})$$

$$= \mathbf{144{,}000\ lbf\cdot ft}$$

$$= (144{,}000\ \text{lbf}\cdot\text{ft})\left(\frac{1\ \text{Btu}}{778.169\ \text{lbf}\cdot\text{ft}}\right) = \mathbf{185\ Btu}$$



(*b*) Since the crane must produce the same amount of work as is required to lift the beam, the work done by the crane is

$$W = \mathbf{144{,}000\ lbf\cdot ft} = \mathbf{185\ Btu}$$

2-31E A man is pushing a cart with its contents up a ramp that is inclined at an angle of 10° from the horizontal. The work needed to move along this ramp is to be determined considering (a) the man and (b) the cart and its contents as the system.

Analysis (*a*) Considering the man as the system, letting l be the displacement along the ramp, and letting θ be the inclination angle of the ramp,

$$W = Fl\sin\theta = (100 + 180\ \text{lbf})(100\ \text{ft})\sin(10) = \mathbf{4862\ lbf\cdot ft}$$

$$= (4862\ \text{lbf}\cdot\text{ft})\left(\frac{1\ \text{Btu}}{778.169\ \text{lbf}\cdot\text{ft}}\right) = \mathbf{6.248\ Btu}$$

This is work that the man must do to raise the weight of the cart and contents, plus his own weight, a distance of $l\sin\theta$.

(*b*) Applying the same logic to the cart and its contents gives

$$W = Fl\sin\theta = (100\ \text{lbf})(100\ \text{ft})\sin(10) = \mathbf{1736\ lbf\cdot ft}$$

$$= (1736\ \text{lbf}\cdot\text{ft})\left(\frac{1\ \text{Btu}}{778.169\ \text{lbf}\cdot\text{ft}}\right) = \mathbf{2.231\ Btu}$$

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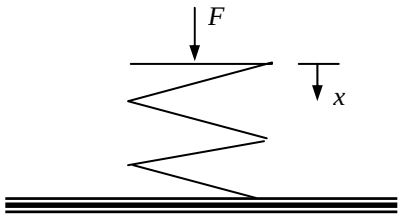
2-32E The work required to compress a spring is to be determined.

Analysis Since there is no preload, $F = kx$. Substituting this into the work expression gives

$$W = \int_1^2 F ds = \int_1^2 kx dx = k\int_1^2 x dx = \frac{k}{2}(x_2^2 - x_1^2)$$

$$= \frac{200\ \text{lbf/in}}{2}\left[(1\ \text{in})^2 - 0^2\right]\left(\frac{1\ \text{ft}}{12\ \text{in}}\right) = \mathbf{8.33\ lbf \cdot ft}$$

$$= (8.33\ \text{lbf} \cdot \text{ft})\left(\frac{1\ \text{Btu}}{778.169\ \text{lbf} \cdot \text{ft}}\right) = \mathbf{0.0107\ Btu}$$



2-33E The work required to expand a soap bubble is to be determined.

Analysis The surface tension work is determined from

$$W = \int_1^2 \sigma_s dA = \sigma(A_1 - A_2) = (0.005\ \text{lbf/ft})4\pi\left[(2/12\ \text{ft})^2 - (0.5/12\ \text{ft})^2\right]$$

$$= 0.00164\ \text{lbf} \cdot \text{ft} = (0.00164\ \text{lbf} \cdot \text{ft})\left(\frac{1\ \text{Btu}}{778.2\ \text{lbf} \cdot \text{ft}}\right) = \mathbf{2.11 \times 10^{-6}\ Btu}$$

2-34E The work required to stretch a steel rod in a specieid length is to be determined.

Assumptions The Young's modulus does not change as the rod is stretched.

Analysis The original volume of the rod is

$$V_0 = \frac{\pi D^2}{4} L = \frac{\pi(0.5\ \text{in})^2}{4}(12\ \text{in}) = 2.356\ \text{in}^3$$

The work required to stretch the rod 0.125 in is

$$W = \frac{V_0 E}{2}(\varepsilon_2^2 - \varepsilon_1^2)$$

$$= \frac{(2.356\ \text{in}^3)(30{,}000\ \text{lbf/in}^2)}{2}\left[(0.125/12\ \text{in})^2 - 0^2\right]$$

$$= 2.835\ \text{lbf} \cdot \text{in} = (2.835\ \text{lbf} \cdot \text{in})\left(\frac{1\ \text{Btu}}{9338\ \text{lbf} \cdot \text{in}}\right) = \mathbf{4.11 \times 10^{-4}\ Btu}$$

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2-35E The work required to compress a spring is to be determined.

Analysis The force at any point during the deflection of the spring is given by $F = F_0 + kx$, where F_0 is the initial force and x is the deflection as measured from the point where the initial force occurred. From the perspective of the spring, this force acts in the direction opposite to that in which the spring is deflected. Then,

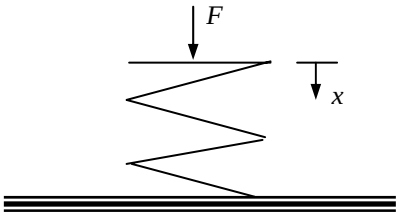
$$W = \int_1^2 F ds = \int_1^2 (F_0 + kx) dx$$

$$= F_0 (x_2 - x_1) + \frac{k}{2}(x_2^2 - x_1^2)$$

$$= (100 \text{ lbf})\left[(1-0)\text{in}\right] + \frac{200 \text{ lbf/in}}{2}(1^2 - 0^2)\text{in}^2$$

$$= 200 \text{ lbf} \cdot \text{in}$$

$$= (200 \text{ lbf} \cdot \text{in})\left(\frac{1 \text{ Btu}}{778.169 \text{ lbf} \cdot \text{ft}}\right)\left(\frac{1 \text{ ft}}{12 \text{ in}}\right) = \mathbf{0.0214 \text{ Btu}}$$



2-36 The work required to compress a spring is to be determined.

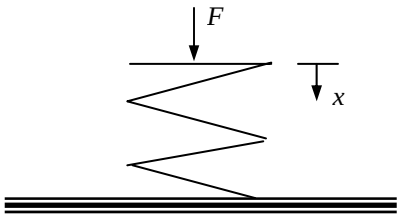
Analysis Since there is no preload, $F = kx$. Substituting this into the work expression gives

$$W = \int_1^2 F ds = \int_1^2 kx dx = k\int_1^2 x dx = \frac{k}{2}(x_2^2 - x_1^2)$$

$$= \frac{300 \text{ kN/m}}{2}\left[(0.03 \text{ m})^2 - 0^2\right]$$

$$= 0.135 \text{ kN} \cdot \text{m}$$

$$= (0.135 \text{ kN} \cdot \text{m})\left(\frac{1 \text{ kJ}}{1 \text{ kN} \cdot \text{m}}\right) = \mathbf{0.135 \text{ kJ}}$$



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2-37 A ski lift is operating steadily at 10 km/h. The power required to operate and also to accelerate this ski lift from rest to the operating speed are to be determined.

Assumptions **1** Air drag and friction are negligible. **2** The average mass of each loaded chair is 250 kg. **3** The mass of chairs is small relative to the mass of people, and thus the contribution of returning empty chairs to the motion is disregarded (this provides a safety factor).

Analysis The lift is 1000 m long and the chairs are spaced 20 m apart. Thus at any given time there are 1000/20 = 50 chairs being lifted. Considering that the mass of each chair is 250 kg, the load of the lift at any given time is

Load = (50 chairs)(250 kg/chair) = 12,500 kg

Neglecting the work done on the system by the returning empty chairs, the work needed to raise this mass by 200 m is

$$W_g = mg(z_2 - z_1) = (12{,}500\ \text{kg})(9.81\ \text{m/s}^2)(200\ \text{m})\left(\frac{1\ \text{kJ}}{1000\ \text{kg}\cdot\text{m}^2/\text{s}^2}\right) = 24{,}525\ \text{kJ}$$

At 10 km/h, it will take

$$\Delta t = \frac{\text{distance}}{\text{velocity}} = \frac{1\ \text{km}}{10\ \text{km/h}} = 0.1\ \text{h} = 360\ \text{s}$$

to do this work. Thus the power needed is

$$\dot{W}_g = \frac{W_g}{\Delta t} = \frac{24{,}525\ \text{kJ}}{360\ \text{s}} = \mathbf{68.1\ kW}$$

The velocity of the lift during steady operation, and the acceleration during start up are

$$V = (10\ \text{km/h})\left(\frac{1\ \text{m/s}}{3.6\ \text{km/h}}\right) = 2.778\ \text{m/s}$$

$$a = \frac{\Delta V}{\Delta t} = \frac{2.778\ \text{m/s - 0}}{5\ \text{s}} = 0.556\ \text{m/s}^2$$

During acceleration, the power needed is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(12{,}500\ \text{kg})\left((2.778\ \text{m/s})^2 - 0\right)\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)/(5\ \text{s}) = 9.6\ \text{kW}$$

Assuming the power applied is constant, the acceleration will also be constant and the vertical distance traveled during acceleration will be

$$h = \frac{1}{2}at^2\sin\alpha = \frac{1}{2}at^2\frac{200\ \text{m}}{1000\ \text{m}} = \frac{1}{2}(0.556\ \text{m/s}^2)(5\ \text{s})^2(0.2) = 1.39\ \text{m}$$

and

$$\dot{W}_g = mg(z_2 - z_1)/\Delta t = (12{,}500\ \text{kg})(9.81\ \text{m/s}^2)(1.39\ \text{m})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{kg}\cdot\text{m}^2/\text{s}^2}\right)/(5\ \text{s}) = 34.1\ \text{kW}$$

Thus,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 9.6 + 34.1 = \mathbf{43.7\ kW}$$

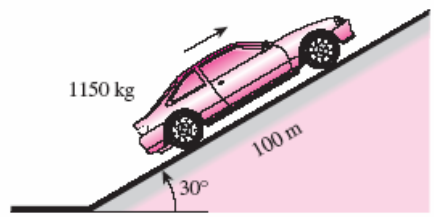
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2-38 A car is to climb a hill in 12 s. The power needed is to be determined for three different cases.

Assumptions Air drag, friction, and rolling resistance are negligible.

Analysis The total power required for each case is the sum of the rates of changes in potential and kinetic energies. That is,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g$$



(*a*) $\dot{W}_a = 0$ since the velocity is constant. Also, the vertical rise is h = (100 m)(sin 30°) = 50 m. Thus,

$$\dot{W}_g = mg(z_2 - z_1)/\Delta t = (1150\ \text{kg})(9.81\ \text{m/s}^2)(50\ \text{m})\left(\frac{1\ \text{kJ}}{1000\ \text{kg}\cdot\text{m}^2/\text{s}^2}\right)/(12\ \text{s}) = 47.0\ \text{kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 0 + 47.0 = \mathbf{47.0\ kW}$

(*b*) The power needed to accelerate is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(1150\ \text{kg})\left[(30\ \text{m/s})^2 - 0\right]\left(\frac{1\ \text{kJ}}{1000\ \text{kg}\cdot\text{m}^2/\text{s}^2}\right)/(12\ \text{s}) = 43.1\ \text{kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 47.0 + 43.1 = \mathbf{90.1\ kW}$

(*c*) The power needed to decelerate is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(1150\ \text{kg})\left[(5\ \text{m/s})^2 - (35\ \text{m/s})^2\right]\left(\frac{1\ \text{kJ}}{1000\ \text{kg}\cdot\text{m}^2/\text{s}^2}\right)/(12\ \text{s}) = -57.5\ \text{kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = -57.5 + 47.1 = \mathbf{-10.5\ kW}$ (breaking power)

2-39 A damaged car is being towed by a truck. The extra power needed is to be determined for three different cases.

Assumptions Air drag, friction, and rolling resistance are negligible.

Analysis The total power required for each case is the sum of the rates of changes in potential and kinetic energies. That is,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g$$



(*a*) Zero.

(*b*) $\dot{W}_a = 0$. Thus,

$$\dot{W}_{\text{total}} = \dot{W}_g = mg(z_2 - z_1)/\Delta t = mg\frac{\Delta z}{\Delta t} = mgV_z = mgV\sin 30^\circ$$

$$= (1200\ \text{kg})(9.81\text{m/s}^2)\left(\frac{50{,}000\ \text{m}}{3600\ \text{s}}\right)\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)(0.5) = \mathbf{81.7\ kW}$$

(*c*) $\dot{W}_g = 0$. Thus,

$$\dot{W}_{\text{total}} = \dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(1200\ \text{kg})\left(\left(\frac{90{,}000\ \text{m}}{3600\ \text{s}}\right)^2 - 0\right)\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)/(12\ \text{s}) = \mathbf{31.3\ kW}$$

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The First Law of Thermodynamics

2-40C No. This is the case for adiabatic systems only.

2-41C Energy can be transferred to or from a control volume as heat, various forms of work, and by mass transport.

2-42C Warmer. Because energy is added to the room air in the form of electrical work.

2-43E The high rolling resistance tires of a car are replaced by low rolling resistance ones. For a specified unit fuel cost, the money saved by switching to low resistance tires is to be determined.

Assumptions **1**The low rolling resistance tires deliver 2 mpg over all velocities. **2** The car is driven 15,000 miles per year.

Analysis The annual amount of fuel consumed by this car on high- and low-rolling resistance tires are

$$\text{Annual Fuel Consumption}_{\text{High}} = \frac{\text{Miles driven per year}}{\text{Miles per gallon}} = \frac{15{,}000 \text{ miles/year}}{35 \text{ miles/gal}} = 428.6 \text{ gal/year}$$

$$\text{Annual Fuel Consumption}_{\text{Low}} = \frac{\text{Miles driven per year}}{\text{Miles per gallon}} = \frac{15{,}000 \text{ miles/year}}{37 \text{ miles/gal}} = 405.4 \text{ gal/year}$$

Then the fuel and money saved per year become

$$\text{Fuel Savings} = \text{Annual Fuel Consumption}_{\text{High}} - \text{Annual Fuel Consumption}_{\text{Low}}$$
$$= 428.6 \text{ gal/year} - 405.4 \text{ gal/year} = 23.2 \text{ gal/year}$$

$$\text{Cost savings} = (\text{Fuel savings})(\text{Unit cost of fuel}) = (23.2 \text{ gal/year})(\$2.20/\text{gal}) = \mathbf{\$51.0/year}$$

Discussion A typical tire lasts about 3 years, and thus the low rolling resistance tires have the potential to save about $150 to the car owner over the life of the tires, which is comparable to the installation cost of the tires.

2-44 The specific energy change of a system which is accelerated is to be determined.

Analysis Since the only property that changes for this system is the velocity, only the kinetic energy will change. The change in the specific energy is

$$\Delta \text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(30 \text{ m/s})^2 - (0 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = \mathbf{0.45 \text{ kJ/kg}}$$

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2-45 The specific energy change of a system which is raised is to be determined.

Analysis Since the only property that changes for this system is the elevation, only the potential energy will change. The change in the specific energy is then

$$\Delta pe = g(z_2 - z_1) = (9.8\ \text{m/s}^2)(100-0)\ \text{m}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{0.98\ kJ/kg}$$

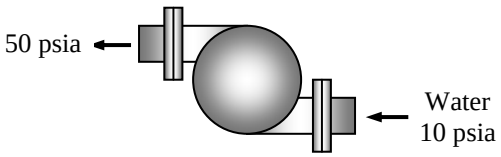
2-46E A water pump increases water pressure. The power input is to be determined.

Analysis The power input is determined from

$$\dot{W} = \dot{\boldsymbol{V}}(P_2 - P_1)$$

$$= (1.2\ \text{ft}^3/\text{s})(50-10)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right)\left(\frac{1\ \text{hp}}{0.7068\ \text{Btu/s}}\right)$$

$$= \mathbf{12.6\ hp}$$



The water temperature at the inlet does not have any significant effect on the required power.

2-47 A classroom is to be air-conditioned using window air-conditioning units. The cooling load is due to people, lights, and heat transfer through the walls and the windows. The number of 5-kW window air conditioning units required is to be determined.

Assumptions There are no heat dissipating equipment (such as computers, TVs, or ranges) in the room.

Analysis The total cooling load of the room is determined from

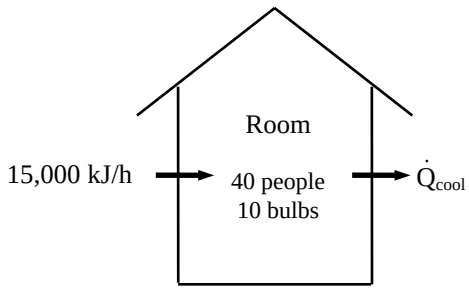
$$\dot{Q}_{\text{cooling}} = \dot{Q}_{\text{lights}} + \dot{Q}_{\text{people}} + \dot{Q}_{\text{heat gain}}$$

where

$$\dot{Q}_{\text{lights}} = 10 \times 100\ \text{W} = 1\ \text{kW}$$

$$\dot{Q}_{\text{people}} = 40 \times 360\ \text{kJ/h} = 4\ \text{kW}$$

$$\dot{Q}_{\text{heat gain}} = 15{,}000\ \text{kJ/h} = 4.17\ \text{kW}$$



Substituting,

$$\dot{Q}_{\text{cooling}} = 1 + 4 + 4.17 = 9.17\ \text{kW}$$

Thus the number of air-conditioning units required is

$$\frac{9.17\ \text{kW}}{5\ \text{kW/unit}} = 1.83 \longrightarrow \mathbf{2\ units}$$

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2-48 The lighting energy consumption of a storage room is to be reduced by installing motion sensors. The amount of energy and money that will be saved as well as the simple payback period are to be determined.

Assumptions The electrical energy consumed by the ballasts is negligible.

Analysis The plant operates 12 hours a day, and thus currently the lights are on for the entire 12 hour period. The motion sensors installed will keep the lights on for 3 hours, and off for the remaining 9 hours every day. This corresponds to a total of 9×365 = 3285 off hours per year. Disregarding the ballast factor, the annual energy and cost savings become

Energy Savings = (Number of lamps)(Lamp wattage)(Reduction of annual operating hours)

= (24 lamps)(60 W/lamp)(3285 hours/year)

= **4730 kWh/year**

Cost Savings = (Energy Savings)(Unit cost of energy)

= (4730 kWh/year)($0.08/kWh)

= **$378/year**



The implementation cost of this measure is the sum of the purchase price of the sensor plus the labor,

Implementation Cost = Material + Labor = $32 + $40 = $72

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$72}{\$378/\text{year}} = \textbf{0.19 year}\ \ (2.3\,\text{months})$$

Therefore, the motion sensor will pay for itself in about 2 months.

2-49 The classrooms and faculty offices of a university campus are not occupied an average of 4 hours a day, but the lights are kept on. The amounts of electricity and money the campus will save per year if the lights are turned off during unoccupied periods are to be determined.

Analysis The total electric power consumed by the lights in the classrooms and faculty offices is

$$\dot{E}_{\text{lighting, classroom}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (200 \times 12 \times 110\ \text{W}) = 264{,}000 = 264\ \text{kW}$$

$$\dot{E}_{\text{lighting, offices}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (400 \times 6 \times 110\ \text{W}) = 264{,}000 = 264\ \text{kW}$$

$$\dot{E}_{\text{lighting, total}} = \dot{E}_{\text{lighting, classroom}} + \dot{E}_{\text{lighting, offices}} = 264 + 264 = 528\ \text{kW}$$

Noting that the campus is open 240 days a year, the total number of unoccupied work hours per year is

Unoccupied hours = (4 hours/day)(240 days/year) = 960 h/yr

Then the amount of electrical energy consumed per year during unoccupied work period and its cost are

Energy savings = $(\dot{E}_{\text{lighting, total}})$(Unoccupied hours) = (528 kW)(960 h/yr) = 506,880 kWh

Cost savings = (Energy savings)(Unit cost of energy) = (506,880 kWh/yr)($0.082/kWh) = **$41,564/yr**

Discussion Note that simple conservation measures can result in significant energy and cost savings.

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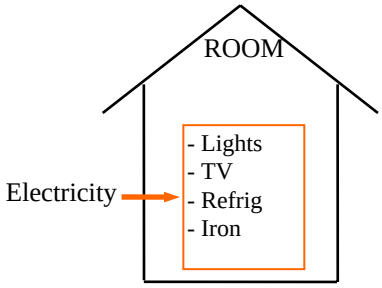
2-50 A room contains a light bulb, a TV set, a refrigerator, and an iron. The rate of increase of the energy content of the room when all of these electric devices are on is to be determined.

Assumptions **1** The room is well sealed, and heat loss from the room is negligible. **2** All the appliances are kept on.

Analysis Taking the room as the system, the rate form of the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}}/dt}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \quad \rightarrow \quad dE_{\text{room}}/dt = \dot{E}_{in}$$

since no energy is leaving the room in any form, and thus $\dot{E}_{out} = 0$. Also,



$$\begin{aligned}\dot{E}_{\text{in}} &= \dot{E}_{\text{lights}} + \dot{E}_{\text{TV}} + \dot{E}_{\text{refrig}} + \dot{E}_{\text{iron}} \\ &= 100 + 110 + 200 + 1000 \text{ W} \\ &= 1410 \text{ W}\end{aligned}$$

Substituting, the rate of increase in the energy content of the room becomes

$$dE_{\text{room}}/dt = \dot{E}_{\text{in}} = \mathbf{1410\ W}$$

Discussion Note that some appliances such as refrigerators and irons operate intermittently, switching on and off as controlled by a thermostat. Therefore, the rate of energy transfer to the room, in general, will be less.

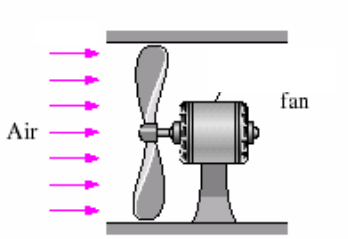
2-51 A fan is to accelerate quiescent air to a specified velocity at a specified flow rate. The minimum power that must be supplied to the fan is to be determined.

Assumptions The fan operates steadily.

Properties The density of air is given to be $\rho = 1.18$ kg/m^3.

Analysis A fan transmits the mechanical energy of the shaft (shaft power) to mechanical energy of air (kinetic energy). For a control volume that encloses the fan, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}}/dt \overset{\nearrow 0 \text{ (steady)}}{} = 0}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$



$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}}\, \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$

where

$$\dot{m}_{\text{air}} = \rho \dot{V} = (1.18 \text{ kg/m}^3)(9 \text{ m}^3/\text{s}) = 10.62 \text{ kg/s}$$

Substituting, the minimum power input required is determined to be

$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2} = (10.62 \text{ kg/s}) \frac{(8 \text{ m/s})^2}{2} \left(\frac{1 \text{ J/kg}}{1 \text{ m}^2/\text{s}^2} \right) = 340 \text{ J/s} = \mathbf{340\ W}$$

Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher because of the losses associated with the conversion of mechanical shaft energy to kinetic energy of air.

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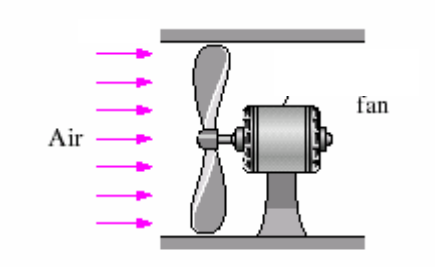
2-52E A fan accelerates air to a specified velocity in a square duct. The minimum electric power that must be supplied to the fan motor is to be determined.

Assumptions **1** The fan operates steadily. **2** There are no conversion losses.

Properties The density of air is given to be $\rho = 0.075$ lbm/ft^3.

Analysis A fan motor converts electrical energy to mechanical shaft energy, and the fan transmits the mechanical energy of the shaft (shaft power) to mechanical energy of air (kinetic energy). For a control volume that encloses the fan-motor unit, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$



$$\dot{W}_{\text{elect, in}} = \dot{m}_{\text{air}}\, \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$

where

$$\dot{m}_{\text{air}} = \rho V A = (0.075\,\text{lbm/ft}^3)(3\times 3\,\text{ft}^2)(22\,\text{ft/s}) = 14.85\,\text{lbm/s}$$

Substituting, the minimum power input required is determined to be

$$\dot{W}_{\text{in}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2} = (14.85\,\text{lbm/s}) \frac{(22\,\text{ft/s})^2}{2} \left(\frac{1\,\text{Btu/lbm}}{25{,}037\,\text{ft}^2/\text{s}^2} \right) = 0.1435\,\text{Btu/s} = \mathbf{151\,W}$$

since 1 Btu = 1.055 kJ and 1 kJ/s = 1000 W.

Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-kinetic energy of air.

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2-53 A gasoline pump raises the pressure to a specified value while consuming electric power at a specified rate. The maximum volume flow rate of gasoline is to be determined.

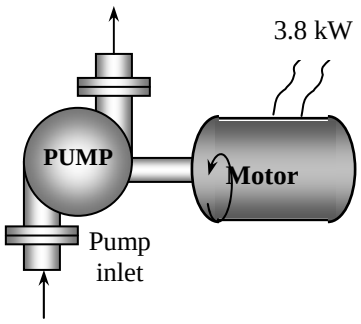
Assumptions **1** The gasoline pump operates steadily. **2** The changes in kinetic and potential energies across the pump are negligible.

Analysis For a control volume that encloses the pump-motor unit, the energy balance can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}(P\nu)_1 = \dot{m}(P\nu)_2 \ \rightarrow \ \dot{W}_{\text{in}} = \dot{m}(P_2 - P_1)\nu = \dot{V}\,\Delta P$$

since $\dot{m} = \dot{V}/\nu$ and the changes in kinetic and potential energies of gasoline are negligible, Solving for volume flow rate and substituting, the maximum flow rate is determined to be



$$\dot{V}_{\max} = \frac{\dot{W}_{\text{in}}}{\Delta \text{P}} = \frac{3.8\ \text{kJ/s}}{7\ \text{kPa}}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{1\ \text{kJ}}\right) = \mathbf{0.543\ m^3/s}$$

Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the volume flow rate will be less because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-flow energy.

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2-54 An inclined escalator is to move a certain number of people upstairs at a constant velocity. The minimum power required to drive this escalator is to be determined.

Assumptions **1** Air drag and friction are negligible. **2** The average mass of each person is 75 kg. **3** The escalator operates steadily, with no acceleration or breaking. **4** The mass of escalator itself is negligible.

Analysis At design conditions, the total mass moved by the escalator at any given time is

Mass = (30 persons)(75 kg/person) = 2250 kg

The vertical component of escalator velocity is

$$V_{\text{vert}} = V \sin 45° = (0.8 \text{ m/s})\sin 45°$$

Under stated assumptions, the power supplied is used to increase the potential energy of people. Taking the people on elevator as the closed system, the energy balance in the rate form can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = dE_{\text{sys}}/dt \cong \frac{\Delta E_{\text{sys}}}{\Delta t}$$

$$\dot{W}_{\text{in}} = \frac{\Delta PE}{\Delta t} = \frac{mg\Delta z}{\Delta t} = mgV_{\text{vert}}$$

That is, under stated assumptions, the power input to the escalator must be equal to the rate of increase of the potential energy of people. Substituting, the required power input becomes

$$\dot{W}_{\text{in}} = mgV_{\text{vert}} = (2250 \text{ kg})(9.81 \text{ m/s}^2)(0.8 \text{ m/s})\sin 45° \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 12.5 \text{ kJ/s} = \mathbf{12.5 \text{ kW}}$$

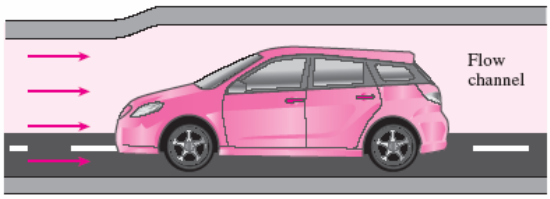
When the escalator velocity is doubled to V = 1.6 m/s, the power needed to drive the escalator becomes

$$\dot{W}_{\text{in}} = mgV_{\text{vert}} = (2250 \text{ kg})(9.81 \text{ m/s}^2)(1.6 \text{ m/s})\sin 45° \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 25.0 \text{ kJ/s} = \mathbf{25.0 \text{ kW}}$$

Discussion Note that the power needed to drive an escalator is proportional to the escalator velocity.

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2-55 An automobile moving at a given velocity is considered. The power required to move the car and the area of the effective flow channel behind the car are to be determined.



Analysis The absolute pressure of the air is

$$P = (700\ \text{mm Hg})\left(\frac{0.1333\ \text{kPa}}{1\ \text{mm Hg}}\right) = 93.31\ \text{kPa}$$

and the specific volume of the air is

$$\nu = \frac{RT}{P} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})}{93.31\ \text{kPa}} = 0.9012\ \text{m}^3/\text{kg}$$

The mass flow rate through the control volume is

$$\dot{m} = \frac{A_1 V_1}{\nu} = \frac{(3\ \text{m}^2)(90/3.6\ \text{m/s})}{0.9012\ \text{m}^3/\text{kg}} = 83.22\ \text{kg/s}$$

The power requirement is

$$\dot{W} = \dot{m}\frac{V_1^2 - V_2^2}{2} = (83.22\ \text{kg/s})\frac{(90/3.6\ \text{m/s})^2 - (82/3.6\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{4.42\ kW}$$

The outlet area is

$$\dot{m} = \frac{A_2 V_2}{\nu} \longrightarrow A_2 = \frac{\dot{m}\nu}{V_2} = \frac{(83.22\ \text{kg/s})(0.9012\ \text{m}^3/\text{kg})}{(82/3.6)\ \text{m/s}} = \mathbf{3.29\ m^2}$$

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Energy Conversion Efficiencies

2-56C *Mechanical efficiency* is defined as the ratio of the mechanical energy output to the mechanical energy input. A mechanical efficiency of 100% for a hydraulic turbine means that the entire mechanical energy of the fluid is converted to mechanical (shaft) work.

2-57C The *combined pump-motor efficiency* of a pump/motor system is defined as the ratio of the increase in the mechanical energy of the fluid to the electrical power consumption of the motor,

$$\eta_{\text{pump-motor}} = \eta_{\text{pump}}\eta_{\text{motor}} = \frac{\dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}}}{\dot{W}_{\text{elect,in}}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{\dot{W}_{\text{pump}}}{\dot{W}_{\text{elect,in}}}$$

The combined pump-motor efficiency cannot be greater than either of the pump or motor efficiency since both pump and motor efficiencies are less than 1, and the product of two numbers that are less than one is less than either of the numbers.

2-58C The turbine efficiency, generator efficiency, and *combined turbine-generator efficiency* are defined as follows:

$$\eta_{\text{turbine}} = \frac{\text{Mechanical energy output}}{\text{Mechanical energy extracted from the fluid}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|}$$

$$\eta_{\text{generator}} = \frac{\text{Electrical power output}}{\text{Mechanical power input}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}$$

$$\eta_{\text{turbine-gen}} = \eta_{\text{turbine}}\eta_{\text{generator}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{E}_{\text{mech,in}} - \dot{E}_{\text{mech,out}}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|}$$

2-59C No, the combined pump-motor efficiency cannot be greater that either of the pump efficiency of the motor efficiency. This is because $\eta_{\text{pump-motor}} = \eta_{\text{pump}}\eta_{\text{motor}}$, and both η_{pump} and η_{motor} are less than one, and a number gets smaller when multiplied by a number smaller than one.

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2-60 A hooded electric open burner and a gas burner are considered. The amount of the electrical energy used directly for cooking and the cost of energy per "utilized" kWh are to be determined.

Analysis The efficiency of the electric heater is given to be 73 percent. Therefore, a burner that consumes 3-kW of electrical energy will supply



$$\eta_{\text{gas}} = 38\%$$

$$\eta_{\text{electric}} = 73\%$$

$$\dot{Q}_{\text{utilized}} = (\text{Energy input}) \times (\text{Efficiency}) = (2.4\,\text{kW})(0.73) = \mathbf{1.75\,kW}$$

of useful energy. The unit cost of utilized energy is inversely proportional to the efficiency, and is determined from

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$0.10/\text{kWh}}{0.73} = \mathbf{\$0.137/kWh}$$

Noting that the efficiency of a gas burner is 38 percent, the energy input to a gas burner that supplies utilized energy at the same rate (1.75 kW) is

$$\dot{Q}_{\text{input, gas}} = \frac{\dot{Q}_{\text{utilized}}}{\text{Efficiency}} = \frac{1.75\,\text{kW}}{0.38} = \mathbf{4.61kW}\ \ (= 15{,}700\,\text{Btu/h})$$

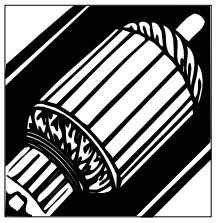
since 1 kW = 3412 Btu/h. Therefore, a gas burner should have a rating of at least 15,700 Btu/h to perform as well as the electric unit. Noting that 1 therm = 29.3 kWh, the unit cost of utilized energy in the case of gas burner is determined the same way to be

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$1.20/(29.3\,\text{kWh})}{0.38} = \mathbf{\$0.108/kWh}$$

2-61 A worn out standard motor is replaced by a high efficiency one. The reduction in the internal heat gain due to the higher efficiency under full load conditions is to be determined.

Assumptions **1** The motor and the equipment driven by the motor are in the same room. **2** The motor operates at full load so that $f_{\text{load}} = 1$.

Analysis The heat generated by a motor is due to its inefficiency, and the difference between the heat generated by two motors that deliver the same shaft power is simply the difference between the electric power drawn by the motors,



$$\dot{W}_{\text{in, electric, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746\,\text{W})/0.91 = 61{,}484\,\text{W}$$

$$\dot{W}_{\text{in, electric, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746\,\text{W})/0.954 = 58{,}648\,\text{W}$$

Then the reduction in heat generation becomes

$$\dot{Q}_{\text{reduction}} = \dot{W}_{\text{in, electric, standard}} - \dot{W}_{\text{in, electric, efficient}} = 61{,}484 - 58{,}648 = \mathbf{2836\,W}$$

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2-62 An electric car is powered by an electric motor mounted in the engine compartment. The rate of heat supply by the motor to the engine compartment at full load conditions is to be determined.

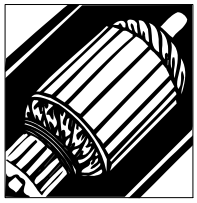
Assumptions The motor operates at full load so that the load factor is 1.

Analysis The heat generated by a motor is due to its inefficiency, and is equal to the difference between the electrical energy it consumes and the shaft power it delivers,

$$\dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (90\text{ hp})/0.91 = 98.90\text{ hp}$$

$$\dot{Q}_{\text{generation}} = \dot{W}_{\text{in, electric}} - \dot{W}_{\text{shaft out}} = 98.90 - 90 = 8.90\text{ hp} = \mathbf{6.64\ kW}$$

since 1 hp = 0.746 kW.



Discussion Note that the electrical energy not converted to mechanical power is converted to heat.

2-63 A worn out standard motor is to be replaced by a high efficiency one. The amount of electrical energy and money savings as a result of installing the high efficiency motor instead of the standard one as well as the simple payback period are to be determined.

Assumptions The load factor of the motor remains constant at 0.75.

Analysis The electric power drawn by each motor and their difference can be expressed as

$$\dot{W}_{\text{electric in, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{standard}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{standard}}$$

$$\dot{W}_{\text{electric in, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{efficient}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{efficient}}$$

$$\text{Power savings} = \dot{W}_{\text{electric in, standard}} - \dot{W}_{\text{electric in, efficient}}$$

$$= (\text{Power rating})(\text{Load factor})[1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}}]$$

where η_{standard} is the efficiency of the standard motor, and $\eta_{\text{efficient}}$ is the efficiency of the comparable high efficiency motor. Then the annual energy and cost savings associated with the installation of the high efficiency motor are determined to be

Energy Savings = (Power savings)(Operating Hours)

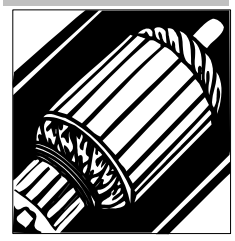
= (Power Rating)(Operating Hours)(Load Factor)($1/\eta_{\text{standard}}$- $1/\eta_{\text{efficient}}$)

= (75 hp)(0.746 kW/hp)(4,368 hours/year)(0.75)(1/0.91 - 1/0.954)

= **9,290 kWh/year**

$\eta_{\text{old}} = 91.0\%$

$\eta_{\text{new}} = 95.4\%$



Cost Savings = (Energy savings)(Unit cost of energy)

= (9,290 kWh/year)($0.08/kWh)

= **$743/year**

The implementation cost of this measure consists of the excess cost the high efficiency motor over the standard one. That is,

Implementation Cost = Cost differential = $5,520 - $5,449 = $71

This gives a simple payback period of

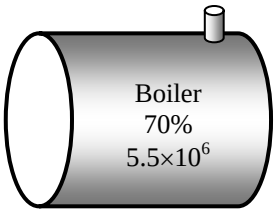
$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$71}{\$743/\text{year}} = \mathbf{0.096\ year}\ (\text{or } 1.1\text{ months})$$

Therefore, the high-efficiency motor will pay for its cost differential in about one month.

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2-64E The combustion efficiency of a furnace is raised from 0.7 to 0.8 by tuning it up. The annual energy and cost savings as a result of tuning up the boiler are to be determined.

Assumptions The boiler operates at full load while operating.



Analysis The heat output of boiler is related to the fuel energy input to the boiler by

Boiler output = (Boiler input)(Combustion efficiency)

or $\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} \eta_{\text{furnace}}$

The current rate of heat input to the boiler is given to be $\dot{Q}_{\text{in, current}} = 5.5 \times 10^6$ Btu/h .

Then the rate of useful heat output of the boiler becomes

$$\dot{Q}_{\text{out}} = (\dot{Q}_{\text{in}} \eta_{\text{furnace}})_{\text{current}} = (5.5 \times 10^6 \text{ Btu/h})(0.7) = 3.85 \times 10^6 \text{ Btu/h}$$

The boiler must supply useful heat at the same rate after the tune up. Therefore, the rate of heat input to the boiler after the tune up and the rate of energy savings become

$$\dot{Q}_{\text{in, new}} = \dot{Q}_{\text{out}} / \eta_{\text{furnace, new}} = (3.85 \times 10^6 \text{ Btu/h})/0.8 = 4.81 \times 10^6 \text{ Btu/h}$$

$$\dot{Q}_{\text{in, saved}} = \dot{Q}_{\text{in, current}} - \dot{Q}_{\text{in, new}} = 5.5 \times 10^6 - 4.81 \times 10^6 = 0.69 \times 10^6 \text{ Btu/h}$$

Then the annual energy and cost savings associated with tuning up the boiler become

Energy Savings $= \dot{Q}_{\text{in, saved}}$ (Operation hours)

= (0.69×10^6 Btu/h)(4200 h/year) = **2.89×10^9 Btu/yr**

Cost Savings = (Energy Savings)(Unit cost of energy)

= (2.89×10^9 Btu/yr)($4.35/10^6 Btu) = **$12,600/year**

Discussion Notice that tuning up the boiler will save $12,600 a year, which is a significant amount. The implementation cost of this measure is negligible if the adjustment can be made by in-house personnel. Otherwise it is worthwhile to have an authorized representative of the boiler manufacturer to service the boiler twice a year.

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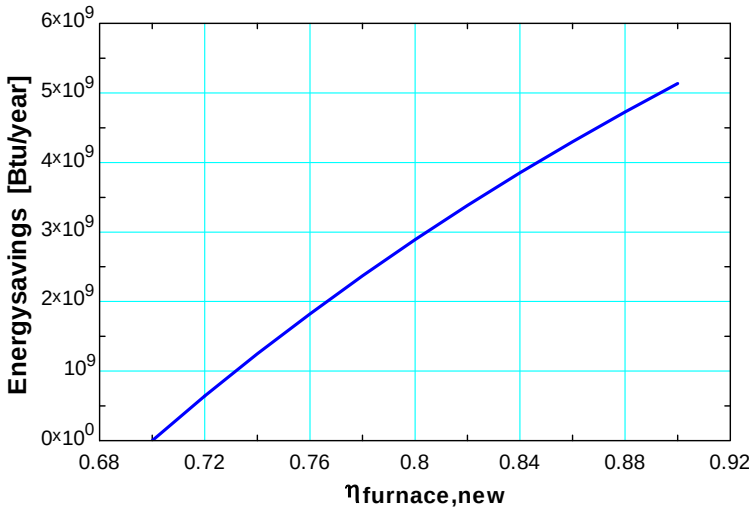
2-65E Problem 2-64E is reconsidered. The effects of the unit cost of energy and combustion efficiency on the annual energy used and the cost savings as the efficiency varies from 0.7 to 0.9 and the unit cost varies from $4 to $6 per million Btu are the investigated. The annual energy saved and the cost savings are to be plotted against the efficiency for unit costs of $4, $5, and $6 per million Btu.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
Q_dot_in_current=5.5E6 [Btu/h]
eta_furnace_current=0.7
eta_furnace_new=0.8
Hours=4200 [h/year]
UnitCost=4.35E-6 [$/Btu]

"Analysis"
Q_dot_out=Q_dot_in_current*eta_furnace_current
Q_dot_in_new=Q_dot_out/eta_furnace_new
Q_dot_in_saved=Q_dot_in_current-Q_dot_in_new
Energysavings=Q_dot_in_saved*Hours
CostSavings=EnergySavings*UnitCost
```

$\eta_{furnace,new}$	EnergySavings [Btu/year]	CostSavings [$/year]
0.7	0.00E+00	0
0.72	6.42E+08	3208
0.74	1.25E+09	6243
0.76	1.82E+09	9118
0.78	2.37E+09	11846
0.8	2.89E+09	14437
0.82	3.38E+09	16902
0.84	3.85E+09	19250
0.86	4.30E+09	21488
0.88	4.73E+09	23625
0.9	5.13E+09	25667



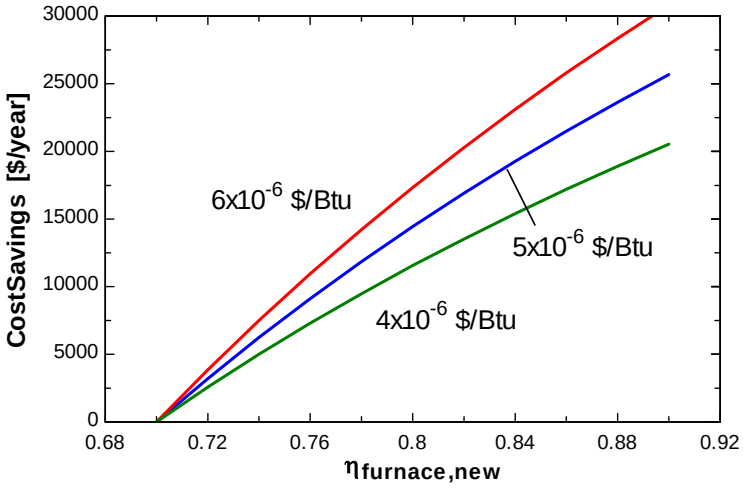


Table values are for UnitCost = 5E-5 [$/Btu]

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2-66 Several people are working out in an exercise room. The rate of heat gain from people and the equipment is to be determined.

Assumptions The average rate of heat dissipated by people in an exercise room is 525 W.

Analysis The 8 weight lifting machines do not have any motors, and thus they do not contribute to the internal heat gain directly. The usage factors of the motors of the treadmills are taken to be unity since they are used constantly during peak periods. Noting that 1 hp = 746 W, the total heat generated by the motors is

$$\dot{Q}_{\text{motors}} = (\text{No. of motors}) \times \dot{W}_{\text{motor}} \times f_{\text{load}} \times f_{\text{usage}} / \eta_{\text{motor}}$$
$$= 4 \times (2.5 \times 746 \text{ W}) \times 0.70 \times 1.0/0.77 = 6782 \text{ W}$$



The heat gain from 14 people is

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 14 \times (525 \text{ W}) = 7350 \text{ W}$$

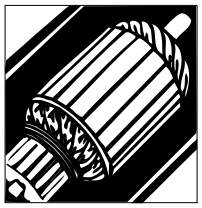
Then the total rate of heat gain of the exercise room during peak period becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{motors}} + \dot{Q}_{\text{people}} = 6782 + 7350 = \mathbf{14{,}132\ W}$$

2-67 A room is cooled by circulating chilled water through a heat exchanger, and the air is circulated through the heat exchanger by a fan. The contribution of the fan-motor assembly to the cooling load of the room is to be determined.

Assumptions The fan motor operates at full load so that $f_{\text{load}} = 1$.

Analysis The entire electrical energy consumed by the motor, including the shaft power delivered to the fan, is eventually dissipated as heat. Therefore, the contribution of the fan-motor assembly to the cooling load of the room is equal to the electrical energy it consumes,



$$\dot{Q}_{\text{internal generation}} = \dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}}$$
$$= (0.25 \text{ hp})/0.54 = 0.463 \text{ hp} = \mathbf{345\ W}$$

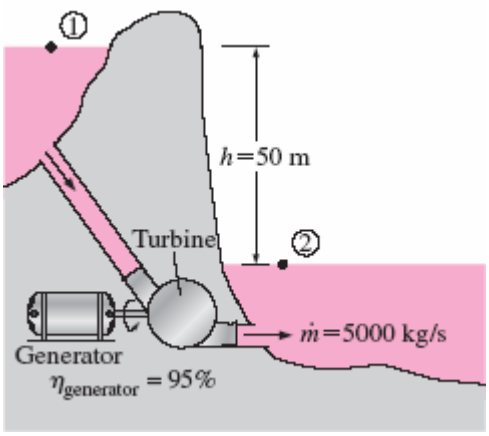
since 1 hp = 746 W.

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2-68 A hydraulic turbine-generator is to generate electricity from the water of a lake. The overall efficiency, the turbine efficiency, and the shaft power are to be determined.

Assumptions **1** The elevation of the lake and that of the discharge site remains constant. **2** Irreversible losses in the pipes are negligible.

Properties The density of water can be taken to be $\rho = 1000$ kg/m^3. The gravitational acceleration is $g = 9.81$ m/s^2.



Analysis (*a*) We take the bottom of the lake as the reference level for convenience. Then kinetic and potential energies of water are zero, and the mechanical energy of water consists of pressure energy only which is

$$e_{\text{mech,in}} - e_{\text{mech,out}} = \frac{P}{\rho} = gh$$
$$= (9.81\,\text{m/s}^2)(50\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right)$$
$$= 0.491\,\text{kJ/kg}$$

Then the rate at which mechanical energy of fluid supplied to the turbine and the overall efficiency become

$$|\Delta\dot{E}_{\text{mech,fluid}}| = \dot{m}(e_{\text{mech,in}} - e_{\text{mech,in}}) = (5000\,\text{kg/s})(0.491\,\text{kJ/kg}) = 2455\,\text{kW}$$

$$\eta_{\text{overall}} = \eta_{\text{turbine-gen}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|} = \frac{1862\,\text{kW}}{2455\,\text{kW}} = \mathbf{0.760}$$

(*b*) Knowing the overall and generator efficiencies, the mechanical efficiency of the turbine is determined from

$$\eta_{\text{turbine-gen}} = \eta_{\text{turbine}}\eta_{\text{generator}} \quad \rightarrow \quad \eta_{\text{turbine}} = \frac{\eta_{\text{turbine-gen}}}{\eta_{\text{generator}}} = \frac{0.76}{0.95} = \mathbf{0.800}$$

(*c*) The shaft power output is determined from the definition of mechanical efficiency,

$$\dot{W}_{\text{shaft,out}} = \eta_{\text{turbine}}|\Delta\dot{E}_{\text{mech,fluid}}| = (0.800)(2455\,\text{kW}) = 1964\,\text{kW} \approx \mathbf{1960\,kW}$$

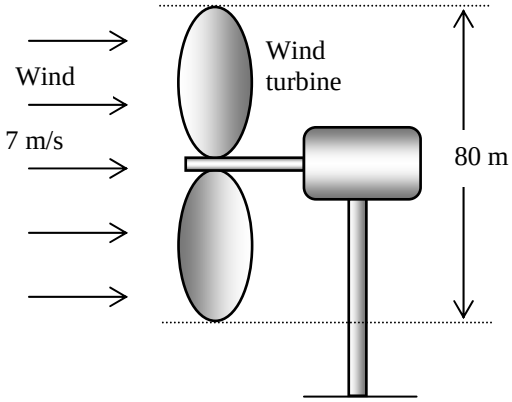
Therefore, the lake supplies 2455 kW of mechanical energy to the turbine, which converts 1964 kW of it to shaft work that drives the generator, which generates 1862 kW of electric power.

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2-69 Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass, the power generation potential, and the actual electric power generation are to be determined.

Assumptions **1** The wind is blowing steadily at a constant uniform velocity. **2** The efficiency of the wind turbine is independent of the wind speed.

Properties The density of air is given to be $\rho = 1.25$ kg/m^3.



Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(7\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.0245\text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V\frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(7\text{ m/s})\frac{\pi(80\text{ m})^2}{4} = 43{,}982\text{ kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (43{,}982\text{ kg/s})(0.0245\text{ kJ/kg}) = \mathbf{1078\text{ kW}}$$

The actual electric power generation is determined by multiplying the power generation potential by the efficiency,

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}}\dot{W}_{\max} = (0.30)(1078\text{ kW}) = \mathbf{323\text{ kW}}$$

Therefore, 323 kW of actual power can be generated by this wind turbine at the stated conditions.

Discussion The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.

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2-70 Problem 2-69 is reconsidered. The effect of wind velocity and the blade span diameter on wind power generation as the velocity varies from 5 m/s to 20 m/s in increments of 5 m/s, and the diameter varies from 20 m to 120 m in increments of 20 m is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

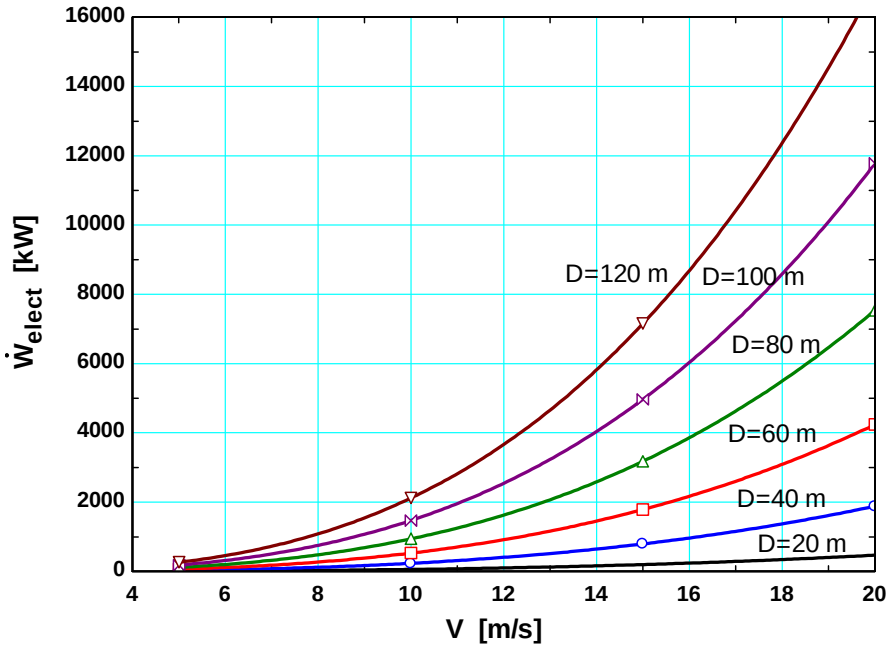
```
"Given"
V=7 [m/s]
D=80 [m]
eta_overall=0.30
rho=1.25 [kg/m^3]

"Analysis"
g=9.81 [m/s^2]
A=pi*D^2/4
m_dot=rho*A*V
W_dot_max=m_dot*V^2/2*Convert(m^2/s^2, kJ/kg)
W_dot_elect=eta_overall*W_dot_max
```

Parametric Table

Table 1

1..24	D [m]	V [m/s]	$\dot{m}$ [kg/s]	$\dot{W}_{elect}$ [kW]
Run 1	20	5	1963	7.363
Run 2	20	10	3927	58.9
Run 3	20	15	5890	198.8
Run 4	20	20	7854	471.2
Run 5	40	5	7854	29.45
Run 6	40	10	15708	235.6
Run 7	40	15	23562	795.2
Run 8	40	20	31416	1885
Run 9	60	5	17671	66.27
Run 10	60	10	35343	530.1
Run 11	60	15	53014	1789
Run 12	60	20	70686	4241
Run 13	80	5	31416	117.8
Run 14	80	10	62832	942.5
Run 15	80	15	94248	3181
Run 16	80	20	125664	7540
Run 17	100	5	49087	184.1
Run 18	100	10	98175	1473
Run 19	100	15	147262	4970
Run 20	100	20	196350	11781
Run 21	120	5	70686	265.1
Run 22	120	10	141372	2121
Run 23	120	15	212058	7157
Run 24	120	20	282743	16965

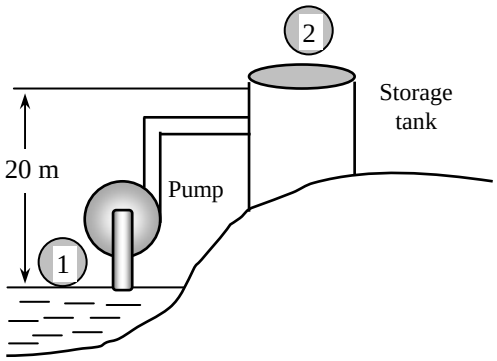


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2-71 Water is pumped from a lake to a storage tank at a specified rate. The overall efficiency of the pump-motor unit and the pressure difference between the inlet and the exit of the pump are to be determined.

Assumptions **1** The elevations of the tank and the lake remain constant. **2** Frictional losses in the pipes are negligible. **3** The changes in kinetic energy are negligible. **4** The elevation difference across the pump is negligible.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.



Analysis (*a*) We take the free surface of the lake to be point 1 and the free surfaces of the storage tank to be point 2. We also take the lake surface as the reference level ($z_1 = 0$), and thus the potential energy at points 1 and 2 are $pe_1 = 0$ and $pe_2 = gz_2$. The flow energy at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$). Further, the kinetic energy at both points is zero ($ke_1 = ke_2 = 0$) since the water at both locations is essentially stationary. The mass flow rate of water and its potential energy at point 2 are

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1000 \text{ kg/m}^3)(0.070 \text{ m}^3/\text{s}) = 70 \text{ kg/s}$$

$$pe_2 = gz_2 = (9.81 \text{ m/s}^2)(20 \text{ m})\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 0.196 \text{ kJ/kg}$$

Then the rate of increase of the mechanical energy of water becomes

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}(pe_2 - 0) = \dot{m} pe_2 = (70 \text{ kg/s})(0.196 \text{ kJ/kg}) = 13.7 \text{ kW}$$

The overall efficiency of the combined pump-motor unit is determined from its definition,

$$\eta_{\text{pump-motor}} = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{13.7 \text{ kW}}{20.4 \text{ kW}} = 0.672 \quad \text{or} \quad \mathbf{67.2\%}$$

(*b*) Now we consider the pump. The change in the mechanical energy of water as it flows through the pump consists of the change in the flow energy only since the elevation difference across the pump and the change in the kinetic energy are negligible. Also, this change must be equal to the useful mechanical energy supplied by the pump, which is 13.7 kW:

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}\frac{P_2 - P_1}{\rho} = \dot{\boldsymbol{V}} \Delta P$$

Solving for ΔP and substituting,

$$\Delta P = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{\boldsymbol{V}}} = \frac{13.7 \text{ kJ/s}}{0.070 \text{ m}^3/\text{s}}\left(\frac{1 \text{ kPa} \cdot \text{m}^3}{1 \text{ kJ}}\right) = \mathbf{196 \text{ kPa}}$$

Therefore, the pump must boost the pressure of water by 196 kPa in order to raise its elevation by 20 m.

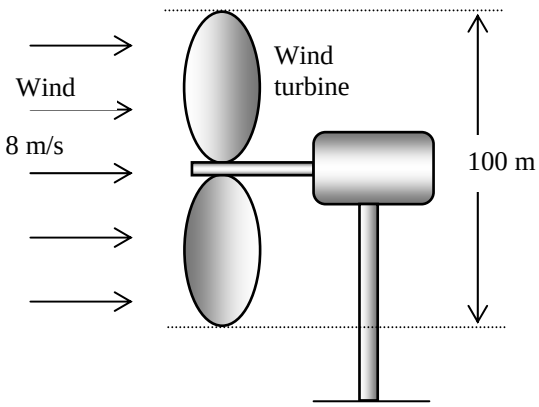
Discussion Note that only two-thirds of the electric energy consumed by the pump-motor is converted to the mechanical energy of water; the remaining one-third is wasted because of the inefficiencies of the pump and the motor.

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2-72 A large wind turbine is installed at a location where the wind is blowing steadily at a certain velocity. The electric power generation, the daily electricity production, and the monetary value of this electricity are to be determined.

Assumptions **1** The wind is blowing steadily at a constant uniform velocity. **2** The efficiency of the wind turbine is independent of the wind speed.

Properties The density of air is given to be $\rho = 1.25$ kg/m^3.



Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(8\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.032\text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V\frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(8\text{ m/s})\frac{\pi(100\text{ m})^2}{4} = 78{,}540\text{ kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (78{,}540\text{ kg/s})(0.032\text{ kJ/kg}) = 2513\text{ kW}$$

The actual electric power generation is determined from

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}}\dot{W}_{\max} = (0.32)(2513\text{ kW}) = \mathbf{804.2\text{ kW}}$$

Then the amount of electricity generated per day and its monetary value become

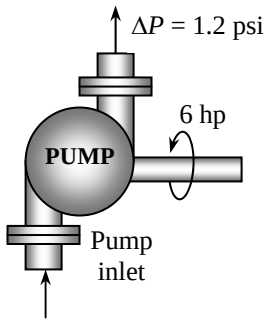
Amount of electricity = (Wind power)(Operating hours)=(804.2 kW)(24 h) =**19,300 kWh**

Revenues = (Amount of electricity)(Unit price) = (19,300 kWh)($0.06/kWh) = **$1158** (per day)

Discussion Note that a single wind turbine can generate several thousand dollars worth of electricity every day at a reasonable cost, which explains the overwhelming popularity of wind turbines in recent years.

2-73E A water pump raises the pressure of water by a specified amount at a specified flow rate while consuming a known amount of electric power. The mechanical efficiency of the pump is to be determined.

Assumptions **1** The pump operates steadily. **2** The changes in velocity and elevation across the pump are negligible. **3** Water is incompressible.



Analysis To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}[(P\nu)_2 - (P\nu)_1] = \dot{m}(P_2 - P_1)\nu$$

$$= \dot{\mathbf{V}}(P_2 - P_1) = (15\text{ ft}^3/\text{s})(1.2\text{ psi})\left(\frac{1\text{ Btu}}{5.404\text{ psi}\cdot\text{ft}^3}\right) = 3.33\text{ Btu/s} = 4.71\text{ hp}$$

since 1 hp = 0.7068 Btu/s, $\dot{m} = \rho\dot{\mathbf{V}} = \dot{\mathbf{V}}/\nu$, and there is no change in kinetic and potential energies of the fluid. Then the mechanical efficiency of the pump becomes

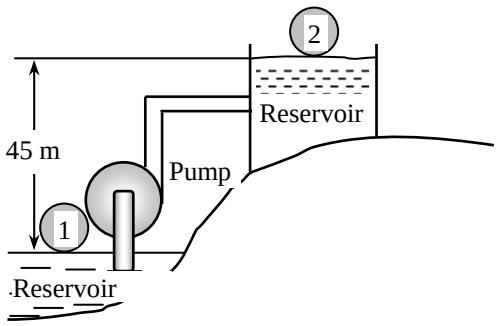
$$\eta_{\text{pump}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump, shaft}}} = \frac{4.71\text{ hp}}{6\text{ hp}} = 0.786 \quad \text{or} \quad \mathbf{78.6\%}$$

Discussion The overall efficiency of this pump will be lower than 83.8% because of the inefficiency of the electric motor that drives the pump.

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2-74 Water is pumped from a lower reservoir to a higher reservoir at a specified rate. For a specified shaft power input, the power that is converted to thermal energy is to be determined.

Assumptions **1** The pump operates steadily. **2** The elevations of the reservoirs remain constant. **3** The changes in kinetic energy are negligible.



Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy, which is *gz* per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,

$$\Delta \dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho \dot{V} g\Delta z$$

$$= (1000\ \text{kg/m}^3)(0.03\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(45\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = 13.2\ \text{kW}$$

Then the mechanical power lost because of frictional effects becomes

$$\dot{W}_{\text{frict}} = \dot{W}_{\text{pump, in}} - \Delta \dot{E}_{\text{mech}} = 20 - 13.2\ \text{kW} = \mathbf{6.8\ kW}$$

Discussion The 6.8 kW of power is used to overcome the friction in the piping system. The effect of frictional losses in a pump is always to convert mechanical energy to an equivalent amount of thermal energy, which results in a slight rise in fluid temperature. Note that this pumping process could be accomplished by a 13.2 kW pump (rather than 20 kW) if there were no frictional losses in the system. In this ideal case, the pump would function as a turbine when the water is allowed to flow from the upper reservoir to the lower reservoir and extract 13.2 kW of power from the water.

2-75 The mass flow rate of water through the hydraulic turbines of a dam is to be determined.

Analysis The mass flow rate is determined from

$$\dot{W} = \dot{m}g(z_2 - z_1) \longrightarrow \dot{m} = \frac{\dot{W}}{g(z_2 - z_1)} = \frac{100{,}000\ \text{kJ/s}}{(9.8\ \text{m/s}^2)(206 - 0)\ \text{m}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)} = \mathbf{49{,}500\ kg/s}$$

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2-76 A pump is pumping oil at a specified rate. The pressure rise of oil in the pump is measured, and the motor efficiency is specified. The mechanical efficiency of the pump is to be determined.

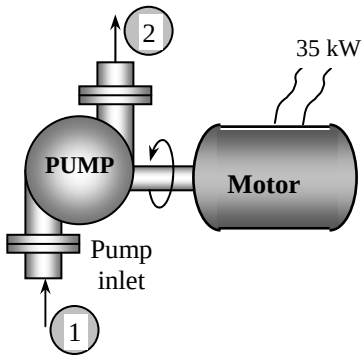
Assumptions **1** The flow is steady and incompressible. **2** The elevation difference across the pump is negligible.

Properties The density of oil is given to be $\rho = 860$ kg/m^3.

Analysis Then the total mechanical energy of a fluid is the sum of the potential, flow, and kinetic energies, and is expressed per unit mass as $e_{\text{mech}} = gh + Pv + V^2/2$. To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}\left((Pv)_2 + \frac{V_2^2}{2} - (Pv)_1 - \frac{V_1^2}{2} \right) = \dot{\boldsymbol{V}}\left((P_2 - P_1) + \rho \frac{V_2^2 - V_1^2}{2} \right)$$

since $\dot{m} = \rho \dot{\boldsymbol{V}} = \dot{\boldsymbol{V}} / \boldsymbol{v}$, and there is no change in the potential energy of the fluid. Also,



$$V_1 = \frac{\dot{\boldsymbol{V}}}{A_1} = \frac{\dot{\boldsymbol{V}}}{\pi D_1^2/4} = \frac{0.1\,\text{m}^3/\text{s}}{\pi(0.08\,\text{m})^2/4} = 19.9\,\text{m/s}$$

$$V_2 = \frac{\dot{\boldsymbol{V}}}{A_2} = \frac{\dot{\boldsymbol{V}}}{\pi D_2^2/4} = \frac{0.1\,\text{m}^3/\text{s}}{\pi(0.12\,\text{m})^2/4} = 8.84\,\text{m/s}$$

Substituting, the useful pumping power is determined to be

$$\begin{aligned} \dot{W}_{\text{pump,u}} &= \Delta \dot{E}_{\text{mech,fluid}} \\ &= (0.1\,\text{m}^3/\text{s})\left(400\,\text{kN/m}^2 + (860\,\text{kg/m}^3)\frac{(8.84\,\text{m/s})^2 - (19.9\,\text{m/s})^2}{2}\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)\right)\left(\frac{1\,\text{kW}}{1\,\text{kN}\cdot\text{m/s}}\right) \\ &= 26.3\,\text{kW} \end{aligned}$$

Then the shaft power and the mechanical efficiency of the pump become

$$\dot{W}_{\text{pump,shaft}} = \eta_{\text{motor}} \dot{W}_{\text{electric}} = (0.90)(35\,\text{kW}) = 31.5\,\text{kW}$$

$$\eta_{\text{pump}} = \frac{\dot{W}_{\text{pump, u}}}{\dot{W}_{\text{pump, shaft}}} = \frac{26.3\,\text{kW}}{31.5\,\text{kW}} = 0.836 = \mathbf{83.6\%}$$

Discussion The overall efficiency of this pump/motor unit is the product of the mechanical and motor efficiencies, which is $0.9 \times 0.836 = 0.75$.

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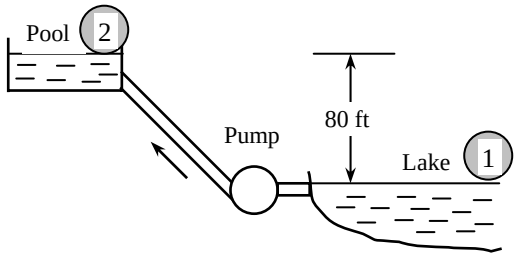
2-77E Water is pumped from a lake to a nearby pool by a pump with specified power and efficiency. The mechanical power used to overcome frictional effects is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the lake and the free surface of the pool is constant. **3** The average flow velocity is constant since pipe diameter is constant.

Properties We take the density of water to be $\rho = 62.4$ lbm/ft^3.

Analysis The useful mechanical pumping power delivered to water is

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}} \dot{W}_{\text{pump}} = (0.80)(20\text{ hp}) = 16\text{ hp}$$



The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy, which is *gz* per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,

$$\Delta \dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho \dot{V} g\Delta z$$

Substituting, the rate of change of mechanical energy of water becomes

$$\Delta \dot{E}_{\text{mech}} = (62.4\text{ lbm/ft}^3)(1.5\text{ ft}^3\text{/s})(32.2\text{ ft/s}^2)(80\text{ ft})\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\text{ hp}}{550\text{ lbf}\cdot\text{ft/s}}\right) = 13.63\text{ hp}$$

Then the mechanical power lost in piping because of frictional effects becomes

$$\dot{W}_{\text{frict}} = \dot{W}_{\text{pump, u}} - \Delta \dot{E}_{\text{mech}} = 16 - 13.63\text{ hp} = \mathbf{2.37\ hp}$$

Discussion Note that the pump must supply to the water an additional useful mechanical power of 2.37 hp to overcome the frictional losses in pipes.

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2-78 A wind turbine produces 180 kW of power. The average velocity of the air and the conversion efficiency of the turbine are to be determined.

Assumptions The wind turbine operates steadily.

Properties The density of air is given to be 1.31 kg/m^3.

Analysis (*a*) The blade diameter and the blade span area are

$$D = \frac{V_{\text{tip}}}{\pi \dot{n}} = \frac{(250\ \text{km/h})\left(\frac{1\ \text{m/s}}{3.6\ \text{km/h}}\right)}{\pi(15\ \text{L/min})\left(\frac{1\ \text{min}}{60\ \text{s}}\right)} = 88.42\ \text{m}$$

$$A = \frac{\pi D^2}{4} = \frac{\pi(88.42\ \text{m})^2}{4} = 6140\ \text{m}^2$$

Then the average velocity of air through the wind turbine becomes

$$V = \frac{\dot{m}}{\rho A} = \frac{42{,}000\ \text{kg/s}}{(1.31\ \text{kg/m}^3)(6140\ \text{m}^2)} = \mathbf{5.23\ m/s}$$

(*b*) The kinetic energy of the air flowing through the turbine is

$$\dot{\text{KE}} = \frac{1}{2}\dot{m}\text{V}^2 = \frac{1}{2}(42{,}000\ \text{kg/s})(5.23\ \text{m/s})^2 = 574.3\ \text{kW}$$

Then the conversion efficiency of the turbine becomes

$$\eta = \frac{\dot{W}}{\dot{\text{KE}}} = \frac{180\ \text{kW}}{574.3\ \text{kW}} = \mathbf{0.313} = \mathbf{31.3\%}$$

Discussion Note that about one-third of the kinetic energy of the wind is converted to power by the wind turbine, which is typical of actual turbines.

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Energy and Environment

2-79C Energy conversion pollutes the soil, the water, and the air, and the environmental pollution is a serious threat to vegetation, wild life, and human health. The emissions emitted during the combustion of fossil fuels are responsible for smog, acid rain, and global warming and climate change. The primary chemicals that pollute the air are hydrocarbons (HC, also referred to as volatile organic compounds, VOC), nitrogen oxides (NOx), and carbon monoxide (CO). The primary source of these pollutants is the motor vehicles.

2-80C Smog is the brown haze that builds up in a large stagnant air mass, and hangs over populated areas on calm hot summer days. Smog is made up mostly of ground-level ozone (O_3), but it also contains numerous other chemicals, including carbon monoxide (CO), particulate matter such as soot and dust, volatile organic compounds (VOC) such as benzene, butane, and other hydrocarbons. Ground-level ozone is formed when hydrocarbons and nitrogen oxides react in the presence of sunlight in hot calm days. Ozone irritates eyes and damage the air sacs in the lungs where oxygen and carbon dioxide are exchanged, causing eventual hardening of this soft and spongy tissue. It also causes shortness of breath, wheezing, fatigue, headaches, nausea, and aggravate respiratory problems such as asthma.

2-81C Fossil fuels include small amounts of sulfur. The sulfur in the fuel reacts with oxygen to form sulfur dioxide (SO_2), which is an air pollutant. The sulfur oxides and nitric oxides react with water vapor and other chemicals high in the atmosphere in the presence of sunlight to form sulfuric and nitric acids. The acids formed usually dissolve in the suspended water droplets in clouds or fog. These acid-laden droplets are washed from the air on to the soil by rain or snow. This is known as *acid rain*. It is called "rain" since it comes down with rain droplets.

As a result of acid rain, many lakes and rivers in industrial areas have become too acidic for fish to grow. Forests in those areas also experience a slow death due to absorbing the acids through their leaves, needles, and roots. Even marble structures deteriorate due to acid rain.

2-82C Carbon monoxide, which is a colorless, odorless, poisonous gas that deprives the body's organs from getting enough oxygen by binding with the red blood cells that would otherwise carry oxygen. At low levels, carbon monoxide decreases the amount of oxygen supplied to the brain and other organs and muscles, slows body reactions and reflexes, and impairs judgment. It poses a serious threat to people with heart disease because of the fragile condition of the circulatory system and to fetuses because of the oxygen needs of the developing brain. At high levels, it can be fatal, as evidenced by numerous deaths caused by cars that are warmed up in closed garages or by exhaust gases leaking into the cars.

2-83C Carbon dioxide (CO_2), water vapor, and trace amounts of some other gases such as methane and nitrogen oxides act like a blanket and keep the earth warm at night by blocking the heat radiated from the earth. This is known as the *greenhouse effect*. The greenhouse effect makes life on earth possible by keeping the earth warm. But excessive amounts of these gases disturb the delicate balance by trapping too much energy, which causes the average temperature of the earth to rise and the climate at some localities to change. These undesirable consequences of the greenhouse effect are referred to as *global warming* or *global climate change*. The greenhouse effect can be reduced by reducing the net production of CO_2 by consuming less energy (for example, by buying energy efficient cars and appliances) and planting trees.

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2-84E A person trades in his Ford Taurus for a Ford Explorer. The extra amount of CO_2 emitted by the Explorer within 5 years is to be determined.

Assumptions The Explorer is assumed to use 940 gallons of gasoline a year compared to 715 gallons for Taurus.

Analysis The extra amount of gasoline the Explorer will use within 5 years is

Extra Gasoline = (Extra per year)(No. of years)
= (940 – 715 gal/yr)(5 yr)
= 1125 gal

Extra CO_2 produced = (Extra gallons of gasoline used)(CO_2 emission per gallon)
= (1125 gal)(19.7 lbm/gal)
= **22,163 lbm CO_2**

Discussion Note that the car we choose to drive has a significant effect on the amount of greenhouse gases produced.

2-85 A power plant that burns natural gas produces 0.59 kg of carbon dioxide (CO_2) per kWh. The amount of CO_2 production that is due to the refrigerators in a city is to be determined.

Assumptions The city uses electricity produced by a natural gas power plant.

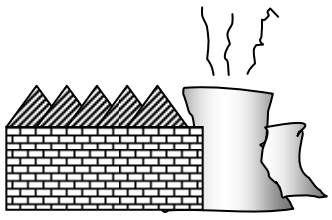
Properties 0.59 kg of CO_2 is produced per kWh of electricity generated (given).

Analysis Noting that there are 300,000 households in the city and each household consumes 700 kWh of electricity for refrigeration, the total amount of CO_2 produced is

$$\begin{aligned}\text{Amount of } CO_2 \text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of } CO_2 \text{ per kWh}) \\ &= (300{,}000 \text{ household})(700 \text{ kWh/year household})(0.59 \text{ kg/kWh}) \\ &= 1.23\times 10^8 \ CO_2 \text{ kg/year} \\ &= \mathbf{123{,}000\ CO_2\ ton/year}\end{aligned}$$

Therefore, the refrigerators in this city are responsible for the production of 123,000 tons of CO_2.

2-86 A power plant that burns coal, produces 1.1 kg of carbon dioxide (CO_2) per kWh. The amount of CO_2 production that is due to the refrigerators in a city is to be determined.



Assumptions The city uses electricity produced by a coal power plant.

Properties 1.1 kg of CO_2 is produced per kWh of electricity generated (given).

Analysis Noting that there are 300,000 households in the city and each household consumes 700 kWh of electricity for refrigeration, the total amount of CO_2 produced is

$$\begin{aligned}\text{Amount of } CO_2 \text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of } CO_2 \text{ per kWh}) \\ &= (300{,}000 \text{ household})(700 \text{ kWh/household})(1.1 \text{ kg/kWh}) \\ &= 2.31\times 10^8 \ CO_2 \text{ kg/year} \\ &= \mathbf{231{,}000\ CO_2\ ton/year}\end{aligned}$$

Therefore, the refrigerators in this city are responsible for the production of 231,000 tons of CO_2.

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2-87E A household uses fuel oil for heating, and electricity for other energy needs. Now the household reduces its energy use by 20%. The reduction in the CO_2 production this household is responsible for is to be determined.

Properties The amount of CO_2 produced is 1.54 lbm per kWh and 26.4 lbm per gallon of fuel oil (given).

Analysis Noting that this household consumes 11,000 kWh of electricity and 1500 gallons of fuel oil per year, the amount of CO_2 production this household is responsible for is

$$\begin{aligned}\text{Amount of } CO_2 \text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of } CO_2 \text{ per kWh}) \\ &\quad + (\text{Amount of fuel oil consumed})(\text{Amount of } CO_2 \text{ per gallon}) \\ &= (11{,}000\ \text{kWh/yr})(1.54\ \text{lbm/kWh}) + (1500\ \text{gal/yr})(26.4\ \text{lbm/gal}) \\ &= 56{,}540\ CO_2\ \text{lbm/year}\end{aligned}$$

Then reducing the electricity and fuel oil usage by 15% will reduce the annual amount of CO_2 production by this household by

$$\begin{aligned}\text{Reduction in } CO_2 \text{ produced} &= (0.15)(\text{Current amount of } CO_2 \text{ production}) \\ &= (0.15)(56{,}540\ CO_2\ \text{kg/year}) \\ &= \mathbf{8481\ CO_2\ lbm/year}\end{aligned}$$

Therefore, any measure that saves energy also reduces the amount of pollution emitted to the environment.

2-88 A household has 2 cars, a natural gas furnace for heating, and uses electricity for other energy needs. The annual amount of NO_x emission to the atmosphere this household is responsible for is to be determined.

Properties The amount of NO_x produced is 7.1 g per kWh, 4.3 g per therm of natural gas, and 11 kg per car (given).



Analysis Noting that this household has 2 cars, consumes 1200 therms of natural gas, and 9,000 kWh of electricity per year, the amount of NO_x production this household is responsible for is

$$\begin{aligned}\text{Amount of } NO_x \text{ produced} &= (\text{No. of cars})(\text{Amount of } NO_x \text{ produced per car}) \\ &\quad + (\text{Amount of electricity consumed})(\text{Amount of } NO_x \text{ per kWh}) \\ &\quad + (\text{Amount of gas consumed})(\text{Amount of } NO_x \text{ per gallon}) \\ &= (2\ \text{cars})(11\ \text{kg/car}) + (9000\ \text{kWh/yr})(0.0071\ \text{kg/kWh}) \\ &\quad + (1200\ \text{therms/yr})(0.0043\ \text{kg/therm}) \\ &= \mathbf{91.06\ NO_x\ kg/year}\end{aligned}$$

Discussion Any measure that saves energy will also reduce the amount of pollution emitted to the atmosphere.

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Special Topic: Mechanisms of Heat Transfer

2-89C The three mechanisms of heat transfer are conduction, convection, and radiation.

2-90C Diamond has a higher thermal conductivity than silver, and thus diamond is a better conductor of heat.

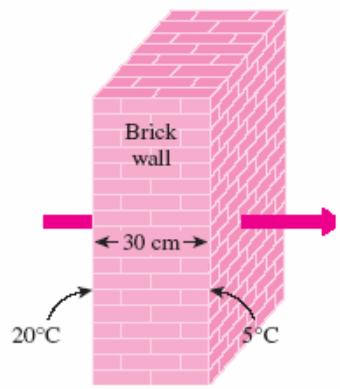
2-91C No. It is purely by radiation.

2-92C In forced convection, the fluid is forced to move by external means such as a fan, pump, or the wind. The fluid motion in natural convection is due to buoyancy effects only.

2-93C A blackbody is an idealized body that emits the maximum amount of radiation at a given temperature, and that absorbs all the radiation incident on it. Real bodies emit and absorb less radiation than a blackbody at the same temperature.

2-94C Emissivity is the ratio of the radiation emitted by a surface to the radiation emitted by a blackbody at the same temperature. Absorptivity is the fraction of radiation incident on a surface that is absorbed by the surface. The Kirchhoff's law of radiation states that the emissivity and the absorptivity of a surface are equal at the same temperature and wavelength.

2-95 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of heat transfer through the wall is to be determined.



Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Thermal properties of the wall are constant.

Properties The thermal conductivity of the wall is given to be $k = 0.69$ W/m·°C.

Analysis Under steady conditions, the rate of heat transfer through the wall is

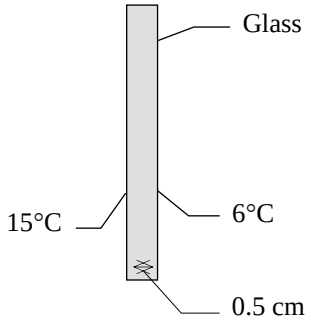
$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.69\ \text{W/m}\cdot\text{°C})(5\times 6\ \text{m}^2)\frac{(20-5)\text{°C}}{0.3\ \text{m}} = \mathbf{1035\ W}$$

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2-96 The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat transferred through the glass in 5 h is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Thermal properties of the glass are constant.

Properties The thermal conductivity of the glass is given to be k = 0.78 W/m·°C.



Analysis Under steady conditions, the rate of heat transfer through the glass by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.78\ \text{W/m}\cdot\text{°C})(2\times 2\ \text{m}^2)\frac{(15-6)\text{°C}}{0.005\ \text{m}} = 5616\ \text{W}$$

Then the amount of heat transferred over a period of 10 h becomes

$$Q = \dot{Q}_{\text{cond}}\Delta t = (5.616\ \text{kJ/s})(10\times 3600\text{s}) = \mathbf{202{,}200\ kJ}$$

If the thickness of the glass is doubled to 1 cm, then the amount of heat transferred will go down by half to **101,100 kJ**.

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2-97 Reconsider Prob. 2-96. Using EES (or other) software, investigate the effect of glass thickness on heat loss for the specified glass surface temperatures. Let the glass thickness vary from 0.2 cm to 2 cm. Plot the heat loss versus the glass thickness, and discuss the results.

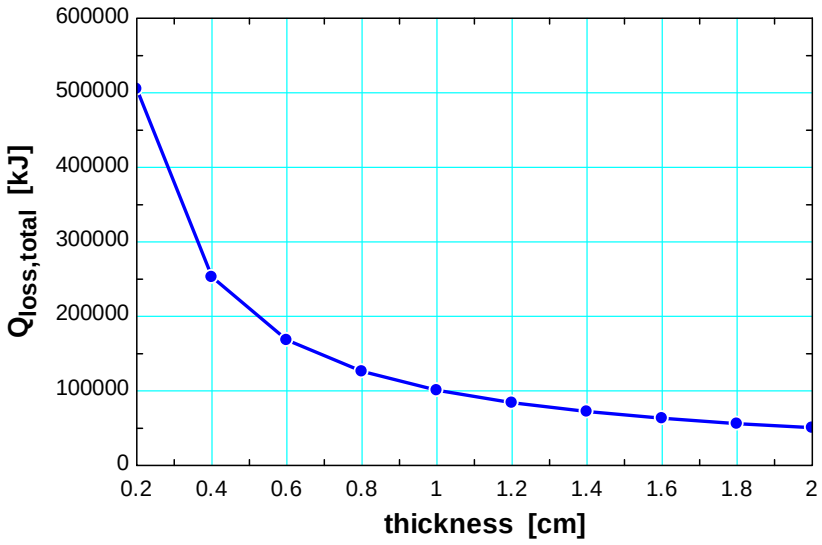
Analysis The problem is solved using EES, and the solution is given below.

```
FUNCTION  klookup(material$)
If material$='Glass'  then klookup:=0.78
If material$='Brick'  then klookup:=0.72
If material$='Fiber Glass'  then klookup:=0.043
If material$='Air'  then klookup:=0.026
If material$='Wood(oak)'  then klookup:=0.17

END

L=2 [m]
W=2 [m]
material$='Glass'
T_in=15 [C]
T_out=6 [C]
k=0.78 [W/m-C]
t=10 [hr]
thickness=0.5 [cm]
A=L*W
Q_dot_loss=A*k*(T_in-T_out)/(thickness*convert(cm,m))
Q_loss_total=Q_dot_loss*t*convert(hr,s)*convert(J,kJ)
```

Thickness [cm]	$Q_{loss,total}$ [kJ
0.2	505440
0.4	252720
0.6	168480
0.8	126360
1	101088
1.2	84240
1.4	72206
1.6	63180
1.8	56160
2	50544



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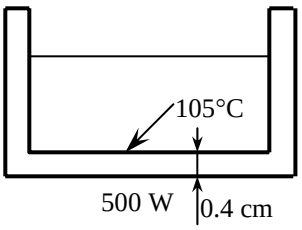
2-98 Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface temperature of the bottom of the pan is given. The temperature of the outer surface is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values. **2** Thermal properties of the aluminum pan are constant.

Properties The thermal conductivity of the aluminum is given to be k = 237 W/m·°C.

Analysis The heat transfer surface area is

$$A = \pi r^2 = \pi(0.1 \text{ m})^2 = 0.0314 \text{ m}^2$$



Under steady conditions, the rate of heat transfer through the bottom of the pan by conduction is

$$\dot{Q} = kA\frac{\Delta T}{L} = kA\frac{T_2 - T_1}{L}$$

Substituting,

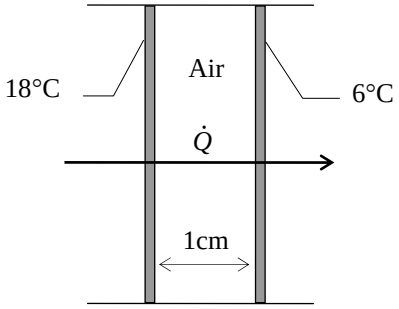
$$500 \text{ W} = (237 \text{ W/m}\cdot^\circ\text{C})(0.0314 \text{ m}^2)\frac{T_2 - 105^\circ\text{C}}{0.004 \text{ m}}$$

which gives

$$T_2 = \mathbf{105.3°C}$$

2-99 The inner and outer glasses of a double pane window with a 1-cm air space are at specified temperatures. The rate of heat transfer through the window is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Heat transfer through the window is one-dimensional. **3** Thermal properties of the air are constant. **4** The air trapped between the two glasses is still, and thus heat transfer is by conduction only.



Properties The thermal conductivity of air at room temperature is k = 0.026 W/m.°C (Table 2-3).

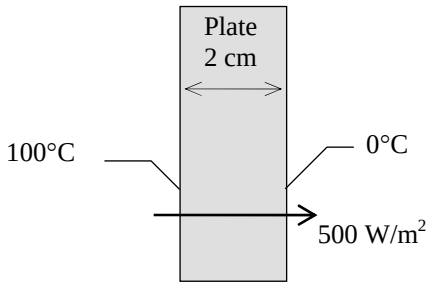
Analysis Under steady conditions, the rate of heat transfer through the window by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.026 \text{ W/m}\cdot^\circ\text{C})(2\times 2 \text{ m}^2)\frac{(18-6)^\circ\text{C}}{0.01 \text{ m}} = \mathbf{125 \text{ W}} = \mathbf{0.125 \text{ kW}}$$

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2-100 Two surfaces of a flat plate are maintained at specified temperatures, and the rate of heat transfer through the plate is measured. The thermal conductivity of the plate material is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the plate remain constant at the specified values. **2** Heat transfer through the plate is one-dimensional. **3** Thermal properties of the plate are constant.



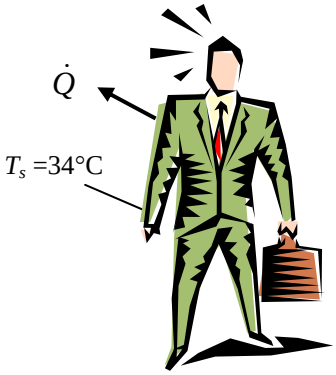
Analysis The thermal conductivity is determined directly from the steady one-dimensional heat conduction relation to be

$$\dot{Q} = kA\frac{T_1 - T_2}{L}$$

$$k = \frac{(\dot{Q}/A)L}{T_1 - T_2} = \frac{(500\ \text{W/m}^2)(0.02\ \text{m})}{(100 - 0)^\circ\text{C}} = \mathbf{0.1\ W/m.^\circ C}$$

2-101 A person is standing in a room at a specified temperature. The rate of heat transfer between a person and the surrounding air by convection is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The environment is at a uniform temperature.



Analysis The heat transfer surface area of the person is

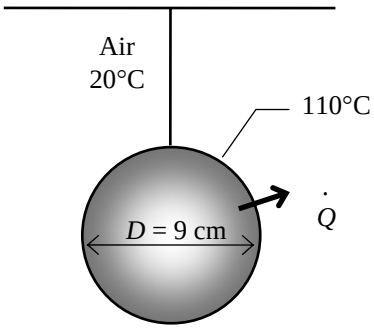
$$A = \pi DL = \pi(0.3\ \text{m})(1.70\ \text{m}) = 1.60\ \text{m}^2$$

Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T = (15\ \text{W/m}^2\cdot^\circ\text{C})(1.60\ \text{m}^2)(34 - 20)^\circ\text{C} = \mathbf{336\ W}$$

2-102 A spherical ball whose surface is maintained at a temperature of 110°C is suspended in the middle of a room at 20°C. The total rate of heat transfer from the ball is to be determined.

Assumptions **1** Steady operating conditions exist since the ball surface and the surrounding air and surfaces remain at constant temperatures. **2** The thermal properties of the ball and the convection heat transfer coefficient are constant and uniform.



Properties The emissivity of the ball surface is given to be ε = 0.8.

Analysis The heat transfer surface area is

$$A = \pi D^2 = \pi(0.09\ \text{m})^2 = 0.02545\ \text{m}^2$$

Under steady conditions, the rates of convection and radiation heat transfer are

$$\dot{Q}_{\text{conv}} = hA\Delta T = (15\ \text{W/m}^2\cdot^\circ\text{C})(0.02545\ \text{m}^2)(110 - 20)^\circ\text{C} = 34.35\ \text{W}$$

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A(T_s^4 - T_o^4) = 0.8(0.02545\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[(383\ \text{K})^4 - (293\ \text{K})^4] = 16.33\ \text{W}$$

Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 34.35 + 16.33 = \mathbf{50.7\ W}$$

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2-103 Reconsider Prob. 2-102. Using EES (or other) software, investigate the effect of the convection heat transfer coefficient and surface emissivity on the heat transfer rate from the ball. Let the heat transfer coefficient vary from 5 W/m^2.°C to 30 W/m^2.°C. Plot the rate of heat transfer against the convection heat transfer coefficient for the surface emissivities of 0.1, 0.5, 0.8, and 1, and discuss the results.

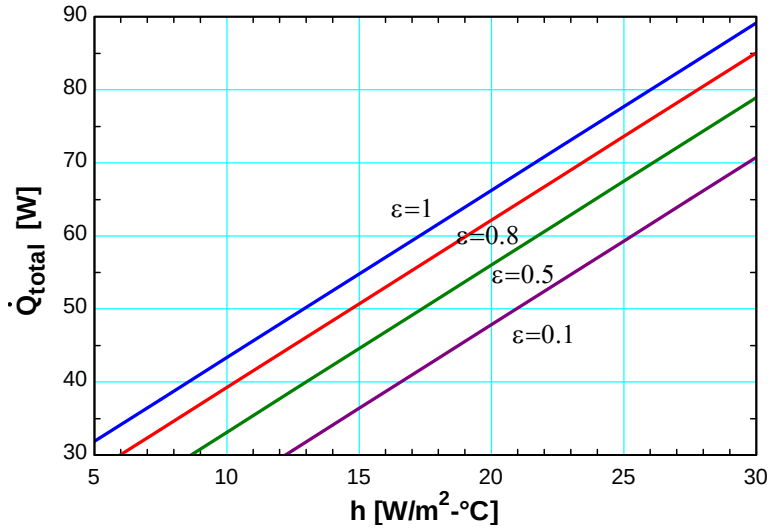
Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
D=0.09 [m]
T_s=ConvertTemp(C,K,110)
T_f=ConvertTemp(C,K,20)
h=15 [W/m^2-C]
epsilon=0.8

"Properties"
sigma=5.67E-8 [W/m^2-K^4]

"Analysis"
A=pi*D^2
Q_dot_conv=h*A*(T_s-T_f)
Q_dot_rad=epsilon*sigma*A*(T_s^4-T_f^4)
Q_dot_total=Q_dot_conv+Q_dot_rad
```

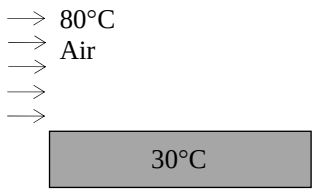
h [W/m^2-C]	Q_{total} [W]
5	27.8
7.5	33.53
10	39.25
12.5	44.98
15	50.7
17.5	56.43
20	62.16
22.5	67.88
25	73.61
27.5	79.33
30	85.06



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2-104 Hot air is blown over a flat surface at a specified temperature. The rate of heat transfer from the air to the plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The convection heat transfer coefficient is constant and uniform over the surface.

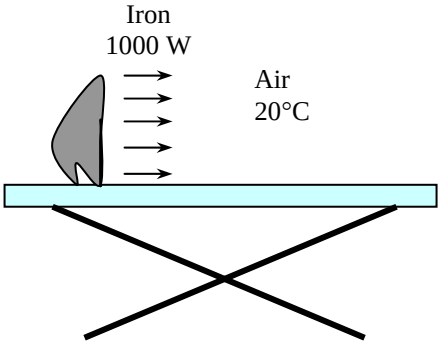


Analysis Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T$$
$$= (55\ \text{W/m}^2 \cdot {}^\circ\text{C})(2 \times 4\ \text{m}^2)(80 - 30)^\circ\text{C}$$
$$= \mathbf{22{,}000\ W} = \mathbf{22\ kW}$$

2-105 A 1000-W iron is left on the iron board with its base exposed to the air at 20°C. The temperature of the base of the iron is to be determined in steady operation.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the iron base and the convection heat transfer coefficient are constant and uniform. **3** The temperature of the surrounding surfaces is the same as the temperature of the surrounding air.



Properties The emissivity of the base surface is given to be $\varepsilon = 0.6$.

Analysis At steady conditions, the 1000 W of energy supplied to the iron will be dissipated to the surroundings by convection and radiation heat transfer. Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 1000\ \text{W}$$

where

$$\dot{Q}_{\text{conv}} = hA\Delta T = (35\ \text{W/m}^2 \cdot \text{K})(0.02\ \text{m}^2)(T_s - 293\ \text{K}) = 0.7(T_s - 293\ \text{K})\ \text{W}$$

and

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A(T_s^4 - T_o^4) = 0.6(0.02\ \text{m}^2)(5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4)[T_s^4 - (293\ \text{K})^4]$$
$$= 0.06804 \times 10^{-8}[T_s^4 - (293\ \text{K})^4]\ \text{W}$$

Substituting,

$$1000\ \text{W} = 0.7(T_s - 293\ \text{K}) + 0.06804 \times 10^{-8}[T_s^4 - (293\ \text{K})^4]$$

Solving by trial and error gives

$$T_s = \mathbf{947\ K} = \mathbf{674^\circ C}$$

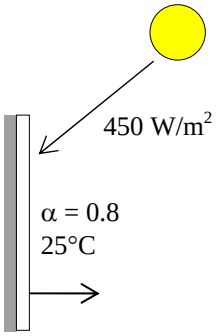
Discussion We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 947 K.

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2-106 The backside of the thin metal plate is insulated and the front side is exposed to solar radiation. The surface temperature of the plate is to be determined when it stabilizes.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the insulated side of the plate is negligible. **3** The heat transfer coefficient is constant and uniform over the plate. **4** Heat loss by radiation is negligible.

Properties The solar absorptivity of the plate is given to be $\alpha = 0.8$.



Analysis When the heat loss from the plate by convection equals the solar radiation absorbed, the surface temperature of the plate can be determined from

$$\dot{Q}_{\text{solarabsorbed}} = \dot{Q}_{\text{conv}}$$

$$\alpha \dot{Q}_{\text{solar}} = hA(T_s - T_o)$$

$$0.8 \times A \times 450 \text{ W/m}^2 = (50 \text{ W/m}^2 \cdot ^\circ\text{C})A(T_s - 25)$$

Canceling the surface area A and solving for T_s gives

$$T_s = \mathbf{32.2^\circ C}$$

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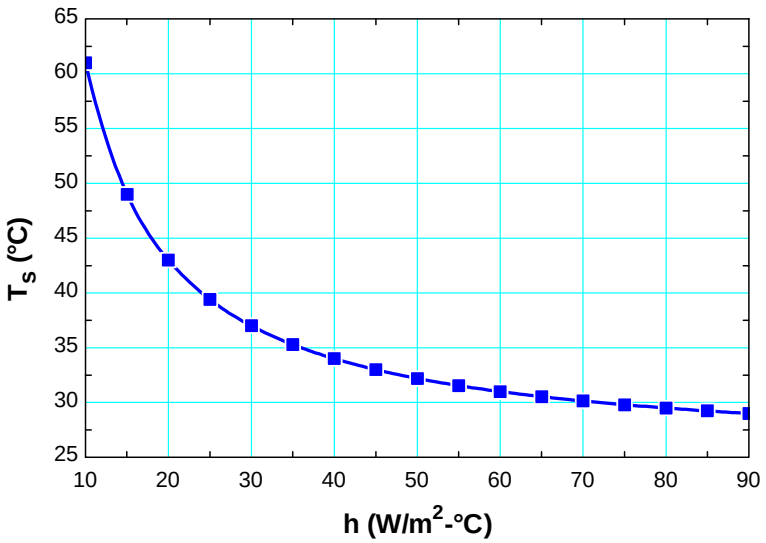
2-107 Reconsider Prob. 2-106. Using EES (or other) software, investigate the effect of the convection heat transfer coefficient on the surface temperature of the plate. Let the heat transfer coefficient vary from 10 W/m^2.°C to 90 W/m^2.°C. Plot the surface temperature against the convection heat transfer coefficient, and discuss the results.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
alpha=0.8
q_dot_solar=450 [W/m^2]
T_f=25 [C]
h=50 [W/m^2-C]

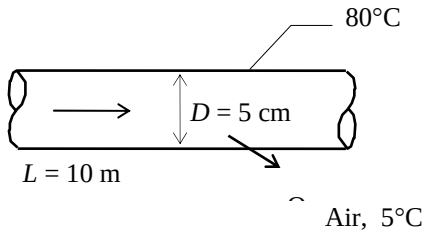
"Analysis"
q_dot_solarabsorbed=alpha*q_dot_solar
q_dot_conv=h*(T_s-T_f)
q_dot_solarabsorbed=q_dot_conv
```

h [W/m^2-C]	T_s [C]
10	61
15	49
20	43
25	39.4
30	37
35	35.29
40	34
45	33
50	32.2
55	31.55
60	31
65	30.54
70	30.14
75	29.8
80	29.5
85	29.24
90	29



2-108 A hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 W/ m^2.°C. The rate of heat loss from the pipe by convection is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The convection heat transfer coefficient is constant and uniform over the surface.



Analysis The heat transfer surface area is

$$A = (\pi D)L = 3.14\text{x}(0.05\text{ m})(10\text{ m}) = 1.571\text{ m}^2$$

Under steady conditions, the rate of heat transfer by convection is

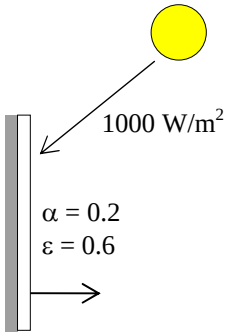
$$\dot{Q}_{\text{conv}} = hA\Delta T = (25\text{ W/m}^2\cdot\text{°C})(1.571\text{ m}^2)(80-5)\text{°C} = \mathbf{2945\ W} = \mathbf{2.95\ kW}$$

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2-109 A spacecraft in space absorbs solar radiation while losing heat to deep space by thermal radiation. The surface temperature of the spacecraft is to be determined when steady conditions are reached..

Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Thermal properties of the spacecraft are constant.

Properties The outer surface of a spacecraft has an emissivity of 0.6 and an absorptivity of 0.2.



Analysis When the heat loss from the outer surface of the spacecraft by radiation equals the solar radiation absorbed, the surface temperature can be determined from

$$\dot{Q}_{\text{solarabsorbed}} = \dot{Q}_{\text{rad}}$$

$$\alpha \dot{Q}_{\text{solar}} = \varepsilon \sigma A (T_s^4 - T_{\text{space}}^4)$$

$$0.2 \times A \times (1000 \text{ W/m}^2) = 0.6 \times A \times (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[T_s^4 - (0 \text{ K})^4]$$

Canceling the surface area A and solving for T_s gives

$$T_s = \mathbf{276.9\ K}$$

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2-110 Reconsider Prob. 2-109. Using EES (or other) software, investigate the effect of the surface emissivity and absorptivity of the spacecraft on the equilibrium surface temperature. Plot the surface temperature against emissivity for solar absorptivities of 0.1, 0.5, 0.8, and 1, and discuss the results.

Analysis The problem is solved using EES, and the solution is given below.

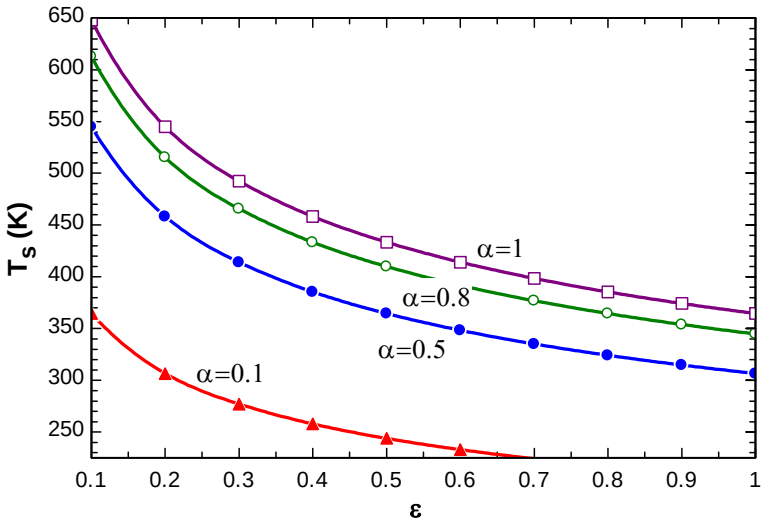
```
"Given"
epsilon=0.2
alpha=0.6
q_dot_solar=1000 [W/m^2]
T_f=0 [K] "space temperature"

"Properties"
sigma=5.67E-8 [W/m^2-K^4]

"Analysis"
q_dot_solarabsorbed=alpha*q_dot_solar
q_dot_rad=epsilon*sigma*(T_s^4-T_f^4)
q_dot_solarabsorbed=q_dot_rad
```

ε	T_s [K]
0.1	648
0.2	544.9
0.3	492.4
0.4	458.2
0.5	433.4
0.6	414.1
0.7	398.4
0.8	385.3
0.9	374.1
1	364.4

Table for ε = 1



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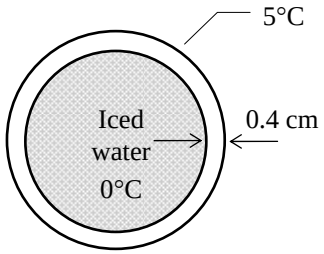
2-111 A hollow spherical iron container is filled with iced water at 0°C. The rate of heat loss from the sphere and the rate at which ice melts in the container are to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Heat transfer through the shell is one-dimensional. **3** Thermal properties of the iron shell are constant. **4** The inner surface of the shell is at the same temperature as the iced water, 0°C.

Properties The thermal conductivity of iron is k = 80.2 W/m·°C (Table 2-3). The heat of fusion of water is at 1 atm is 333.7 kJ/kg.

Analysis This spherical shell can be approximated as a plate of thickness 0.4 cm and surface area

$$A = \pi D^2 = 3.14\times(0.2\text{ m})^2 = 0.126\text{ m}^2$$



Then the rate of heat transfer through the shell by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (80.2\text{ W/m}\cdot^\circ\text{C})(0.126\text{ m}^2)\frac{(5-0)^\circ\text{C}}{0.004\text{ m}} = \mathbf{12{,}632\text{ W}}$$

Considering that it takes 333.7 kJ of energy to melt 1 kg of ice at 0°C, the rate at which ice melts in the container can be determined from

$$\dot{m}_{\text{ice}} = \frac{\dot{Q}}{h_{if}} = \frac{12.632\text{ kJ/s}}{333.7\text{ kJ/kg}} = \mathbf{0.038\text{ kg/s}}$$

Discussion We should point out that this result is slightly in error for approximating a curved wall as a plain wall. The error in this case is very small because of the large diameter to thickness ratio. For better accuracy, we could use the inner surface area (D = 19.2 cm) or the mean surface area (D = 19.6 cm) in the calculations.

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Review Problems

2-112 A classroom has a specified number of students, instructors, and fluorescent light bulbs. The rate of internal heat generation in this classroom is to be determined.

Assumptions **1** There is a mix of men, women, and children in the classroom. **2** The amount of light (and thus energy) leaving the room through the windows is negligible.

Properties The average rate of heat generation from people seated in a room/office is given to be 100 W.

Analysis The amount of heat dissipated by the lamps is equal to the amount of electrical energy consumed by the lamps, including the 10% additional electricity consumed by the ballasts. Therefore,

$$\dot{Q}_{\text{lighting}} = (\text{Energy consumed per lamp}) \times (\text{No. of lamps})$$
$$= (40\ \text{W})(1.1)(18) = 792\ \text{W}$$
$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 56 \times (100\ \text{W}) = 5600\ \text{W}$$



Then the total rate of heat gain (or the internal heat load) of the classroom from the lights and people become

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{lighting}} + \dot{Q}_{\text{people}} = 792 + 5600 = \mathbf{6392\ W}$$

2-113 A decision is to be made between a cheaper but inefficient natural gas heater and an expensive but efficient natural gas heater for a house.

Assumptions The two heaters are comparable in all aspects other than the initial cost and efficiency.

Analysis Other things being equal, the logical choice is the heater that will cost less during its lifetime. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period.

The annual heating cost is given to be $1200. Noting that the existing heater is 55% efficient, only 55% of that energy (and thus money) is delivered to the house, and the rest is wasted due to the inefficiency of the heater. Therefore, the monetary value of the heating load of the house is

Gas Heater
$\eta_1 = 82\%$
$\eta_2 = 95\%$

Cost of useful heat = (55%)(Current annual heating cost)
= 0.55×($1200/yr)=$660/yr

This is how much it would cost to heat this house with a heater that is 100% efficient. For heaters that are less efficient, the annual heating cost is determined by dividing $660 by the efficiency:

82% heater: Annual cost of heating = (Cost of useful heat)/Efficiency = ($660/yr)/0.82 = $805/yr

95% heater: Annual cost of heating = (Cost of useful heat)/Efficiency = ($660/yr)/0.95 = $695/yr

Annual cost savings with the efficient heater = 805 - 695 = $110

Excess initial cost of the efficient heater = 2700 - 1600 = $1100

The simple payback period becomes

$$\text{Simple payback period} = \frac{\text{Excess initial cost}}{\text{Annaul cost savings}} = \frac{\$1100}{\$110/\text{yr}} = \mathbf{10\ years}$$

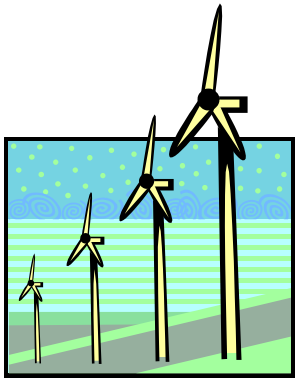
Therefore, the more efficient heater will pay for the $1100 cost differential in this case in 10 years, which is more than the 8-year limit. Therefore, the purchase of the cheaper and less efficient heater is a better buy in this case.

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2-114 A wind turbine is rotating at 20 rpm under steady winds of 30 km/h. The power produced, the tip speed of the blade, and the revenue generated by the wind turbine per year are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The wind turbine operates continuously during the entire year at the specified conditions.

Properties The density of air is given to be $\rho = 1.20$ kg/m^3.



Analysis (*a*) The blade span area and the mass flow rate of air through the turbine are

$$A = \pi D^2/4 = \pi(80\text{ m})^2/4 = 5027\text{ m}^2$$

$$V = (30\text{ km/h})\left(\frac{1000\text{ m}}{1\text{ km}}\right)\left(\frac{1\text{ h}}{3600\text{ s}}\right) = 8.333\text{ m/s}$$

$$\dot{m} = \rho AV = (1.2\text{ kg/m}^3)(5027\text{ m}^2)(8.333\text{ m/s}) = 50{,}270\text{ kg/s}$$

Noting that the kinetic energy of a unit mass is $V^2/2$ and the wind turbine captures 35% of this energy, the power generated by this wind turbine becomes

$$\dot{W} = \eta\left(\frac{1}{2}\dot{m}V^2\right) = (0.35)\frac{1}{2}(50{,}270\text{ kg/s})(8.333\text{ m/s})^2\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{610.9\ kW}$$

(*b*) Noting that the tip of blade travels a distance of πD per revolution, the tip velocity of the turbine blade for an rpm of $\dot{n}$ becomes

$$V_{\text{tip}} = \pi D\dot{n} = \pi(80\text{ m})(20/\text{min}) = 5027\text{ m/min} = 83.8\text{ m/s} = \mathbf{302\ km/h}$$

(*c*) The amount of electricity produced and the revenue generated per year are

$$\text{Electricity produced} = \dot{W}\Delta t = (610.9\text{ kW})(365\times 24\text{ h/year})$$
$$= 5.351\times 10^6\text{ kWh/year}$$

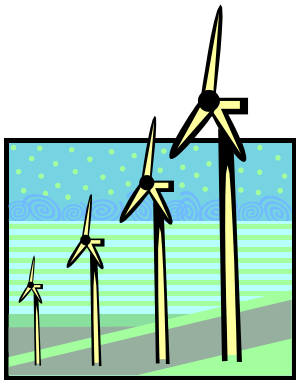
$$\text{Revenue generated} = (\text{Electricity produced})(\text{Unit price}) = (5.351\times 10^6\text{ kWh/year})(\$0.06/\text{kWh})$$
$$= \mathbf{\$321{,}100/year}$$

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2-115 A wind turbine is rotating at 20 rpm under steady winds of 20 km/h. The power produced, the tip speed of the blade, and the revenue generated by the wind turbine per year are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The wind turbine operates continuously during the entire year at the specified conditions.

Properties The density of air is given to be $\rho = 1.20$ kg/m^3.



Analysis (*a*) The blade span area and the mass flow rate of air through the turbine are

$$A = \pi D^2/4 = \pi(80\text{ m})^2/4 = 5027\text{ m}^2$$

$$V = (20\text{ km/h})\left(\frac{1000\text{ m}}{1\text{ km}}\right)\left(\frac{1\text{ h}}{3600\text{ s}}\right) = 5.556\text{ m/s}$$

$$\dot{m} = \rho AV = (1.2\text{ kg/m}^3)(5027\text{ m}^2)(5.556\text{ m/s}) = 33{,}510\text{ kg/s}$$

Noting that the kinetic energy of a unit mass is $V^2/2$ and the wind turbine captures 35% of this energy, the power generated by this wind turbine becomes

$$\dot{W} = \eta\left(\frac{1}{2}\dot{m}V^2\right) = (0.35)\frac{1}{2}(33{,}510\text{ kg/s})(6.944\text{ m/s})^2\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{181.0\text{ kW}}$$

(*b*) Noting that the tip of blade travels a distance of πD per revolution, the tip velocity of the turbine blade for an rpm of $\dot{n}$ becomes

$$V_{\text{tip}} = \pi D\dot{n} = \pi(80\text{ m})(20/\text{min}) = 5027\text{ m/min} = 83.8\text{ m/s} = \mathbf{302\text{ km/h}}$$

(*c*) The amount of electricity produced and the revenue generated per year are

$$\text{Electricity produced} = \dot{W}\Delta t = (181.0\text{ kW})(365\times 24\text{ h/year})$$
$$= 1{,}585{,}535\text{ kWh/year}$$
$$\text{Revenue generated} = (\text{Electricity produced})(\text{Unit price}) = (1{,}585{,}535\text{ kWh/year})(\$0.06/\text{kWh})$$
$$= \mathbf{\$95{,}130/year}$$

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2-116E The energy contents, unit costs, and typical conversion efficiencies of various energy sources for use in water heaters are given. The lowest cost energy source is to be determined.

Assumptions The differences in installation costs of different water heaters are not considered.

Properties The energy contents, unit costs, and typical conversion efficiencies of different systems are given in the problem statement.

Analysis The unit cost of each Btu of useful energy supplied to the water heater by each system can be determined from

$$\text{Unit cost of useful energy} = \frac{\text{Unit cost of energy supplied}}{\text{Conversion efficiency}}$$

Substituting,

Natural gas heater: $$\text{Unit cost of useful energy} = \frac{\$0.012/\text{ft}^3}{0.55}\left(\frac{1\,\text{ft}^3}{1025\,\text{Btu}}\right) = \$21.3\times10^{-6}\ /\ \text{Btu}$$

Heating by oil heater: $$\text{Unit cost of useful energy} = \frac{\$1.15/\text{gal}}{0.55}\left(\frac{1\,\text{gal}}{138{,}700\,\text{Btu}}\right) = \$15.1\times10^{-6}\ /\ \text{Btu}$$

Electric heater: $$\text{Unit cost of useful energy} = \frac{\$0.084/\text{kWh})}{0.90}\left(\frac{1\,\text{kWh}}{3412\,\text{Btu}}\right) = \$27.4\times10^{-6}\ /\ \text{Btu}$$

Therefore, the lowest cost energy source for hot water heaters in this case is **oil**.

2-117 A home owner is considering three different heating systems for heating his house. The system with the lowest energy cost is to be determined.

Assumptions The differences in installation costs of different heating systems are not considered.

Properties The energy contents, unit costs, and typical conversion efficiencies of different systems are given in the problem statement.

Analysis The unit cost of each Btu of useful energy supplied to the house by each system can be determined from

$$\text{Unit cost of useful energy} = \frac{\text{Unit cost of energy supplied}}{\text{Conversion efficiency}}$$

Substituting,

Natural gas heater: $$\text{Unit cost of useful energy} = \frac{\$1.24/\text{therm}}{0.87}\left(\frac{1\,\text{therm}}{105{,}500\,\text{kJ}}\right) = \$13.5\times10^{-6}\ /\ \text{kJ}$$

Heating oil heater: $$\text{Unit cost of useful energy} = \frac{\$1.25/\text{gal}}{0.87}\left(\frac{1\,\text{gal}}{138{,}500\,\text{kJ}}\right) = \$10.4\times10^{-6}\ /\ \text{kJ}$$

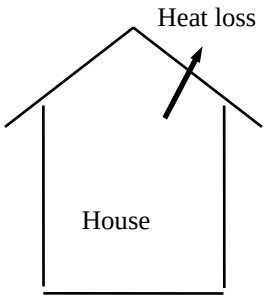
Electric heater: $$\text{Unit cost of useful energy} = \frac{\$0.09/\text{kWh})}{1.0}\left(\frac{1\,\text{kWh}}{3600\,\text{kJ}}\right) = \$25.0\times10^{-6}\ /\ \text{kJ}$$

Therefore, the system with the lowest energy cost for heating the house is the **heating oil heater**.

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2-118 The heating and cooling costs of a poorly insulated house can be reduced by up to 30 percent by adding adequate insulation. The time it will take for the added insulation to pay for itself from the energy it saves is to be determined.

Assumptions It is given that the annual energy usage of a house is $1200 a year, and 46% of it is used for heating and cooling. The cost of added insulation is given to be $200.



Analysis The amount of money that would be saved per year is determined directly from

$$\text{Money saved} = (\$1200/\text{year})(0.46)(0.30) = \$166/\text{yr}$$

Then the simple payback period becomes

$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}} = \frac{\$200}{\$166/\text{yr}} = \mathbf{1.2\ yr}$$

Therefore, the proposed measure will pay for itself in less than one and a half year.

2-119 Caulking and weather-stripping doors and windows to reduce air leaks can reduce the energy use of a house by up to 10 percent. The time it will take for the caulking and weather-stripping to pay for itself from the energy it saves is to be determined.

Assumptions It is given that the annual energy usage of a house is $1100 a year, and the cost of caulking and weather-stripping a house is $60.

Analysis The amount of money that would be saved per year is determined directly from

$$\text{Money saved} = (\$1100/\text{year})(0.10) = \$110/\text{yr}$$

Then the simple payback period becomes

$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}} = \frac{\$60}{\$110/\text{yr}} = \mathbf{0.546\ yr}$$

Therefore, the proposed measure will pay for itself in less than half a year.

2-120E The energy stored in the spring of a railroad car is to be expressed in different units.

Analysis Using appropriate conversion factors, we obtain

(*a*) $$W = (5000\ \text{lbf}\cdot\text{ft})\left(\frac{32.174\ \text{lbm}\cdot\text{ft/s}^2}{1\ \text{lbf}}\right) = \mathbf{160{,}870\ lbm\cdot ft^2/s^2}$$

(*b*) $$W = (5000\ \text{lbf}\cdot\text{ft})\left(\frac{0.33303\ \text{yd}}{1\ \text{ft}}\right) = \mathbf{1665\ lbf\cdot yd}$$

(*c*) $$W = (5000\ \text{lbf}\cdot\text{ft})\left(\frac{32.174\ \text{lbm}\cdot\text{ft/s}^2}{1\ \text{lbf}}\right)\left(\frac{1\ \text{mile}}{5280\ \text{ft}}\right)^2\left(\frac{3600\ \text{s}}{1\ \text{h}}\right)^2 = \mathbf{74{,}785\ lbm\cdot mile^2/h^2}$$

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2-121E The work required to compress a gas in a gas spring is to be determined.

Assumptions All forces except that generated by the gas spring will be neglected.

Analysis When the expression given in the problem statement is substituted into the work integral relation, and advantage is taken of the fact that the force and displacement vectors are collinear, the result is

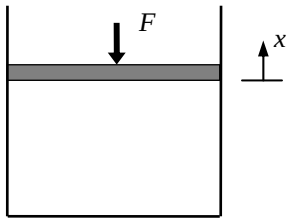
$$W = \int_1^2 F ds = \int_1^2 \frac{\text{Constant}}{x^k} dx$$

$$= \frac{\text{Constant}}{1-k}(x_2^{1-k} - x_1^{1-k})$$

$$= \frac{200\ \text{lbf}\cdot\text{in}^{1.4}}{1-1.4}\left[(4\ \text{in})^{-0.4} - (1\ \text{in})^{-0.4}\right]\left(\frac{1\ \text{ft}}{12\ \text{in}}\right)$$

$$= 17.74\ \text{lbf}\cdot\text{ft}$$

$$= (17.74\ \text{lbf}\cdot\text{ft})\left(\frac{1\ \text{Btu}}{778.169\ \text{lbf}\cdot\text{ft}}\right) = \mathbf{0.0228\ Btu}$$



2-122E A man pushes a block along a horizontal plane. The work required to move the block is to be determined considering (a) the man and (b) the block as the system.

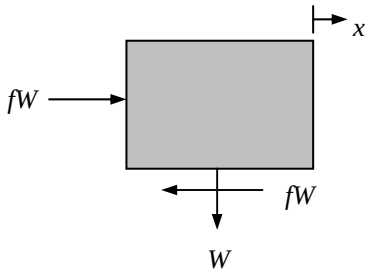
Analysis The work applied to the block to overcome the friction is found by using the work integral,

$$W = \int_1^2 F ds = \int_1^2 fW(x_2 - x_1)$$

$$= (0.2)(100\ \text{lbf})(100\ \text{ft})$$

$$= 2000\ \text{lbf}\cdot\text{ft}$$

$$= (2000\ \text{lbf}\cdot\text{ft})\left(\frac{1\ \text{Btu}}{778.169\ \text{lbf}\cdot\text{ft}}\right) = \mathbf{2.57\ Btu}$$



The man must then produce the amount of work

$$W = \mathbf{2.57\ Btu}$$

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2-123 A diesel engine burning light diesel fuel that contains sulfur is considered. The rate of sulfur that ends up in the exhaust and the rate of sulfurous acid given off to the environment are to be determined.

Assumptions **1** All of the sulfur in the fuel ends up in the exhaust. **2** For one kmol of sulfur in the exhaust, one kmol of sulfurous acid is added to the environment.

Properties The molar mass of sulfur is 32 kg/kmol.

Analysis The mass flow rates of fuel and the sulfur in the exhaust are

$$\dot{m}_{\text{fuel}} = \frac{\dot{m}_{\text{air}}}{\text{AF}} = \frac{(336\ \text{kg air/h})}{(18\ \text{kg air/kg fuel})} = 18.67\ \text{ kg fuel/h}$$

$$\dot{m}_{\text{Sulfur}} = (750\times10^{-6})\dot{m}_{\text{fuel}} = (750\times10^{-6})(18.67\ \text{kg/h}) = \mathbf{0.014\ kg/h}$$

The rate of sulfurous acid given off to the environment is

$$\dot{m}_{\text{H2SO3}} = \frac{M_{\text{H2SO3}}}{M_{\text{Sulfur}}}\dot{m}_{\text{Sulfur}} = \frac{2\times1+32+3\times16}{32}(0.014\ \text{kg/h}) = \mathbf{0.036\ kg/h}$$

Discussion This problem shows why the sulfur percentage in diesel fuel must be below certain value to satisfy regulations.

2-124 Lead is a very toxic engine emission. Leaded gasoline contains lead that ends up in the exhaust. The amount of lead put out to the atmosphere per year for a given city is to be determined.

Assumptions 35% of lead is exhausted to the environment.

Analysis The gasoline consumption and the lead emission are

$$\text{Gasoline Consumption} = (5000\ \text{cars})(15{,}000\ \text{ km/car - year})(8.5\,\text{L}/100\ \text{ km}) = 6.375\times10^{6}\ \text{L/year}$$

$$\begin{aligned}\text{Lead Emission} &= (\text{GaolineConsumption})m_{\text{lead}}\,f_{\text{lead}}\\ &= (6.375\times10^{6}\ \text{L/year})(0.15\times10^{-3}\ \text{kg/L})(0.35)\\ &= \mathbf{335\ kg/year}\end{aligned}$$

Discussion Note that a huge amount of lead emission is avoided by the use of unleaded gasoline.

2-125E The power required to pump a specified rate of water to a specified elevation is to be determined.

Properties The density of water is taken to be 62.4 lbm/ft^3 (Table A-3E).

Analysis The required power is determined from

$$\begin{aligned}\dot{W} &= \dot{m}g(z_2 - z_1) = \rho\dot{V}g(z_2 - z_1)\\ &= (62.4\ \text{lbm/ft}^3)(200\ \text{gal/min})\left(\frac{35.315\ \text{ft}^3/\text{s}}{15{,}850\ \text{gal/min}}\right)(32.174\ \text{ft/s}^2)(300\ \text{ft})\left(\frac{1\ \text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/s}^2}\right)\\ &= 8342\ \text{lbf}\cdot\text{ft/s} = (8342\ \text{lbf}\cdot\text{ft/s})\left(\frac{1\ \text{kW}}{737.56\ \text{lbf}\cdot\text{ft/s}}\right) = \mathbf{11.3\ kW}\end{aligned}$$

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2-126 The power that could be produced by a water wheel is to be determined.

Properties The density of water is taken to be 1000 m^3/kg (Table A-3).

Analysis The power production is determined from

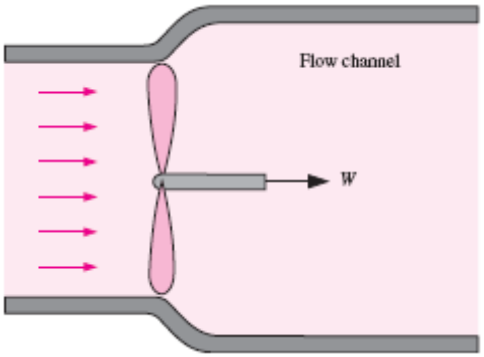
$$\dot{W} = \dot{m}g(z_2 - z_1) = \rho\dot{V}g(z_2 - z_1)$$

$$= (1000\,\text{kg/m}^3)(0.320/60\,\text{m}^3/\text{s})(9.81\,\text{m/s}^2)(14\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{0.732\ kW}$$

2-127 The flow of air through a flow channel is considered. The diameter of the wind channel downstream from the rotor and the power produced by the windmill are to be determined.

Analysis The specific volume of the air is

$$\nu = \frac{RT}{P} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{100\,\text{kPa}} = 0.8409\,\text{m}^3/\text{kg}$$



The diameter of the wind channel downstream from the rotor is

$$A_1V_1 = A_2V_2 \longrightarrow (\pi D_1^2/4)V_1 = (\pi D_2^2/4)V_2$$

$$\longrightarrow D_2 = D_1\sqrt{\frac{V_1}{V_2}} = (7\,\text{m})\sqrt{\frac{10\,\text{m/s}}{9\,\text{m/s}}} = \mathbf{7.38\ m}$$

The mass flow rate through the wind mill is

$$\dot{m} = \frac{A_1V_1}{\nu} = \frac{\pi(7\,\text{m})^2(10\,\text{m/s})}{4(0.8409\ \text{m}^3/\text{kg})} = 457.7\,\text{kg/s}$$

The power produced is then

$$\dot{W} = \dot{m}\frac{V_1^2 - V_2^2}{2} = (457.7\,\text{kg/s})\frac{(10\,\text{m/s})^2 - (9\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{4.35\ kW}$$

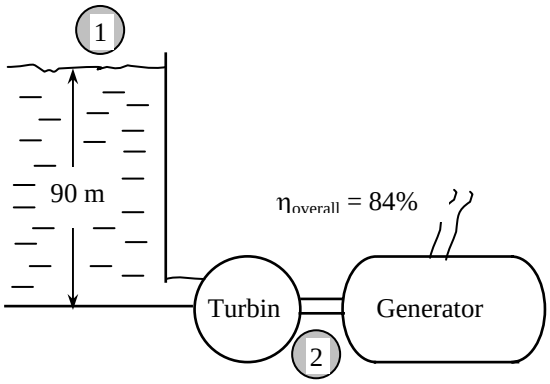
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2-128 The available head, flow rate, and efficiency of a hydroelectric turbine are given. The electric power output is to be determined.

Assumptions **1** The flow is steady. **2** Water levels at the reservoir and the discharge site remain constant. **3** Frictional losses in piping are negligible.

Properties We take the density of water to be $\rho = 1000\ \text{kg/m}^3 = 1\ \text{kg/L}$.

Analysis The total mechanical energy the water in a dam possesses is equivalent to the potential energy of water at the free surface of the dam (relative to free surface of discharge water), and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is gz per unit mass, and $\dot{m}gz$ for a given mass flow rate.



$$e_{\text{mech}} = pe = gz = (9.81\,\text{m/s}^2)(90\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 0.8829\,\text{kJ/kg}$$

The mass flow rate is

$$\dot{m} = \rho\dot{V} = (1000\,\text{kg/m}^3)(65\,\text{m}^3/\text{s}) = 65{,}000\,\text{kg/s}$$

Then the maximum and actual electric power generation become

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (65{,}000\,\text{kg/s})(0.8829\,\text{kJ/kg})\left(\frac{1\,\text{MW}}{1000\,\text{kJ/s}}\right) = 57.39\,\text{MW}$$

$$\dot{W}_{\text{electric}} = \eta_{\text{overall}}\dot{W}_{\max} = 0.84(57.39\,\text{MW}) = \mathbf{48.2\ MW}$$

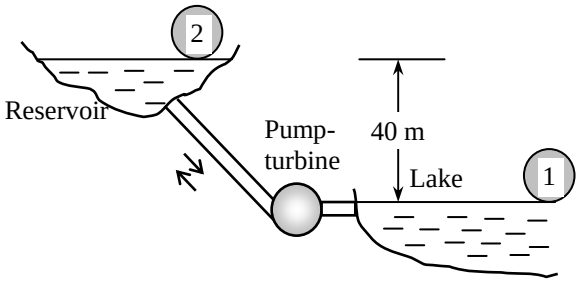
Discussion Note that the power generation would increase by more than 1 MW for each percentage point improvement in the efficiency of the turbine–generator unit.

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2-129 An entrepreneur is to build a large reservoir above the lake level, and pump water from the lake to the reservoir at night using cheap power, and let the water flow from the reservoir back to the lake during the day, producing power. The potential revenue this system can generate per year is to be determined.

Assumptions **1** The flow in each direction is steady and incompressible. **2** The elevation difference between the lake and the reservoir can be taken to be constant, and the elevation change of reservoir during charging and discharging is disregarded. **3** Frictional losses in piping are negligible. **4** The system operates every day of the year for 10 hours in each mode.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.



Analysis The total mechanical energy of water in an upper reservoir relative to water in a lower reservoir is equivalent to the potential energy of water at the free surface of this reservoir relative to free surface of the lower reservoir. Therefore, the power potential of water is its potential energy, which is *gz* per unit mass, and $\dot{m}gz$ for a given mass flow rate. This also represents the minimum power required to pump water from the lower reservoir to the higher reservoir.

$$\dot{W}_{\text{max, turbine}} = \dot{W}_{\text{min, pump}} = \dot{W}_{\text{ideal}} = \Delta \dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho \dot{V} g\Delta z$$

$$= (1000\,\text{kg/m}^3)(2\,\text{m}^3/\text{s})(9.81\,\text{m/s}^2)(40\,\text{m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kW}}{1000\,\text{N}\cdot\text{m/s}}\right)$$

$$= 784.8\,\text{kW}$$

The actual pump and turbine electric powers are

$$\dot{W}_{\text{pump, elect}} = \frac{\dot{W}_{\text{ideal}}}{\eta_{\text{pump-motor}}} = \frac{784.8\,\text{kW}}{0.75} = 1046\,\text{kW}$$

$$\dot{W}_{\text{turbine}} = \eta_{\text{turbine-gen}}\dot{W}_{\text{ideal}} = 0.75(784.8\,\text{kW}) = 588.6\ \text{kW}$$

Then the power consumption cost of the pump, the revenue generated by the turbine, and the net income (revenue minus cost) per year become

$$\text{Cost} = \dot{W}_{\text{pump, elect}}\Delta t \times \text{Unit price} = (1046\,\text{kW})(365\times 10\,\text{h/year})(\$0.03/\text{kWh}) = \$114{,}500/\text{year}$$

$$\text{Reveue} = \dot{W}_{\text{turbine}}\Delta t \times \text{Unit price} = (588.6\,\text{kW})(365\times 10\,\text{h/year})(\$0.08/\text{kWh}) = \$171{,}900/\text{year}$$

Net income = Revenue – Cost = 171,900 –114,500 = **$57,400/year**

Discussion It appears that this pump-turbine system has a potential to generate net revenues of about $57,000 per year. A decision on such a system will depend on the initial cost of the system, its life, the operating and maintenance costs, the interest rate, and the length of the contract period, among other things.

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Fundamentals of Engineering (FE) Exam Problems

2-130 A 2-kW electric resistance heater in a room is turned on and kept on for 50 min. The amount of energy transferred to the room by the heater is

(a) 2 kJ (b) 100 kJ (c) 3000 kJ (d) 6000 kJ (e) 12,000 kJ

Answer (d) 6000 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
We= 2 "kJ/s"
time=50*60 "s"
We_total=We*time "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_Etotal=We*time/60 "using minutes instead of s"
W2_Etotal=We "ignoring time"
```

2-131 In a hot summer day, the air in a well-sealed room is circulated by a 0.50-hp (shaft) fan driven by a 65% efficient motor. (Note that the motor delivers 0.50 hp of net shaft power to the fan). The rate of energy supply from the fan-motor assembly to the room is

(a) 0.769 kJ/s (b) 0.325 kJ/s (c) 0.574 kJ/s (d) 0.373 kJ/s (e) 0.242 kJ/s

Answer (c) 0.574 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Eff=0.65
W_fan=0.50*0.7457 "kW"
E=W_fan/Eff  "kJ/s"

"Some Wrong Solutions with Common Mistakes:"
W1_E=W_fan*Eff "Multiplying by efficiency"
W2_E=W_fan "Ignoring efficiency"
W3_E=W_fan/Eff/0.7457 "Using hp instead of kW"
```

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2-132 A fan is to accelerate quiescent air to a velocity to 12 m/s at a rate of 3 m^3/min. If the density of air is 1.15 kg/m^3, the minimum power that must be supplied to the fan is

(a) 248 W (b) 72 W (c) 497 W (d) 216 W (e) 162 W

Answer (a) 248 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1.15
V=12
Vdot=3 "m3/s"
mdot=rho*Vdot "kg/s"
We=mdot*V^2/2

"Some Wrong Solutions with Common Mistakes:"
W1_We=Vdot*V^2/2 "Using volume flow rate"
W2_We=mdot*V^2 "forgetting the 2"
W3_We=V^2/2 "not using mass flow rate"
```

2-133 A 900-kg car cruising at a constant speed of 60 km/h is to accelerate to 100 km/h in 4 s. The additional power needed to achieve this acceleration is

(a) 56 kW (b) 222 kW (c) 2.5 kW (d) 62 kW (e) 90 kW

Answer (a) 56 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=900 "kg"
V1=60 "km/h"
V2=100 "km/h"
Dt=4 "s"
Wa=m*((V2/3.6)^2-(V1/3.6)^2)/2000/Dt "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Wa=((V2/3.6)^2-(V1/3.6)^2)/2/Dt "Not using mass"
W2_Wa=m*((V2)^2-(V1)^2)/2000/Dt "Not using conversion factor"
W3_Wa=m*((V2/3.6)^2-(V1/3.6)^2)/2000 "Not using time interval"
W4_Wa=m*((V2/3.6)-(V1/3.6))/1000/Dt "Using velocities"
```

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2-134 The elevator of a large building is to raise a net mass of 400 kg at a constant speed of 12 m/s using an electric motor. Minimum power rating of the motor should be

(a) 0 kW (b) 4.8 kW (c) 47 kW (d) 12 kW (e) 36 kW

Answer (c) 47 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=400 "kg"
V=12 "m/s"
g=9.81 "m/s2"
Wg=m*g*V/1000 "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Wg=m*V "Not using g"
W2_Wg=m*g*V^2/2000 "Using kinetic energy"
W3_Wg=m*g/V "Using wrong relation"
```

2-135 Electric power is to be generated in a hydroelectric power plant that receives water at a rate of 70 m^3/s from an elevation of 65 m using a turbine–generator with an efficiency of 85 percent. When frictional losses in piping are disregarded, the electric power output of this plant is

(a) 3.9 MW (b) 38 MW (c) 45 MW (d) 53 MW (e) 65 MW

Answer (b) 38 MW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vdot=70 "m3/s"
z=65 "m"
g=9.81 "m/s2"
Eff=0.85
rho=1000 "kg/m3"
We=rho*Vdot*g*z*Eff/10^6 "MW"

"Some Wrong Solutions with Common Mistakes:"
W1_We=rho*Vdot*z*Eff/10^6 "Not using g"
W2_We=rho*Vdot*g*z/Eff/10^6 "Dividing by efficiency"
W3_We=rho*Vdot*g*z/10^6 "Not using efficiency"
```

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2-136 A 75 hp (shaft) compressor in a facility that operates at full load for 2500 hours a year is powered by an electric motor that has an efficiency of 93 percent. If the unit cost of electricity is $0.06/kWh, the annual electricity cost of this compressor is

(a) $7802 (b) $9021 (c) $12,100 (d) $8389 (e) $10,460

Answer (b) $9021

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Wcomp=75 "hp"
Hours=2500 "h/year"
Eff=0.93
price=0.06 "$/kWh"
We=Wcomp*0.7457*Hours/Eff
Cost=We*price

"Some Wrong Solutions with Common Mistakes:"
W1_cost= Wcomp*0.7457*Hours*price*Eff "multiplying by efficiency"
W2_cost= Wcomp*Hours*price/Eff "not using conversion"
W3_cost= Wcomp*Hours*price*Eff "multiplying by efficiency and not using conversion"
W4_cost= Wcomp*0.7457*Hours*price "Not using efficiency"
```

2-137 Consider a refrigerator that consumes 320 W of electric power when it is running. If the refrigerator runs only one quarter of the time and the unit cost of electricity is $0.09/kWh, the electricity cost of this refrigerator per month (30 days) is

(a) $3.56 (b) $5.18 (c) $8.54 (d) $9.28 (e) $20.74

Answer (b) $5.18

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
We=0.320 "kW"
Hours=0.25*(24*30) "h/year"
price=0.09 "$/kWh"
Cost=We*hours*price

"Some Wrong Solutions with Common Mistakes:"
W1_cost= We*24*30*price "running continuously"
```

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2-138 A 2-kW pump is used to pump kerosene (ρ = 0.820 kg/L) from a tank on the ground to a tank at a higher elevation. Both tanks are open to the atmosphere, and the elevation difference between the free surfaces of the tanks is 30 m. The maximum volume flow rate of kerosene is

(a) 8.3 L/s (b) 7.2 L/s (c) 6.8 L/s (d) 12.1 L/s (e) 17.8 L/s

Answer (a) 8.3 L/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W=2 "kW"
rho=0.820 "kg/L"
z=30 "m"
g=9.81 "m/s2"
W=rho*Vdot*g*z/1000

"Some Wrong Solutions with Common Mistakes:"
W=W1_Vdot*g*z/1000 "Not using density"
```

2-139 A glycerin pump is powered by a 5-kW electric motor. The pressure differential between the outlet and the inlet of the pump at full load is measured to be 211 kPa. If the flow rate through the pump is 18 L/s and the changes in elevation and the flow velocity across the pump are negligible, the overall efficiency of the pump is

(a) 69% (b) 72% (c) 76% (d) 79% (e) 82%

Answer (c) 76%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
We=5 "kW"
Vdot= 0.018 "m3/s"
DP=211 "kPa"
Emech=Vdot*DP
Emech=Eff*We
```

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The following problems are based on the optional special topic of heat transfer

2-140 A 10-cm high and 20-cm wide circuit board houses on its surface 100 closely spaced chips, each generating heat at a rate of 0.08 W and transferring it by convection to the surrounding air at 25°C. Heat transfer from the back surface of the board is negligible. If the convection heat transfer coefficient on the surface of the board is 10 W/m^2.°C and radiation heat transfer is negligible, the average surface temperature of the chips is

(a) 26°C (b) 45°C (c) 15°C (d) 80°C (e) 65°C

Answer (e) 65°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
A=0.10*0.20 "m^2"
Q= 100*0.08 "W"
Tair=25 "C"
h=10 "W/m^2.C"
Q= h*A*(Ts-Tair) "W"

"Some Wrong Solutions with Common Mistakes:"
Q= h*(W1_Ts-Tair) "Not using area"
Q= h*2*A*(W2_Ts-Tair) "Using both sides of surfaces"
Q= h*A*(W3_Ts+Tair) "Adding temperatures instead of subtracting"
Q/100= h*A*(W4_Ts-Tair) "Considering 1 chip only"
```

2-141 A 50-cm-long, 0.2-cm-diameter electric resistance wire submerged in water is used to determine the boiling heat transfer coefficient in water at 1 atm experimentally. The surface temperature of the wire is measured to be 130°C when a wattmeter indicates the electric power consumption to be 4.1 kW. Then the heat transfer coefficient is

(a) 43,500 W/m^2.°C (b) 137 W/m^2.°C (c) 68,330 W/m^2.°C (d) 10,038 W/m^2.°C

(e) 37,540 W/m^2.°C

Answer (a) 43,500 W/m^2.°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
L=0.5 "m"
D=0.002 "m"
A=pi*D*L "m^2"
We=4.1 "kW"
Ts=130 "C"
Tf=100 "C    (Boiling temperature of water at 1 atm)"
We= h*A*(Ts-Tf) "W"

"Some Wrong Solutions with Common Mistakes:"
We= W1_h*(Ts-Tf) "Not using area"
We= W2_h*(L*pi*D^2/4)*(Ts-Tf) "Using volume instead of area"
We= W3_h*A*Ts "Using Ts instead of temp difference"
```

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2-142 A 3-m^2 hot black surface at 80°C is losing heat to the surrounding air at 25°C by convection with a convection heat transfer coefficient of 12 W/m^2.°C, and by radiation to the surrounding surfaces at 15°C. The total rate of heat loss from the surface is

(a) 1987 W (b) 2239 W (c) 2348 W (d) 3451 W (e) 3811 W

Answer (d) 3451 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
sigma=5.67E-8 "W/m^2.K^4"
eps=1
A=3 "m^2"
h_conv=12 "W/m^2.C"
Ts=80 "C"
Tf=25 "C"
Tsurr=15 "C"
Q_conv=h_conv*A*(Ts-Tf)  "W"
Q_rad=eps*sigma*A*((Ts+273)^4-(Tsurr+273)^4) "W"
Q_total=Q_conv+Q_rad "W"

"Some Wrong Solutions with Common Mistakes:"
W1_Ql=Q_conv "Ignoring radiation"
W2_Q=Q_rad "ignoring convection"
W3_Q=Q_conv+eps*sigma*A*(Ts^4-Tsurr^4) "Using C in radiation calculations"
W4_Q=Q_total/A "not using area"
```

2-143 Heat is transferred steadily through a 0.2-m thick 8 m by 4 m wall at a rate of 2.4 kW. The inner and outer surface temperatures of the wall are measured to be 15°C to 5°C. The average thermal conductivity of the wall is

(a) 0.002 W/m.°C (b) 0.75 W/m.°C (c) 1.0 W/m.°C (d) 1.5 W/m.°C (e) 3.0 W/m.°C

Answer (d) 1.5 W/m.°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
A=8*4 "m^2"
L=0.2 "m"
T1=15 "C"
T2=5 "C"
Q=2400  "W"
Q=k*A*(T1-T2)/L "W"

"Some Wrong Solutions with Common Mistakes:"
Q=W1_k*(T1-T2)/L "Not using area"
Q=W2_k*2*A*(T1-T2)/L  "Using areas of both surfaces"
Q=W3_k*A*(T1+T2)/L  "Adding temperatures instead of subtracting"
Q=W4_k*A*L*(T1-T2) "Multiplying by thickness instead of dividing by it"
```

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2-144 The roof of an electrically heated house is 7 m long, 10 m wide, and 0.25 m thick. It is made of a flat layer of concrete whose thermal conductivity is 0.92 W/m.°C. During a certain winter night, the temperatures of the inner and outer surfaces of the roof are measured to be 15°C and 4°C, respectively. The average rate of heat loss through the roof that night was

(a) 41 W (b) 177 W (c) 4894 W (d) 5567 W (e) 2834 W

Answer (e) 2834 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
A=7*10 "m^2"
L=0.25 "m"
k=0.92 "W/m.C"
T1=15 "C"
T2=4 "C"
Q_cond=k*A*(T1-T2)/L "W"

"Some Wrong Solutions with Common Mistakes:"
W1_Q=k*(T1-T2)/L "Not using area"
W2_Q=k*2*A*(T1-T2)/L  "Using areas of both surfaces"
W3_Q=k*A*(T1+T2)/L  "Adding temperatures instead of subtracting"
W4_Q=k*A*L*(T1-T2) "Multiplying by thickness instead of dividing by it"
```

2-145 … 2-151 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 3
PROPERTIES OF PURE SUBSTANCES

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Pure Substances, Phase Change Processes, Property Diagrams

3-1C A liquid that is about to vaporize is saturated liquid; otherwise it is compressed liquid.

3-2C A vapor that is about to condense is saturated vapor; otherwise it is superheated vapor.

3-3C No.

3-4C The temperature will also increase since the boiling or saturation temperature of a pure substance depends on pressure.

3-5C Because one cannot be varied while holding the other constant. In other words, when one changes, so does the other one.

3-6C At critical point the saturated liquid and the saturated vapor states are identical. At triple point the three phases of a pure substance coexist in equilibrium.

3-7C Yes.

3-8C Case (c) when the pan is covered with a heavy lid. Because the heavier the lid, the greater the pressure in the pan, and thus the greater the cooking temperature.

3-9C At supercritical pressures, there is no distinct phase change process. The liquid uniformly and gradually expands into a vapor. At subcritical pressures, there is always a distinct surface between the phases.

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Property Tables

3-10C A perfectly fitting pot and its lid often stick after cooking as a result of the vacuum created inside as the temperature and thus the corresponding saturation pressure inside the pan drops. An easy way of removing the lid is to reheat the food. When the temperature rises to boiling level, the pressure rises to atmospheric value and thus the lid will come right off.

3-11C The molar mass of gasoline (C_8H_{18}) is 114 kg/kmol, which is much larger than the molar mass of air that is 29 kg/kmol. Therefore, the gasoline vapor will settle down instead of rising even if it is at a much higher temperature than the surrounding air. As a result, the warm mixture of air and gasoline on top of an open gasoline will most likely settle down instead of rising in a cooler environment

3-12C Yes. Otherwise we can create energy by alternately vaporizing and condensing a substance.

3-13C No. Because in the thermodynamic analysis we deal with the changes in properties; and the changes are independent of the selected reference state.

3-14C The term h_{fg} represents the amount of energy needed to vaporize a unit mass of saturated liquid at a specified temperature or pressure. It can be determined from $h_{fg} = h_g - h_f$.

3-15C Yes. It decreases with increasing pressure and becomes zero at the critical pressure.

3-16C Yes; the higher the temperature the lower the h_{fg} value.

3-17C Quality is the fraction of vapor in a saturated liquid-vapor mixture. It has no meaning in the superheated vapor region.

3-18C Completely vaporizing 1 kg of saturated liquid at 1 atm pressure since the higher the pressure, the lower the h_{fg} .

3-19C No. Quality is a mass ratio, and it is not identical to the volume ratio.

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3-20C The compressed liquid can be approximated as a saturated liquid at the given temperature. Thus $v_{T,P} \cong v_{f@T}$.

3-21C Ice can be made by evacuating the air in a water tank. During evacuation, vapor is also thrown out, and thus the vapor pressure in the tank drops, causing a difference between the vapor pressures at the water surface and in the tank. This pressure difference is the driving force of vaporization, and forces the liquid to evaporate. But the liquid must absorb the heat of vaporization before it can vaporize, and it absorbs it from the liquid and the air in the neighborhood, causing the temperature in the tank to drop. The process continues until water starts freezing. The process can be made more efficient by insulating the tank well so that the entire heat of vaporization comes essentially from the water.

3-22 *Complete the following table for H_2O*:

***T*, °C**	***P*, kPa**	***v*, m^3/kg**	**Phase description**
50	***12.35***	7.72	***Saturated mixture***
143.6	400	***0.4624***	Saturated vapor
250	500	***0.4744***	***Superheated vapor***
110	350	***0.001051***	***Compressed liquid***

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3-23 Problem 3-22 is reconsidered. The missing properties of water are to be determined using EES, and the solution is to be repeated for refrigerant-134a, refrigerant-22, and ammonia.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
T[1]=50 [C]
v[1]=7.72 [m^3/kg]
P[2]=400 [kPa]
x[2]=1
T[3]=250 [C]
P[3]=500 [kPa]
T[4]=110 [C]
P[4]=350 [kPa]

"Analysis"
Fluid$='steam_iapws' "Change the Fluid to R134a, R22 and Ammonia and solve"
P[1]=pressure(Fluid$, T=T[1], v=v[1])
x[1]=quality(Fluid$, T=T[1], v=v[1])
T[2]=temperature(Fluid$, P=P[2], x=x[2])
v[2]=volume(Fluid$, P=P[2], x=x[2])
v[3]=volume(Fluid$, P=P[3], T=T[3])
x[3]=quality(Fluid$, P=P[3], T=T[3])
v[4]=volume(Fluid$, P=P[4], T=T[4])
x[4]=quality(Fluid$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"
```

SOLUTION for water

T [C]	P [kPa]	x	v [kg/m^3]
50.00	12.35	0.6419	7.72
143.61	400.00	1	0.4624
250.00	500.00	100	0.4744
110.00	350.00	-100	0.001051

SOLUTION for R-134a

T [C]	P [kPa]	x	v [kg/m^3]
50.00	3.41	100	7.72
8.91	400.00	1	0.0512
250.00	500.00	-	-
110.00	350.00	100	0.08666

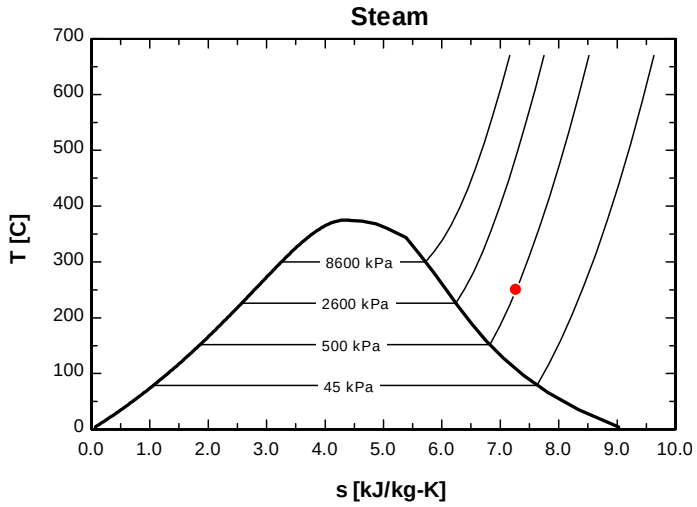
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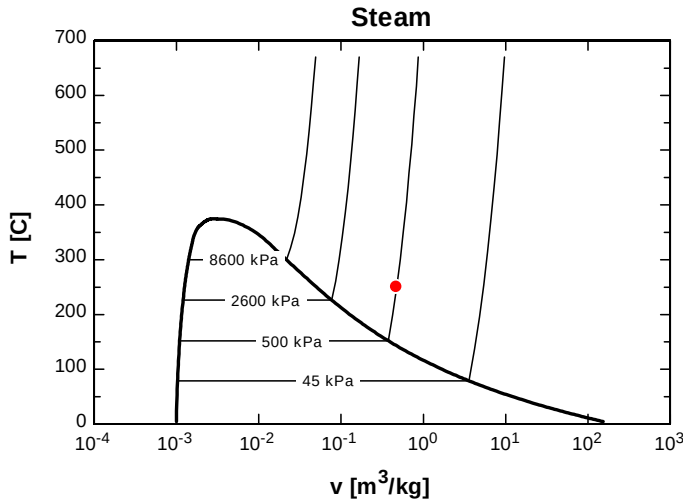
SOLUTION for R-22

T [C]	P [kPa]	X	v [kg/m^3]
50.00	4.02	100	7.72
-6.56	400.00	1	0.05817
250.00	500.00	100	0.09959
110.00	350.00	100	0.103

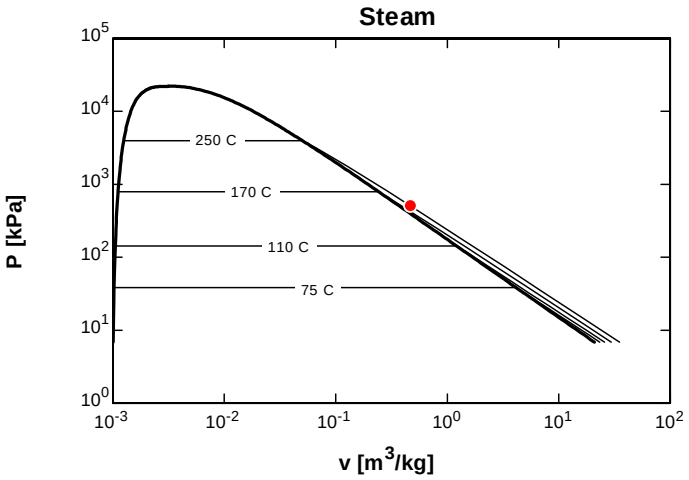
SOLUTION for Ammonia

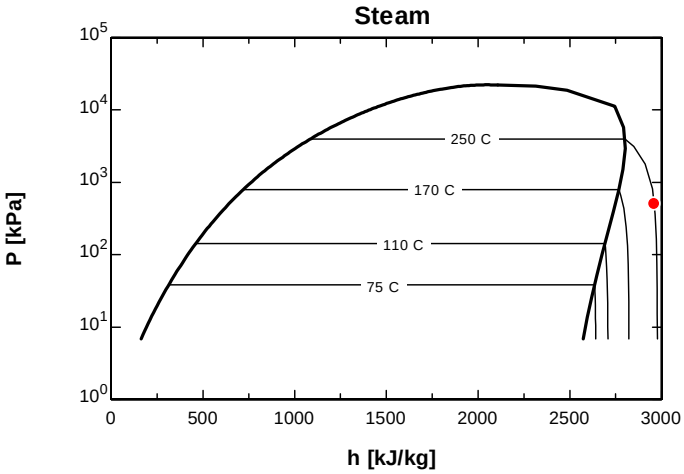
T [C]	P [kPa]	X	v [kg/m^3]
50.00	20.40	100	7.72
-1.89	400.00	1	0.3094
250.00	500.00	100	0.5076
110.00	350.00	100	0.5269

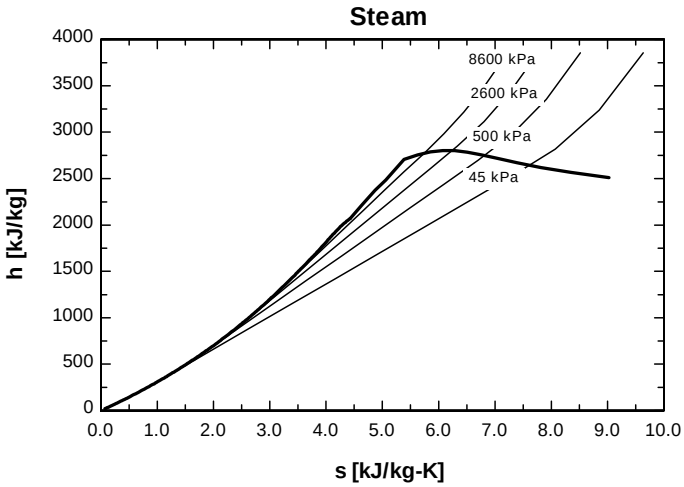




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3-24E *Complete the following table for H_2O:*

T, °F	P, psia	*u*, Btu / lbm	Phase description
300	***67.03***	782	***Saturated mixture***
267.22	40	***236.02***	Saturated liquid
500	120	***1174.4***	***Superheated vapor***
400	400	***373.84***	***Compressed liquid***

3-25E Problem 3-24E is reconsidered. The missing properties of water are to be determined using EES, and the solution is to be repeated for refrigerant-134a, refrigerant-22, and ammonia.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
T[1]=300 [F]
u[1]=782 [Btu/lbm]
P[2]=40 [psia]
x[2]=0
T[3]=500 [F]
P[3]=120 [psia]
T[4]=400 [F]
P[4]=420 [psia]

"Analysis"
Fluid$='steam_iapws'
P[1]=pressure(Fluid$, T=T[1], u=u[1])
x[1]=quality(Fluid$, T=T[1], u=u[1])
T[2]=temperature(Fluid$, P=P[2], x=x[2])
u[2]=intenergy(Fluid$, P=P[2], x=x[2])
u[3]=intenergy(Fluid$, P=P[3], T=T[3])
x[3]=quality(Fluid$, P=P[3], T=T[3])
u[4]=intenergy(Fluid$, P=P[4], T=T[4])
x[4]=quality(Fluid$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"
```

Solution for steam

T, ºF	P, psia	x	u, Btu/lbm
300	67.028	0.6173	782
267.2	40	0	236
500	120	100	1174
400	400	-100	373.8

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3-26 *Complete the following table for H_2O*:

T, °C	P, kPa	*h*, kJ / kg	*x*	Phase description
120.21	200	***2045.8***	0.7	***Saturated mixture***
140	***361.53***	1800	***0.565***	***Saturated mixture***
177.66	950	***752.74***	0.0	***Saturated liquid***
80	500	***335.37***	- - -	***Compressed liquid***
350.0	800	3162.2	- - -	***Superheated vapor***

3-27 *Complete the following table for Refrigerant-134a*:

T, °C	P, kPa	v, m^3 / kg	Phase description
-12	320	***0.000750***	***Compressed liquid***
30	***770.64***	0.0065	***Saturated mixture***
18.73	550	***0.03741***	Saturated vapor
60	600	***0.04139***	***Superheated vapor***

3-28 *Complete the following table for water*:

P, kPa	*T*, °C	v, m^3/kg	*h*, kJ/kg	Condition description and quality, if applicable
200	**120.2**	0.8858	2706.3	***x* = 1, Saturated vapor**
270.3	130		1959.3	***x* = 0.650, Two-phase mixture**
201.8	400	1.5358	3277.0	**Superheated vapor**
800	30	**0.001004**[*]	**125.74**[*]	**Compressed liquid**
450	147.90	-	-	**Insufficient information**

[*] Approximated as saturated liquid at the given temperature of 30°C

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3-29E *Complete the following table for Refrigerant-134a*:

T, °F	P, psia	*h*, Btu / lbm	*x*	Phase description
65.89	80	78	***0.566***	***Saturated mixture***
15	***29.759***	***69.92***	0.6	***Saturated mixture***
10	70	***15.35***	- - -	***Compressed liquid***
160	180	129.46	- - -	***Superheated vapor***
110	***161.16***	***117.23***	1.0	***Saturated vapor***

3-30 A piston-cylinder device contains R-134a at a specified state. Heat is transferred to R-134a. The final pressure, the volume change of the cylinder, and the enthalpy change are to be determined.

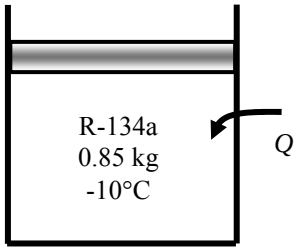
Analysis (a) The final pressure is equal to the initial pressure, which is determined from

$$P_2 = P_1 = P_{\text{atm}} + \frac{m_p g}{\pi D^2/4} = 88\,\text{kPa} + \frac{(12\,\text{kg})(9.81\,\text{m/s}^2)}{\pi(0.25\,\text{m})^2/4}\left(\frac{1\,\text{kN}}{1000\,\text{kg.m/s}^2}\right) = \mathbf{90.4\,kPa}$$

(b) The specific volume and enthalpy of R-134a at the initial state of 90.4 kPa and -10°C and at the final state of 90.4 kPa and 15°C are (from EES)

$v_1 = 0.2302\ \text{m}^3/\text{kg}$ $\quad h_1 = 247.76\ \text{kJ/kg}$

$v_2 = 0.2544\ \text{m}^3/\text{kg}$ $\quad h_2 = 268.16\ \text{kJ/kg}$



The initial and the final volumes and the volume change are

$$V_1 = mv_1 = (0.85\,\text{kg})(0.2302\,\text{m}^3/\text{kg}) = 0.1957\,\text{m}^3$$

$$V_2 = mv_2 = (0.85\,\text{kg})(0.2544\,\text{m}^3/\text{kg}) = 0.2162\,\text{m}^3$$

$$\Delta V = 0.2162 - 0.1957 = \mathbf{0.0205\,m^3}$$

(c) The total enthalpy change is determined from

$$\Delta H = m(h_2 - h_1) = (0.85\,\text{kg})(268.16 - 247.76)\,\text{kJ/kg} = \mathbf{17.4\,kJ/kg}$$

3-31E The temperature of R-134a at a specified state is to be determined.

Analysis Since the specified specific volume is higher than v_g for 120 psia, this is a superheated vapor state. From R-134a tables,

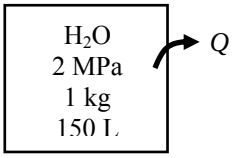
$$\left.\begin{array}{l} P = 120\ \text{psia} \\ v = 0.4619\ \text{ft}^3/\text{lbm} \end{array}\right\} T = \mathbf{140°F}\ \text{(Table A-13E)}$$

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3-32 A rigid container that is filled with water is cooled. The initial temperature and the final pressure are to be determined.

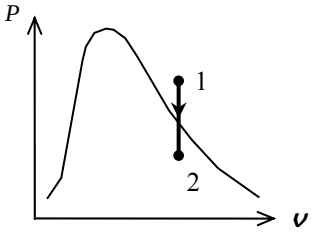
Analysis This is a constant volume process. The specific volume is

$$v_1 = v_2 = \frac{V}{m} = \frac{0.150\ \text{m}^3}{1\ \text{kg}} = 0.150\ \text{m}^3/\text{kg}$$



The initial state is superheated vapor. The temperature is determined to be

$$\left.\begin{array}{l} P_1 = 2\ \text{MPa} \\ v_1 = 0.150\ \text{m}^3/\text{kg} \end{array}\right\} \ T_1 = \mathbf{395°C} \quad \text{(Table A - 6)}$$



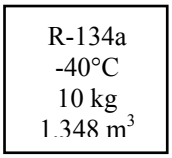
This is a constant volume cooling process ($v = V/m$ = constant). The final state is saturated mixture and thus the pressure is the saturation pressure at the final temperature:

$$\left.\begin{array}{l} T_2 = 40°\text{C} \\ v_2 = v_1 = 0.150\ \text{m}^3/\text{kg} \end{array}\right\} \ P_2 = P_{\text{sat @ 40°C}} = \mathbf{7.385\ kPa} \quad \text{(Table A - 4)}$$

3-33 A rigid container that is filled with R-134a is heated. The final temperature and initial pressure are to be determined.

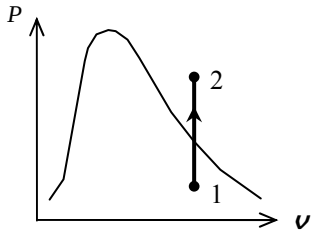
Analysis This is a constant volume process. The specific volume is

$$v_1 = v_2 = \frac{V}{m} = \frac{1.348\ \text{m}^3}{10\ \text{kg}} = 0.1348\ \text{m}^3/\text{kg}$$



The initial state is determined to be a mixture, and thus the pressure is the saturation pressure at the given temperature

$$P_1 = P_{\text{sat @ -40°C}} = \mathbf{51.25\ kPa} \quad \text{(Table A -11)}$$

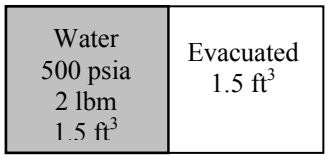


The final state is superheated vapor and the temperature is determined by interpolation to be

$$\left.\begin{array}{l} P_2 = 200\ \text{kPa} \\ v_2 = 0.1348\ \text{m}^3/\text{kg} \end{array}\right\} \ T_2 = \mathbf{66.3°C} \quad \text{(Table A -13)}$$

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3-34E Left chamber of a partitioned system contains water at a specified state while the right chamber is evacuated. The partition is now ruptured and heat is transferred to the water. The pressure and the total internal energy at the final state are to be determined.



Analysis The final specific volume is

$$v_2 = \frac{V_2}{m} = \frac{3\ \text{ft}^3}{2\ \text{lbm}} = 1.5\ \text{ft}^3/\text{lbm}$$

At this specific volume and the final temperature, the state is a saturated mixture, and the pressure is the saturation pressure

$$P_2 = P_{\text{sat}@300°\text{F}} = \mathbf{67.03\ psia} \quad \text{(Table A - 4E)}$$

The quality and internal energy at the final state are

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{(1.5 - 0.01745)\ \text{ft}^3/\text{lbm}}{(6.4663 - 0.01745)\ \text{ft}^3/\text{lbm}} = 0.2299$$
$$u_2 = u_f + x_2 u_{fg} = 269.51 + (0.2299)(830.25) = 460.38\ \text{Btu/lbm}$$

The total internal energy is then

$$U_2 = mu_2 = (2\ \text{lbm})(460.38\ \text{Btu/lbm}) = \mathbf{920.8\ Btu}$$

3-35 The enthalpy of R-134a at a specified state is to be determined.

Analysis The specific volume is

$$v = \frac{V}{m} = \frac{9\ \text{m}^3}{300\ \text{kg}} = 0.03\ \text{m}^3/\text{kg}$$

Inspection of Table A-11 indicates that this is a mixture of liquid and vapor. Using the properties at 10°C line, the quality and the enthalpy are determined to be

$$x = \frac{v - v_f}{v_{fg}} = \frac{(0.03 - 0.0007930)\ \text{m}^3/\text{kg}}{(0.049403 - 0.0007930)\ \text{m}^3/\text{kg}} = 0.6008$$

$$h = h_f + xh_{fg} = 65.43 + (0.6008)(190.73) = \mathbf{180.02\ kJ/kg}$$

3-36 The specific volume of R-134a at a specified state is to be determined.

Analysis Since the given temperature is higher than the saturation temperature for 200 kPa, this is a superheated vapor state. The specific volume is then

$$\left.\begin{array}{l} P = 200\ \text{kPa} \\ T = 25°\text{C} \end{array}\right\} \quad v = \mathbf{0.11646\ m^3/kg} \quad \text{(Table A - 13)}$$

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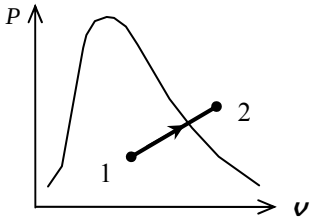
3-37E A spring-loaded piston-cylinder device is filled with R-134a. The water now undergoes a process until its volume increases by 40%. The final temperature and the enthalpy are to be determined.

Analysis From Table A-11E, the initial specific volume is

$$v_1 = v_f + x_1 v_{fg} = 0.01143 + (0.80)(4.4300 - 0.01143) = 3.5463 \text{ ft}^3/\text{lbm}$$

and the initial volume will be

$$V_1 = mv_1 = (0.2 \text{ lbm})(3.5463 \text{ ft}^3/\text{lbm}) = 0.7093 \text{ ft}^3$$



With a 40% increase in the volume, the final volume will be

$$V_2 = 1.4V_1 = 1.4(0.7093 \text{ ft}^3) = 0.9930 \text{ ft}^3$$

The distance that the piston moves between the initial and final conditions is

$$\Delta x = \frac{\Delta V}{A_p} = \frac{\Delta V}{\pi D^2 / 4} = \frac{4(0.9930 - 0.7093)\text{ft}^3}{\pi (1 \text{ ft})^2} = 0.3612 \text{ ft}$$

As a result of the compression of the spring, the pressure difference between the initial and final states is

$$\Delta P = \frac{\Delta F}{A_p} = \frac{k\Delta x}{A_p} = \frac{k\Delta x}{\pi D^2 / 4} = \frac{4(37 \text{ lbf/in})(0.3612 \times 12 \text{ in})}{\pi (12 \text{ in})^2} = 1.42 \text{ lbf/in}^2 = 1.42 \text{ psia}$$

The initial pressure is

$$P_1 = P_{\text{sat @ -30°F}} = 9.87 \text{ psia} \quad \text{(Table A-11E)}$$

The final pressure is then

$$P_2 = P_1 + \Delta P = 9.87 + 1.42 = 11.29 \text{ psia}$$

and the final specific volume is

$$v_2 = \frac{V_2}{m} = \frac{0.9930 \text{ ft}^3}{0.2 \text{ lbm}} = 4.965 \text{ ft}^3/\text{lbm}$$

At this final state, the temperature and enthalpy are

$$\left.\begin{array}{l} P_2 = 11.29 \text{ psia} \\ v_2 = 4.965 \text{ ft}^3/\text{lbm} \end{array}\right\} \begin{array}{l} T_1 = \mathbf{81.5°F} \\ h_1 = \mathbf{119.9 \text{ Btu/lbm}} \end{array} \quad \text{(from EES)}$$

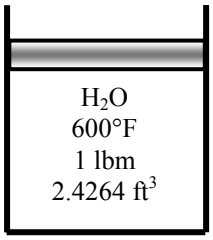
Note that it is very difficult to get the temperature and enthalpy from Table A-13E accurately.

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3-38E A piston-cylinder device that is filled with water is cooled. The final pressure and volume of the water are to be determined.

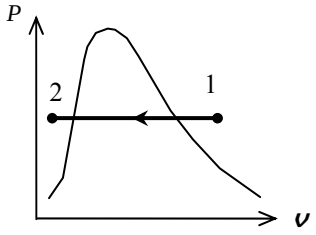
Analysis The initial specific volume is

$$v_1 = \frac{V_1}{m} = \frac{2.4264\ \text{ft}^3}{1\ \text{lbm}} = 2.4264\ \text{ft}^3/\text{lbm}$$



This is a constant-pressure process. The initial state is determined to be superheated vapor and thus the pressure is determined to be

$$\left.\begin{array}{l} T_1 = 600°\text{F} \\ v_1 = 2.4264\ \text{ft}^3/\text{lbm} \end{array}\right\} \quad P_1 = P_2 = \mathbf{250\ psia} \ \text{(Table A - 6E)}$$



The saturation temperature at 250 psia is 400.1°F. Since the final temperature is less than this temperature, the final state is compressed liquid. Using the incompressible liquid approximation,

$$v_2 \cong v_{f\,@\,200°\text{F}} = 0.01663\ \text{ft}^3/\text{lbm} \quad \text{(Table A - 4E)}$$

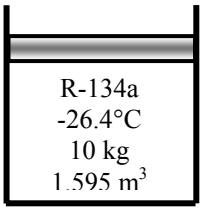
The final volume is then

$$V_2 = mv_2 = (1\ \text{lbm})(0.01663\ \text{ft}^3/\text{lbm}) = \mathbf{0.01663\ ft^3}$$

3-39 A piston-cylinder device that is filled with R-134a is heated. The final volume is to be determined.

Analysis This is a constant pressure process. The initial specific volume is

$$v_1 = \frac{V}{m} = \frac{1.595\ \text{m}^3}{10\ \text{kg}} = 0.1595\ \text{m}^3/\text{kg}$$

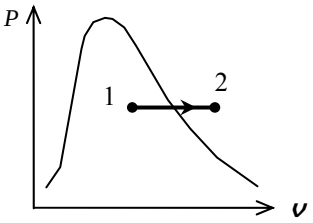


The initial state is determined to be a mixture, and thus the pressure is the saturation pressure at the given temperature

$$P_1 = P_{\text{sat}\,@\,-26.4°\text{C}} = 100\ \text{kPa} \ \text{(Table A - 12)}$$

The final state is superheated vapor and the specific volume is

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ T_2 = 100°\text{C} \end{array}\right\} \quad v_2 = 0.30138\ \text{m}^3/\text{kg} \ \text{(Table A - 13)}$$



The final volume is then

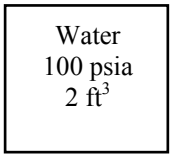
$$V_2 = mv_2 = (10\ \text{kg})(0.30138\ \text{m}^3/\text{kg}) = \mathbf{3.0138\ m^3}$$

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3-40E The total internal energy and enthalpy of water in a container are to be determined.

Analysis The specific volume is

$$v = \frac{V}{m} = \frac{2\ \text{ft}^3}{1\ \text{lbm}} = 2\ \text{ft}^3/\text{lbm}$$



At this specific volume and the given pressure, the state is a saturated mixture. The quality, internal energy, and enthalpy at this state are (Table A-5E)

$$x = \frac{v - v_f}{v_{fg}} = \frac{(2 - 0.01774)\ \text{ft}^3/\text{lbm}}{(4.4327 - 0.01774)\ \text{ft}^3/\text{lbm}} = 0.4490$$
$$u = u_f + xu_{fg} = 298.19 + (0.4490)(807.29) = 660.7\ \text{Btu/lbm}$$
$$h = h_f + xh_{fg} = 298.51 + (0.4490)(888.99) = 697.7\ \text{Btu/lbm}$$

The total internal energy and enthalpy are then

$$U = mu = (1\ \text{lbm})(660.7\ \text{Btu/lbm}) = \mathbf{660.7\ Btu}$$
$$H = mh = (1\ \text{lbm})(697.7\ \text{Btu/lbm}) = \mathbf{697.7\ Btu}$$

3-41 The volume of a container that contains water at a specified state is to be determined.

Analysis The specific volume is determined from steam tables by interpolation to be

$$\left.\begin{array}{l} P = 100\ \text{kPa} \\ T = 250°\text{C} \end{array}\right\} v = 2.4062\ \text{m}^3/\text{kg}\ \ \text{(Table A - 6)}$$



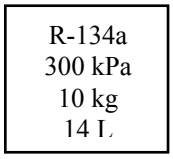
The volume of the container is then

$$V = mv = (3\ \text{kg})(2.4062\ \text{m}^3/\text{kg}) = \mathbf{7.22\ m^3}$$

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3-42 A rigid container that is filled with R-134a is heated. The temperature and total enthalpy are to be determined at the initial and final states.

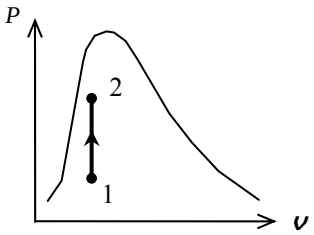
Analysis This is a constant volume process. The specific volume is



$$v_1 = v_2 = \frac{V}{m} = \frac{0.014\,\text{m}^3}{10\,\text{kg}} = 0.0014\,\text{m}^3/\text{kg}$$

The initial state is determined to be a mixture, and thus the temperature is the saturation temperature at the given pressure. From Table A-12 by interpolation

$$T_1 = T_{\text{sat}@300\,\text{kPa}} = \mathbf{0.61°C}$$



and

$$x_1 = \frac{v_1 - v_f}{v_{fg}} = \frac{(0.0014 - 0.0007736)\,\text{m}^3/\text{kg}}{(0.067978 - 0.0007736)\,\text{m}^3/\text{kg}} = 0.009321$$

$$h_1 = h_f + x_1 h_{fg} = 52.67 + (0.009321)(198.13) = 54.52\,\text{kJ/kg}$$

The total enthalpy is then

$$H_1 = mh_1 = (10\,\text{kg})(54.52\,\text{kJ/kg}) = \mathbf{545.2\,kJ}$$

The final state is also saturated mixture. Repeating the calculations at this state,

$$T_2 = T_{\text{sat}@600\,\text{kPa}} = \mathbf{21.55°C}$$

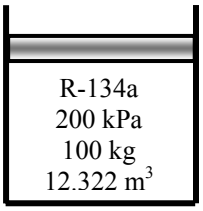
$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{(0.0014 - 0.0008199)\,\text{m}^3/\text{kg}}{(0.034295 - 0.0008199)\,\text{m}^3/\text{kg}} = 0.01733$$

$$h_2 = h_f + x_2 h_{fg} = 81.51 + (0.01733)(180.90) = 84.64\,\text{kJ/kg}$$

$$H_2 = mh_2 = (10\,\text{kg})(84.64\,\text{kJ/kg}) = \mathbf{846.4\,kJ}$$

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3-43 A piston-cylinder device that is filled with R-134a is cooled at constant pressure. The final temperature and the change of total internal energy are to be determined.

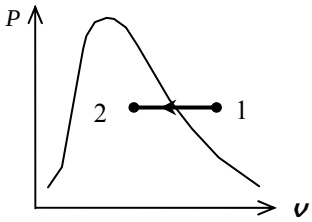


Analysis The initial specific volume is

$$v_1 = \frac{V}{m} = \frac{12.322\,\text{m}^3}{100\,\text{kg}} = 0.12322\,\text{m}^3/\text{kg}$$

The initial state is superheated and the internal energy at this state is

$$\left.\begin{array}{l} P_1 = 200\,\text{kPa} \\ v_1 = 0.12322\,\text{m}^3/\text{kg} \end{array}\right\} u_1 = 263.08\,\text{kJ/kg} \quad \text{(Table A-13)}$$



The final specific volume is

$$v_2 = \frac{v_1}{2} = \frac{0.12322\,\text{m}^3\ /\ kg}{2} = 0.06161\,\text{m}^3/\text{kg}$$

This is a constant pressure process. The final state is determined to be saturated mixture whose temperature is

$$T_2 = T_{\text{sat}\,@\,200\,\text{kPa}} = \mathbf{-10.09°C} \quad \text{(Table A-12)}$$

The internal energy at the final state is (Table A-12)

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{(0.06161 - 0.0007533)\,\text{m}^3/\text{kg}}{(0.099867 - 0.0007533)\,\text{m}^3/\text{kg}} = 0.6140$$

$$u_2 = u_f + x_2 u_{fg} = 38.28 + (0.6140)(186.21) = 152.61\,\text{kJ/kg}$$

Hence, the change in the internal energy is

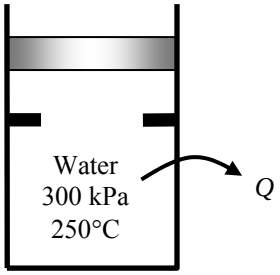
$$\Delta u = u_2 - u_1 = 152.61 - 263.08 = \mathbf{-110.47\ kJ/kg}$$

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3-44 A piston-cylinder device fitted with stops contains water at a specified state. Now the water is cooled until a final pressure. The process is to be indicated on the *T*-v diagram and the change in internal energy is to be determined.

Analysis The process is shown on T-v diagram. The internal energy at the initial state is

$$\left.\begin{array}{l} P_1 = 300\ \text{kPa} \\ T_1 = 250°\text{C} \end{array}\right\} \quad u_1 = 2728.9\ \text{kJ/kg} \quad \text{(Table A - 6)}$$

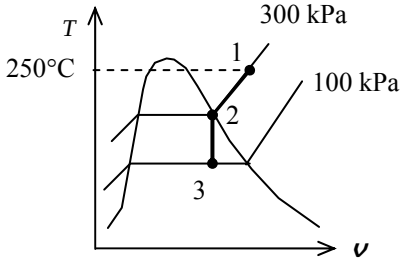


State 2 is saturated vapor at the initial pressure. Then,

$$\left.\begin{array}{l} P_2 = 300\ \text{kPa} \\ x_2 = 1\ (\text{sat. vapor}) \end{array}\right\} \quad v_2 = 0.6058\ \text{m}^3/\text{kg} \quad \text{(Table A - 5)}$$

Process 2-3 is a constant-volume process. Thus,

$$\left.\begin{array}{l} P_3 = 100\ \text{kPa} \\ v_3 = 0.6058\ \text{m}^3/\text{kg} \end{array}\right\} \quad u_3 = 1163.3\ \text{kJ/kg} \quad \text{(Table A - 5)}$$



The overall change in internal energy is

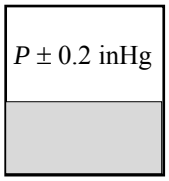
$$\Delta u = u_1 - u_3 = 2728.9 - 1163.3 = \mathbf{1566\ kJ/kg}$$

3-45E The local atmospheric pressure, and thus the boiling temperature, changes with the weather conditions. The change in the boiling temperature corresponding to a change of 0.2 in of mercury in atmospheric pressure is to be determined.

Properties The saturation pressures of water at 200 and 212°F are 11.538 and 14.709 psia, respectively (Table A-4E). One in. of mercury is equivalent to 1 inHg = 3.387 kPa = 0.491 psia (inner cover page).

Analysis A change of 0.2 in of mercury in atmospheric pressure corresponds to

$$\Delta P = (0.2\ \text{inHg})\left(\frac{0.491\ \text{psia}}{1\ \text{inHg}}\right) = 0.0982\ \text{psia}$$



At about boiling temperature, the change in boiling temperature per 1 psia change in pressure is determined using data at 200 and 212°F to be

$$\frac{\Delta T}{\Delta P} = \frac{(212 - 200)°\text{F}}{(14.709 - 11.538)\ \text{psia}} = 3.783\ °\text{F/psia}$$

Then the change in saturation (boiling) temperature corresponding to a change of 0.147 psia becomes

$$\Delta T_{\text{boiling}} = (3.783\ °\text{F/psia})\Delta P = (3.783\ °\text{F/psia})(0.0982\ \text{psia}) = \mathbf{0.37°F}$$

which is very small. Therefore, the effect of variation of atmospheric pressure on the boiling temperature is negligible.

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3-46 A person cooks a meal in a pot that is covered with a well-fitting lid, and leaves the food to cool to the room temperature. It is to be determined if the lid will open or the pan will move up together with the lid when the person attempts to open the pan by lifting the lid up.

Assumptions **1** The local atmospheric pressure is 1 atm = 101.325 kPa. **2** The weight of the lid is small and thus its effect on the boiling pressure and temperature is negligible. **3** No air has leaked into the pan during cooling.

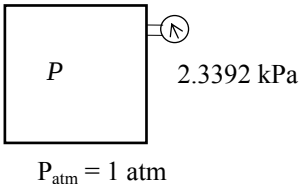
Properties The saturation pressure of water at 20°C is 2.3392 kPa (Table A-4).

Analysis Noting that the weight of the lid is negligible, the reaction force F on the lid after cooling at the pan-lid interface can be determined from a force balance on the lid in the vertical direction to be

$$PA + F = P_{atm}A$$

or,

$$F = A(P_{atm} - P) = (\pi D^2/4)(P_{atm} - P)$$
$$= \frac{\pi(0.3\text{ m})^2}{4}(101{,}325 - 2339.2)\text{ Pa}$$
$$= 6997\text{ m}^2\text{Pa} = \mathbf{6997\ N} \quad (\text{since } 1\text{ Pa} = 1\text{ N/m}^2)$$



The weight of the pan and its contents is

$$W = mg = (8\text{ kg})(9.81\text{ m/s}^2) = \mathbf{78.5\ N}$$

which is much less than the reaction force of 6997 N at the pan-lid interface. Therefore, the pan will **move up** together with the lid when the person attempts to open the pan by lifting the lid up. In fact, it looks like the lid will not open even if the mass of the pan and its contents is several hundred kg.

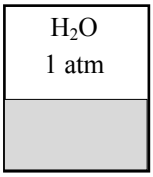
3-47 Water is boiled at 1 atm pressure in a pan placed on an electric burner. The water level drops by 10 cm in 45 min during boiling. The rate of heat transfer to the water is to be determined.

Properties The properties of water at 1 atm and thus at a saturation temperature of T_{sat} = 100°C are h_{fg} = 2256.5 kJ/kg and v_f = 0.001043 m^3/kg (Table A-4).

Analysis The rate of evaporation of water is

$$m_{evap} = \frac{V_{evap}}{v_f} = \frac{(\pi D^2/4)L}{v_f} = \frac{[\pi(0.25\text{ m})^2/4](0.10\text{ m})}{0.001043} = 4.704\text{ kg}$$

$$\dot{m}_{evap} = \frac{m_{evap}}{\Delta t} = \frac{4.704\text{ kg}}{45\times 60\text{ s}} = 0.001742\text{ kg/s}$$



Then the rate of heat transfer to water becomes

$$\dot{Q} = \dot{m}_{evap}h_{fg} = (0.001742\text{ kg/s})(2256.5\text{ kJ/kg}) = \mathbf{3.93\ kW}$$

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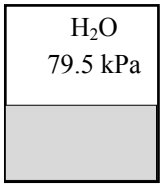
3-48 Water is boiled at a location where the atmospheric pressure is 79.5 kPa in a pan placed on an electric burner. The water level drops by 10 cm in 45 min during boiling. The rate of heat transfer to the water is to be determined.

Properties The properties of water at 79.5 kPa are T_{sat} = 93.3°C, h_{fg} = 2273.9 kJ/kg and v_f = 0.001038 m^3/kg (Table A-5).

Analysis The rate of evaporation of water is

$$m_{evap} = \frac{V_{evap}}{v_f} = \frac{(\pi D^2/4)L}{v_f} = \frac{[\pi(0.25\,\text{m})^2/4](0.10\,\text{m})}{0.001038} = 4.727\,\text{kg}$$

$$\dot{m}_{evap} = \frac{m_{evap}}{\Delta t} = \frac{4.727\,\text{kg}}{45\times 60\,\text{s}} = 0.001751\,\text{kg/s}$$



Then the rate of heat transfer to water becomes

$$\dot{Q} = \dot{m}_{evap} h_{fg} = (0.001751\,\text{kg/s})(2273.9\,\text{kJ/kg}) = \mathbf{3.98\,kW}$$

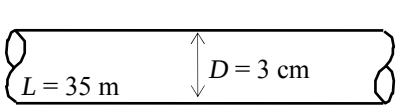
3-49 Saturated steam at T_{sat} = 40°C condenses on the outer surface of a cooling tube at a rate of 130 kg/h. The rate of heat transfer from the steam to the cooling water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The condensate leaves the condenser as a saturated liquid at 30°C.

Properties The properties of water at the saturation temperature of 40°C are h_{fg} = 2406.0 kJ/kg (Table A-4).

Analysis Noting that 2406.0 kJ of heat is released as 1 kg of saturated vapor at 40°C condenses, the rate of heat transfer from the steam to the cooling water in the tube is determined directly from

$$\dot{Q} = \dot{m}_{evap} h_{fg}$$
$$= (130\,\text{kg/h})(2406.0\,\text{kJ/kg}) = 312{,}780\,\text{kJ/h}$$
$$= \mathbf{86.9\,kW}$$

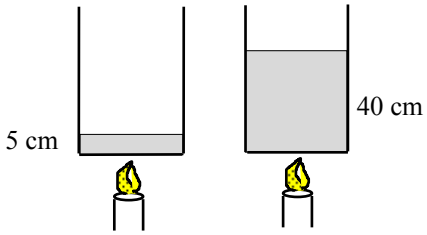


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3-50 The boiling temperature of water in a 5-cm deep pan is given. The boiling temperature in a 40-cm deep pan is to be determined.

Assumptions Both pans are full of water.

Properties The density of liquid water is approximately $\rho = 1000$ kg/m^3.



Analysis The pressure at the bottom of the 5-cm pan is the saturation pressure corresponding to the boiling temperature of 98°C:

$$P = P_{\text{sat@98°C}} = 94.39 \text{ kPa} \qquad \text{(Table A-4)}$$

The pressure difference between the bottoms of two pans is

$$\Delta P = \rho g h = (1000 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.35 \text{ m})\left(\frac{1 \text{ kPa}}{1000 \text{ kg/m} \cdot \text{s}^2}\right) = 3.43 \text{ kPa}$$

Then the pressure at the bottom of the 40-cm deep pan is

$$P = 94.39 + 3.43 = 97.82 \text{ kPa}$$

Then the boiling temperature becomes

$$T_{\text{boiling}} = T_{\text{sat@97.82 kPa}} = \mathbf{99.0°C} \qquad \text{(Table A-5)}$$

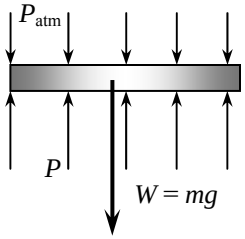
3-51 A vertical piston-cylinder device is filled with water and covered with a 20-kg piston that serves as the lid. The boiling temperature of water is to be determined.

Analysis The pressure in the cylinder is determined from a force balance on the piston,

$$PA = P_{\text{atm}}A + W$$

or,

$$\begin{aligned} P &= P_{\text{atm}} + \frac{mg}{A} \\ &= (100 \text{ kPa}) + \frac{(20 \text{ kg})(9.81 \text{ m/s}^2)}{0.01 \text{ m}^2}\left(\frac{1 \text{ kPa}}{1000 \text{ kg/m} \cdot \text{s}^2}\right) \\ &= 119.61 \text{ kPa} \end{aligned}$$



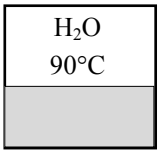
The boiling temperature is the saturation temperature corresponding to this pressure,

$$T = T_{\text{sat@119.61 kPa}} = \mathbf{104.7°C} \qquad \text{(Table A-5)}$$

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3-52 A rigid tank that is filled with saturated liquid-vapor mixture is heated. The temperature at which the liquid in the tank is completely vaporized is to be determined, and the T-v diagram is to be drawn.

Analysis This is a constant volume process ($v = V/m$ = constant), and the specific volume is determined to be



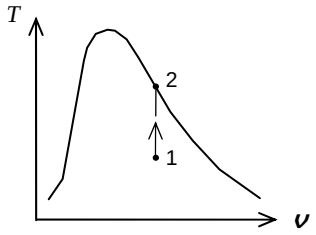
$$v = \frac{V}{m} = \frac{1.8\ \text{m}^3}{15\ \text{kg}} = 0.12\ \text{m}^3/\text{kg}$$

When the liquid is completely vaporized the tank will contain saturated vapor only. Thus,

$$v_2 = v_g = 0.12\ \text{m}^3/\text{kg}$$

The temperature at this point is the temperature that corresponds to this v_g value,

$$T = T_{\text{sat@}v_g=0.12\ \text{m}^3/\text{kg}} = \mathbf{202.9°C} \qquad \text{(Table A-4)}$$



3-53 A piston-cylinder device contains a saturated liquid-vapor mixture of water at 800 kPa pressure. The mixture is heated at constant pressure until the temperature rises to 200°C. The initial temperature, the total mass of water, the final volume are to be determined, and the P-v diagram is to be drawn.

Analysis (a) Initially two phases coexist in equilibrium, thus we have a saturated liquid-vapor mixture. Then the temperature in the tank must be the saturation temperature at the specified pressure,

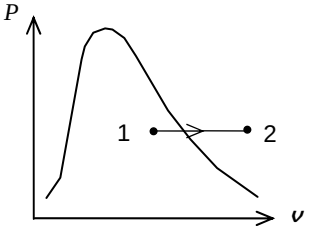
$$T = T_{\text{sat@600 kPa}} = \mathbf{158.8°C}$$

(b) The total mass in this case can easily be determined by adding the mass of each phase,

$$m_f = \frac{V_f}{v_f} = \frac{0.005\ \text{m}^3}{0.001101\ \text{m}^3/\text{kg}} = 4.543\ \text{kg}$$

$$m_g = \frac{V_g}{v_g} = \frac{0.9\ \text{m}^3}{0.3156\ \text{m}^3/\text{kg}} = 2.852\ \text{kg}$$

$$m_t = m_f + m_g = 4.543 + 2.852 = \mathbf{7.395\ kg}$$



(c) At the final state water is superheated vapor, and its specific volume is

$$\left.\begin{array}{l} P_2 = 600\ \text{kPa} \\ T_2 = 200^\circ\text{C} \end{array}\right\} \; v_2 = 0.3521\ \text{m}^3/\text{kg} \quad \text{(Table A-6)}$$

Then,

$$V_2 = m_t v_2 = (7.395\ \text{kg})(0.3521\ \text{m}^3/\text{kg}) = \mathbf{2.604\ m^3}$$

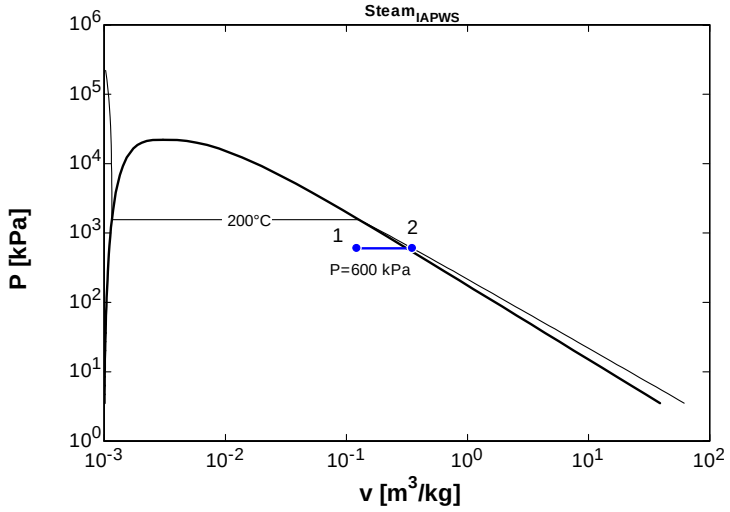
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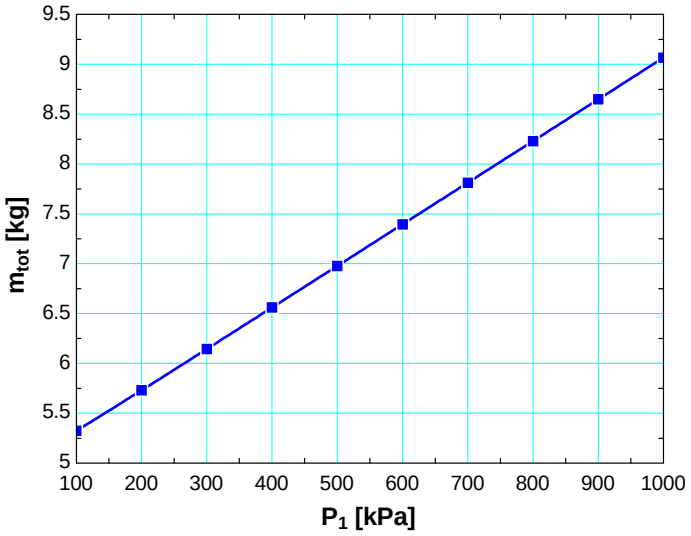
3-54 Problem 3-53 is reconsidered. The effect of pressure on the total mass of water in the tank as the pressure varies from 0.1 MPa to 1 MPa is to be investigated. The total mass of water is to be plotted against pressure, and results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
P[1]=600 [kPa]
P[2]=P[1]
T[2]=200 [C]
V_f1 = 0.005 [m^3]
V_g1=0.9 [m^3]
spvsat_f1=volume(Steam_iapws, P=P[1],x=0)  "sat. liq. specific volume, m^3/kg"
spvsat_g1=volume(Steam_iapws,P=P[1],x=1)  "sat. vap. specific volume, m^3/kg"
m_f1=V_f1/spvsat_f1 "sat. liq. mass, kg"
m_g1=V_g1/spvsat_g1 "sat. vap. mass, kg"
m_tot=m_f1+m_g1
V[1]=V_f1+V_g1
spvol[1]=V[1]/m_tot  "specific volume1, m^3"
T[1]=temperature(Steam_iapws, P=P[1],v=spvol[1])"C"
"The final volume is calculated from the specific volume at the final T and P"
spvol[2]=volume(Steam_iapws, P=P[2], T=T[2]) "specific volume2, m^3/kg"
V[2]=m_tot*spvol[2]
```

P_1 [kPa]	m_{tot} [kg]
100	5.324
200	5.731
300	6.145
400	6.561
500	6.978
600	7.395
700	7.812
800	8.23
900	8.648
1000	9.066



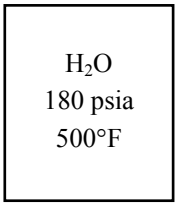


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3-55E Superheated water vapor cools at constant volume until the temperature drops to 250°F. At the final state, the pressure, the quality, and the enthalpy are to be determined.

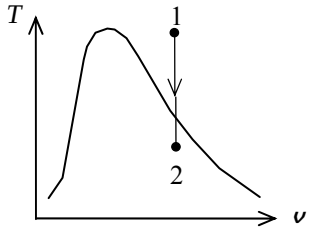
Analysis This is a constant volume process ($v = V/m$ = constant), and the initial specific volume is determined to be

$$\left.\begin{array}{l} P_1 = 180 \text{ psia} \\ T_1 = 500^\circ\text{F} \end{array}\right\} v_1 = 3.0433 \text{ ft}^3/\text{lbm} \qquad \text{(Table A-6E)}$$



At 250°F, $v_f = 0.01700$ ft^3/lbm and $v_g = 13.816$ ft^3/lbm. Thus at the final state, the tank will contain saturated liquid-vapor mixture since $v_f < v < v_g$, and the final pressure must be the saturation pressure at the final temperature,

$$P = P_{\text{sat@250°F}} = \mathbf{29.84 \text{ psia}}$$



(*b*) The quality at the final state is determined from

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{3.0433 - 0.01700}{13.816 - 0.01700} = \mathbf{0.219}$$

(*c*) The enthalpy at the final state is determined from

$$h = h_f + x h_{fg} = 218.63 + 0.219 \times 945.41 = \mathbf{426.0 \text{ Btu/lbm}}$$

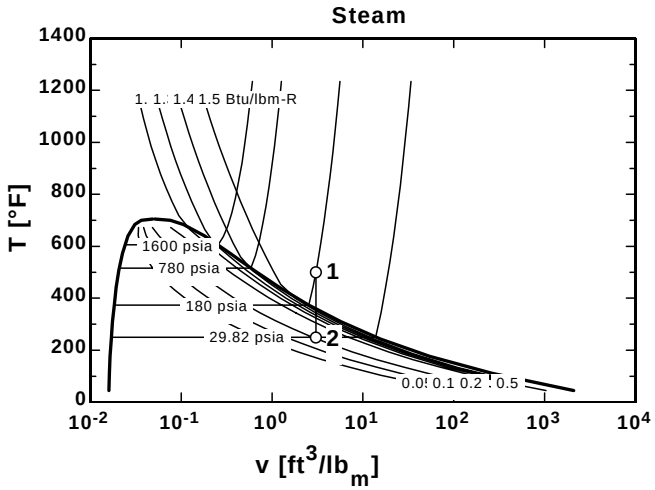
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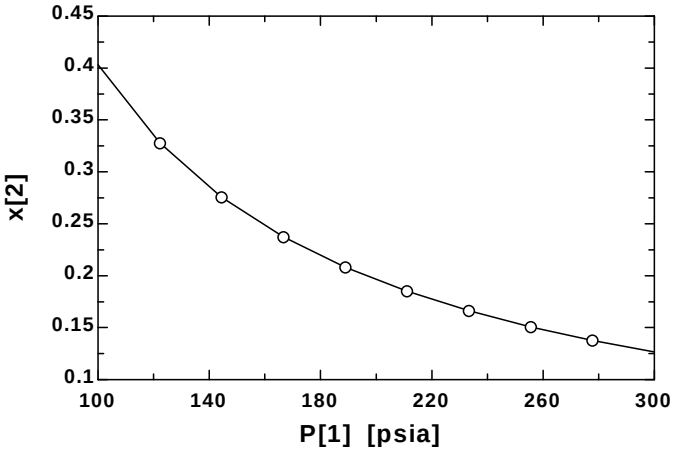
3-56E Problem 3-55E is reconsidered. The effect of initial pressure on the quality of water at the final state as the pressure varies from 100 psi to 300 psi is to be investigated. The quality is to be plotted against initial pressure, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
T[1]=500  [F]
P[1]=180 [psia]
T[2]=250 [F]
v[ 1]=volume(steam_iapws,T=T[1],P=P[1])
v[2]=v[1]
P[2]=pressure(steam_iapws,T=T[2],v=v[2])
h[2]=enthalpy(steam_iapws,T=T[2],v=v[2])
x[2]=quality(steam_iapws,T=T[2],v=v[2])
```

P_1 [psia]	x_2
100	0.4037
122.2	0.3283
144.4	0.2761
166.7	0.2378
188.9	0.2084
211.1	0.1853
233.3	0.1665
255.6	0.151
277.8	0.1379
300	0.1268





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3-57 The properties of compressed liquid water at a specified state are to be determined using the compressed liquid tables, and also by using the saturated liquid approximation, and the results are to be compared.

Analysis Compressed liquid can be approximated as saturated liquid at the given temperature. Then from Table A-4,

$$T = 80°\text{C} \quad \Rightarrow \quad \begin{aligned} v &\cong v_{f@80°\text{C}} = 0.001029 \text{ m}^3/\text{kg} \quad (0.90\% \text{ error}) \\ u &\cong u_{f@80°\text{C}} = 334.97 \text{ kJ/kg} \quad (1.35\% \text{ error}) \\ h &\cong h_{f@80°\text{C}} = 335.02 \text{ kJ/kg} \quad (4.53\% \text{ error}) \end{aligned}$$

From compressed liquid table (Table A-7),

$$\left.\begin{matrix} P = 20 \text{ MPa} \\ T = 80°\text{C} \end{matrix}\right\} \begin{matrix} v = 0.00102 \text{ m}^3/\text{kg} \\ u = 330.50 \text{ kJ/kg} \\ h = 350.90 \text{ kJ/kg} \end{matrix}$$

The percent errors involved in the saturated liquid approximation are listed above in parentheses.

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3-58 Problem 3-57 is reconsidered. Using EES, the indicated properties of compressed liquid are to be determined, and they are to be compared to those obtained using the saturated liquid approximation.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
T=80 [C]
P=20000 [kPa]

"Analysis"
Fluid$='steam_iapws'

"Saturated liquid assumption"
v_f=volume(Fluid$, T=T, x=0)
u_f=intenergy(Fluid$, T=T, x=0)
h_f=enthalpy(Fluid$, T=T, x=0)

"Compressed liquid treatment"
v=volume(Fluid$, T=T, P=P)
u=intenergy(Fluid$, T=T, P=P)
h=enthalpy(Fluid$, T=T, P=P)

"Percentage Errors"
error_v=100*(v_f-v)/v
error_u=100*(u_f-u)/u
error_h=100*(h-h_f)/h

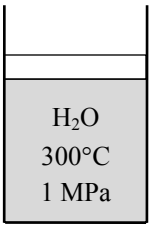

SOLUTION
error_h=4.527
error_u=1.351
error_v=0.8987
Fluid$='steam_iapws'
h=350.90 [kJ/kg]
h_f=335.02 [kJ/kg]
P=20000 [kPa]
T=80 [C]
u=330.50 [kJ/kg]
u_f=334.97 [kJ/kg]
v=0.00102 [m^3/kg]
v_f=0.001029 [m^3/kg]
```

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3-59 Superheated steam in a piston-cylinder device is cooled at constant pressure until half of the mass condenses. The final temperature and the volume change are to be determined, and the process should be shown on a T-v diagram.

Analysis (*b*) At the final state the cylinder contains saturated liquid-vapor mixture, and thus the final temperature must be the saturation temperature at the final pressure,

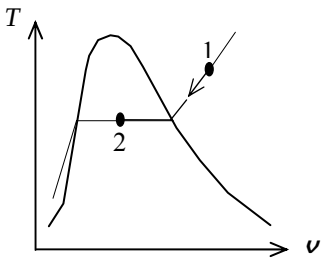
$$T = T_{\text{sat@1 MPa}} = \mathbf{179.88°C} \qquad \text{(Table A-5)}$$



(*c*) The quality at the final state is specified to be $x_2 = 0.5$. The specific volumes at the initial and the final states are

$$\left.\begin{matrix} P_1 = 1.0 \text{ MPa} \\ T_1 = 300^\circ\text{C} \end{matrix}\right\} v_1 = 0.25799 \text{ m}^3/\text{kg} \qquad \text{(Table A-6)}$$

$$\left.\begin{matrix} P_2 = 1.0 \text{ MPa} \\ x_2 = 0.5 \end{matrix}\right\} \begin{aligned} v_2 &= v_f + x_2 v_{fg} \\ &= 0.001127 + 0.5 \times (0.19436 - 0.001127) \\ &= 0.09775 \text{ m}^3/\text{kg} \end{aligned}$$



Thus,

$$\Delta V = m(v_2 - v_1) = (0.8 \text{ kg})(0.09775 - 0.25799)\text{m}^3/\text{kg} = \mathbf{-0.1282 \text{ m}^3}$$

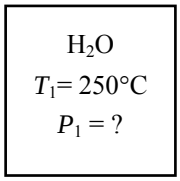
3-60 The water in a rigid tank is cooled until the vapor starts condensing. The initial pressure in the tank is to be determined.

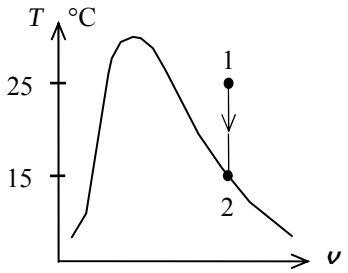
Analysis This is a constant volume process ($v = V/m$ = constant), and the initial specific volume is equal to the final specific volume that is

$$v_1 = v_2 = v_{g@124°\text{C}} = 0.79270 \text{ m}^3/\text{kg} \quad \text{(Table A-4)}$$

since the vapor starts condensing at 150°C. Then from Table A-6,

$$\left.\begin{matrix} T_1 = 250°\text{C} \\ v_1 = 0.79270 \text{ m}^3/\text{kg} \end{matrix}\right\} P_1 = \mathbf{0.30 \text{ MPa}}$$





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3-61 Heat is supplied to a piston-cylinder device that contains water at a specified state. The volume of the tank, the final temperature and pressure, and the internal energy change of water are to be determined.

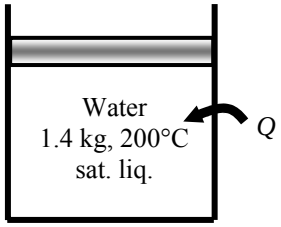
Properties The saturated liquid properties of water at 200°C are: $v_f = 0.001157\ \text{m}^3/\text{kg}$ and $u_f = 850.46\ \text{kJ/kg}$ (Table A-4).

Analysis (*a*) The cylinder initially contains saturated liquid water. The volume of the cylinder at the initial state is

$$V_1 = mv_1 = (1.4\ \text{kg})(0.001157\ \text{m}^3/\text{kg}) = 0.001619\ \text{m}^3$$

The volume at the final state is

$$V = 4(0.001619) = \mathbf{0.006476\ m^3}$$



(*b*) The final state properties are

$$v_2 = \frac{V}{m} = \frac{0.006476\ \text{m}^3}{1.4\ \text{kg}} = 0.004626\ \text{m}^3/\text{kg}$$

$$\left.\begin{array}{l} v_2 = 0.004626\ \text{m}^3/\text{kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{371.3°C} \\ P_2 = \mathbf{21{,}367\ kPa} \\ u_2 = 2201.5\ \text{kJ/kg} \end{array} \quad \text{(Table A-4 or A-5 or EES)}$$

(*c*) The total internal energy change is determined from

$$\Delta U = m(u_2 - u_1) = (1.4\ \text{kg})(2201.5 - 850.46)\ \text{kJ/kg} = \mathbf{1892\ kJ}$$

3-62E The error involved in using the enthalpy of water by the incompressible liquid approximation is to be determined.

Analysis The state of water is compressed liquid. From the steam tables,

$$\left.\begin{array}{l} P = 1500\ \text{psia} \\ T = 400°\text{F} \end{array}\right\} h = 376.51\ \text{Btu/lbm} \quad \text{(Table A - 7E)}$$

Based upon the incompressible liquid approximation,

$$\left.\begin{array}{l} P = 1500\ \text{psia} \\ T = 400°\text{F} \end{array}\right\} h \cong h_{f\,@\,400°\text{F}} = 375.04\ \text{Btu/lbm} \quad \text{(Table A - 4E)}$$

The error involved is

$$\text{Percent Error} = \frac{376.51 - 375.04}{376.51} \times 100 = \mathbf{0.39\%}$$

which is quite acceptable in most engineering calculations.

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3-63 The errors involved in using the specific volume and enthalpy of water by the incompressible liquid approximation are to be determined.

Analysis The state of water is compressed liquid. From the steam tables,

$$\left.\begin{array}{l} P = 20\ \text{MPa} \\ T = 140°\text{C} \end{array}\right\} \begin{array}{l} \nu = 0.0010679\ \text{m}^3/\text{kg} \\ h = 602.07\ \text{kJ/kg} \end{array} \quad \text{(Table A - 7)}$$

Based upon the incompressible liquid approximation,

$$\left.\begin{array}{l} P = 20\ \text{MPa} \\ T = 140°\text{C} \end{array}\right\} \begin{array}{l} \nu \cong \nu_{f\,@140°\text{C}} = 0.001080\ \text{m}^3/\text{kg} \\ h \cong h_{f\,@140°\text{C}} = 589.16\ \text{kJ/kg} \end{array} \quad \text{(Table A - 4)}$$

The errors involved are

$$\text{Percent Error (specific volume)} = \frac{0.0010679 - 0.001080}{0.0010679} \times 100 = \mathbf{-1.13\%}$$

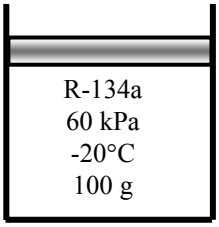
$$\text{Percent Error (enthalpy)} = \frac{602.07 - 589.16}{602.07} \times 100 = \mathbf{2.14\%}$$

which are quite acceptable in most engineering calculations.

3-64 A piston-cylinder device that is filled with R-134a is heated. The volume change is to be determined.

Analysis The initial specific volume is

$$\left.\begin{array}{l} P_1 = 60\ \text{kPa} \\ T_1 = -20°\text{C} \end{array}\right\} \nu_1 = 0.33608\ \text{m}^3/\text{kg} \quad \text{(Table A -13)}$$

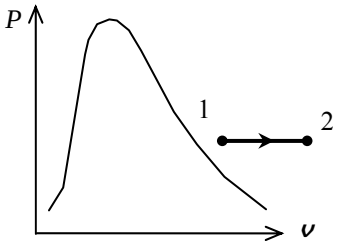


and the initial volume is

$$V_1 = m\nu_1 = (0.100\ \text{kg})(0.33608\ \text{m}^3/\text{kg}) = 0.033608\ \text{m}^3$$

At the final state, we have

$$\left.\begin{array}{l} P_2 = 60\ \text{kPa} \\ T_2 = 100°\text{C} \end{array}\right\} \nu_2 = 0.50410\ \text{m}^3/\text{kg} \quad \text{(Table A -13)}$$



$$V_2 = m\nu_2 = (0.100\ \text{kg})(0.50410\ \text{m}^3/\text{kg}) = 0.050410\ \text{m}^3$$

The volume change is then

$$\Delta V = V_2 - V_1 = 0.050410 - 0.033608 = \mathbf{0.0168\ m^3}$$

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3-65 A rigid vessel is filled with refrigerant-134a. The total volume and the total internal energy are to be determined.

Properties The properties of R-134a at the given state are (Table A-13).

$$\left.\begin{array}{l} P = 800 \text{ kPa} \\ T = 120^\circ\text{C} \end{array}\right\} \begin{array}{l} u = 327.87 \text{ kJ/kg} \\ v = 0.037625 \text{ m}^3\text{/kg} \end{array}$$

R-134a
2 kg
800 kPa
120°C

Analysis The total volume and internal energy are determined from

$$V = mv = (2 \text{ kg})(0.037625 \text{ m}^3\text{/kg}) = \mathbf{0.0753\ m^3}$$
$$U = mu = (2 \text{ kg})(327.87 \text{ kJ/kg}) = \mathbf{655.7\ kJ}$$

3-66 A rigid vessel contains R-134a at specified temperature. The pressure, total internal energy, and the volume of the liquid phase are to be determined.

Analysis (*a*) The specific volume of the refrigerant is

$$v = \frac{V}{m} = \frac{0.5 \text{ m}^3}{10 \text{ kg}} = 0.05 \text{ m}^3\text{/kg}$$

R-134a
10 kg
-20°C

At -20°C, $v_f = 0.0007362$ m^3/kg and $v_g = 0.14729$ m^3/kg (Table A-11). Thus the tank contains saturated liquid-vapor mixture since $v_f < v < v_g$, and the pressure must be the saturation pressure at the specified temperature,

$$P = P_{\text{sat@}-20^\circ\text{C}} = \mathbf{132.82\ kPa}$$

(*b*) The quality of the refrigerant-134a and its total internal energy are determined from

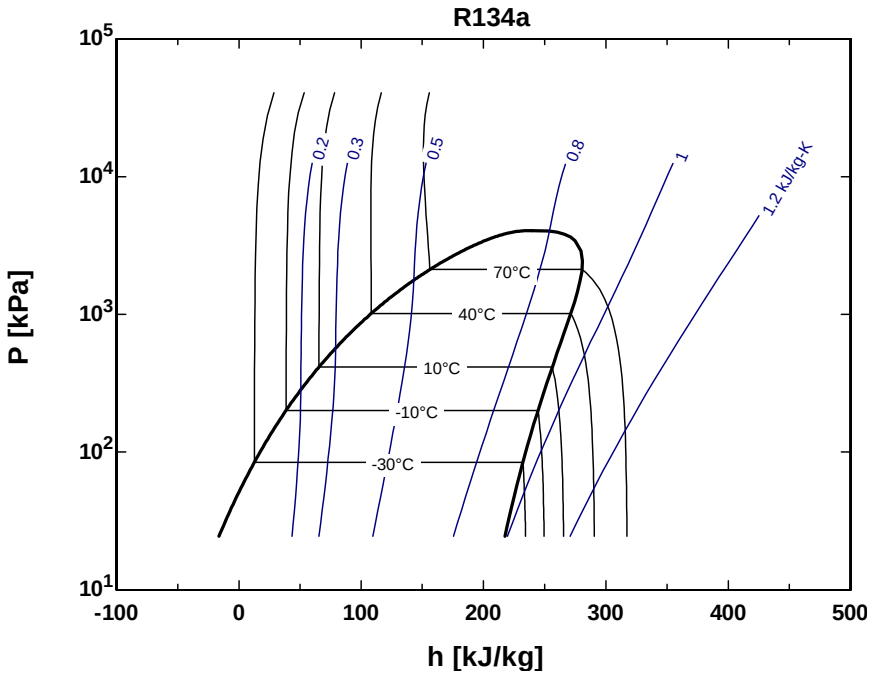
$$x = \frac{v - v_f}{v_{fg}} = \frac{0.05 - 0.0007362}{0.14729 - 0.0007362} = 0.3361$$
$$u = u_f + xu_{fg} = 25.39 + 0.3361 \times 193.45 = 90.42 \text{ kJ/kg}$$
$$U = mu = (10 \text{ kg})(90.42 \text{ kJ/kg}) = \mathbf{904.2\ kJ}$$

(*c*) The mass of the liquid phase and its volume are determined from

$$m_f = (1 - x)m_t = (1 - 0.3361) \times 10 = 6.639 \text{ kg}$$
$$V_f = m_f v_f = (6.639 \text{ kg})(0.0007362 \text{ m}^3\text{/kg}) = \mathbf{0.00489\ m^3}$$

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3-67 The Pessure-Enthalpy diagram of R-134a showing some constant-temperature and constant-entropy lines are obtained using Property Plot feature of EES.



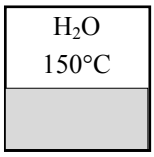
3-68 A rigid vessel that contains a saturated liquid-vapor mixture is heated until it reaches the critical state. The mass of the liquid water and the volume occupied by the liquid at the initial state are to be determined.

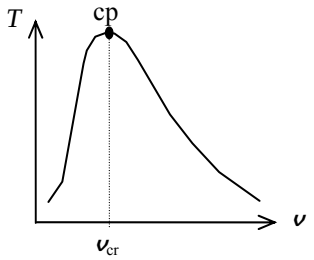
Analysis This is a constant volume process ($v = V/m$ = constant) to the critical state, and thus the initial specific volume will be equal to the final specific volume, which is equal to the critical specific volume of water,

$$v_1 = v_2 = v_{cr} = 0.003106 \text{ m}^3/\text{kg} \quad \text{(last row of Table A-4)}$$

The total mass is

$$m = \frac{V}{v} = \frac{0.3 \text{ m}^3}{0.003106 \text{ m}^3/\text{kg}} = 96.60 \text{ kg}$$





At 150°C, $v_f = 0.001091$ m^3/kg and $v_g = 0.39248$ m^3/kg (Table A-4). Then the quality of water at the initial state is

$$x_1 = \frac{v_1 - v_f}{v_{fg}} = \frac{0.003106 - 0.001091}{0.39248 - 0.001091} = 0.005149$$

Then the mass of the liquid phase and its volume at the initial state are determined from

$$m_f = (1 - x_1)m_t = (1 - 0.005149)(96.60) = \mathbf{96.10 \text{ kg}}$$

$$V_f = m_f v_f = (96.10 \text{ kg})(0.001091 \text{ m}^3/\text{kg}) = \mathbf{0.105 \text{ m}^3}$$

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Ideal Gas

3-69C Mass m is simply the amount of matter; molar mass M is the mass of one mole in grams or the mass of one kmol in kilograms. These two are related to each other by $m = NM$, where N is the number of moles.

3-70C A gas can be treated as an ideal gas when it is at a high temperature or low pressure relative to its critical temperature and pressure.

3-71C R_u is the universal gas constant that is the same for all gases whereas R is the specific gas constant that is different for different gases. These two are related to each other by $R = R_u / M$, where M is the molar mass of the gas.

3-72C Propane (molar mass = 44.1 kg/kmol) poses a greater fire danger than methane (molar mass = 16 kg/kmol) since propane is heavier than air (molar mass = 29 kg/kmol), and it will settle near the floor. Methane, on the other hand, is lighter than air and thus it will rise and leak out.

3-73 The specific volume of nitrogen at a specified state is to be determined.

Assumptions At specified conditions, nitrogen behaves as an ideal gas.

Properties The gas constant of nitrogen is $R = 0.2968$ kJ/kg·K (Table A-1).

Analysis According to the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(227+273\ \text{K})}{300\ \text{kPa}} = \mathbf{0.495\ m^3/kg}$$

3-74E The temperature in a container that is filled with oxygen is to be determined.

Assumptions At specified conditions, oxygen behaves as an ideal gas.

Properties The gas constant of oxygen is $R = 0.3353$ psia·ft^3/lbm·R (Table A-1E).

Analysis The definition of the specific volume gives

$$v = \frac{V}{m} = \frac{3\ \text{ft}^3}{2\ \text{lbm}} = 1.5\ \text{ft}^3/\text{lbm}$$

Using the ideal gas equation of state, the temperature is

$$T = \frac{Pv}{R} = \frac{(80\ \text{psia})(1.5\ \text{ft}^3/\text{lbm})}{0.3353\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}} = \mathbf{358\ R}$$

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3-75 The volume of a container that is filled with helium at a specified state is to be determined.

Assumptions At specified conditions, helium behaves as an ideal gas.

Properties The gas constant of helium is $R = 2.0769$ kJ/kg·K (Table A-1).

Analysis According to the ideal gas equation of state,

$$V = \frac{mRT}{P} = \frac{(2\text{ kg})(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(27+273\text{ K})}{300\text{ kPa}} = \mathbf{4.154\ m^3}$$

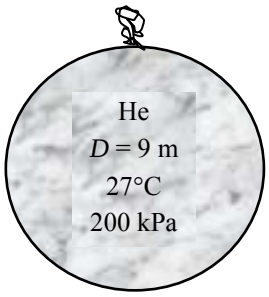
3-76 A balloon is filled with helium gas. The mole number and the mass of helium in the balloon are to be determined.

Assumptions At specified conditions, helium behaves as an ideal gas.

Properties The universal gas constant is $R_u = 8.314$ kPa.m^3/kmol.K. The molar mass of helium is 4.0 kg/kmol (Table A-1).

Analysis The volume of the sphere is

$$V = \frac{4}{3}\pi r^3 = \frac{4}{3}\pi(4.5\text{ m})^3 = 381.7\text{ m}^3$$



Assuming ideal gas behavior, the mole numbers of He is determined from

$$N = \frac{PV}{R_u T} = \frac{(200\text{ kPa})(381.7\text{ m}^3)}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(300\text{ K})} = \mathbf{30.61\ kmol}$$

Then the mass of He can be determined from

$$m = NM = (30.61\text{ kmol})(4.0\text{ kg/kmol}) = \mathbf{123\ kg}$$

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3-77 Problem 3-76 is to be reconsidered. The effect of the balloon diameter on the mass of helium contained in the balloon is to be determined for the pressures of (a) 100 kPa and (b) 200 kPa as the diameter varies from 5 m to 15 m. The mass of helium is to be plotted against the diameter for both cases.

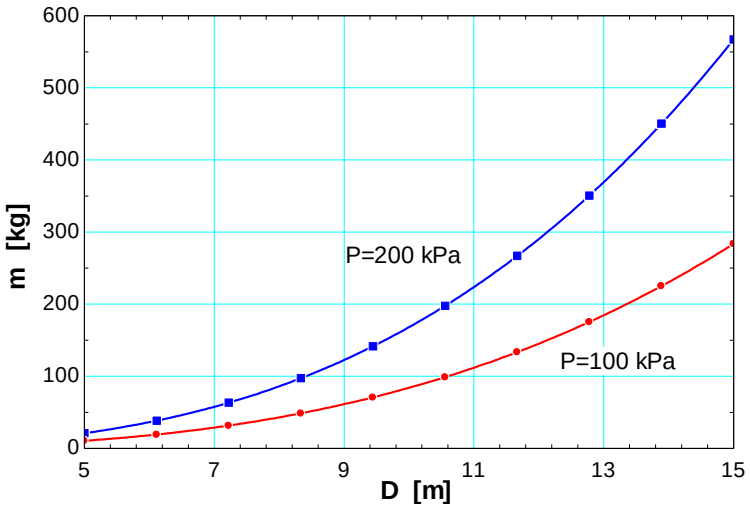
Analysis The problem is solved using EES, and the solution is given below.

```
"Given Data"
{D=9 [m]}
T=27 [C]
P=200 [kPa]
R_u=8.314 [kJ/kmol-K]

"Solution"
P*V=N*R_u*(T+273)
V=4*pi*(D/2)^3/3
m=N*MOLARMASS(Helium)
```

D [m]	m [kg]
5	21.01
6.111	38.35
7.222	63.31
8.333	97.25
9.444	141.6
10.56	197.6
11.67	266.9
12.78	350.6
13.89	450.2
15	567.2

P=200 kPa



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3-78 Two rigid tanks connected by a valve to each other contain air at specified conditions. The volume of the second tank and the final equilibrium pressure when the valve is opened are to be determined.

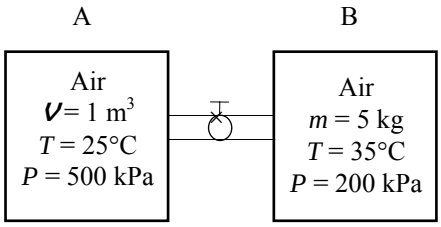
Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1).

Analysis Let's call the first and the second tanks A and B. Treating air as an ideal gas, the volume of the second tank and the mass of air in the first tank are determined to be

$$\boldsymbol{V}_B = \left(\frac{m_1 R T_1}{P_1}\right)_B = \frac{(5\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})}{200\text{ kPa}} = \mathbf{2.21\ m^3}$$

$$m_A = \left(\frac{P_1 \boldsymbol{V}}{R T_1}\right)_A = \frac{(500\text{ kPa})(1.0\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 5.846\text{ kg}$$



Thus,

$$\boldsymbol{V} = \boldsymbol{V}_A + \boldsymbol{V}_B = 1.0 + 2.21 = 3.21\text{ m}^3$$
$$m = m_A + m_B = 5.846 + 5.0 = 10.846\text{ kg}$$

Then the final equilibrium pressure becomes

$$P_2 = \frac{mRT_2}{\boldsymbol{V}} = \frac{(10.846\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{3.21\text{ m}^3} = \mathbf{284.1\ kPa}$$

3-79E An elastic tank contains air at a specified state. The volume is doubled at the same pressure. The initial volume and the final temperature are to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

Analysis According to the ideal gas equation of state,

$$P\boldsymbol{V} = nR_u T$$
$$(32\text{ psia})\boldsymbol{V} = (2.3\text{ lbmol})(10.73\text{ psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(65+460)\text{ R}$$
$$\boldsymbol{V} = \mathbf{404.9\ ft^3}$$

$$\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} = \frac{T_2}{T_1} \longrightarrow 2 = \frac{T_2}{(65+460)\text{ R}} \longrightarrow T_2 = \mathbf{1050\ R} = \mathbf{590°F}$$

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3-80 An ideal gas in a rigid tank is cooled to a final gage pressure. The final temperature is to be determined.

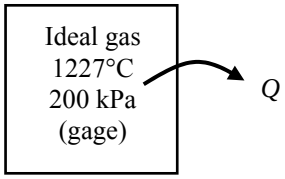
Assumptions The gas is specified as an ideal gas so that ideal gas relation can be used.

Analysis According to the ideal gas equation of state at constant volume,

$$m_1 = m_1$$
$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

Since $V_1 = V_2$

P_{atm} = 100 kPa



Then,

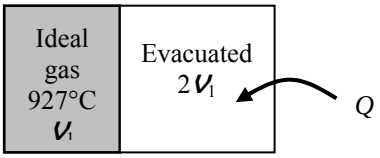
$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$
$$T_2 = T_1 \frac{P_2}{P_2} = \left[(1227 + 273)\ \text{K}\right] \frac{(50 + 100)\ \text{kPa}}{(200 + 100)\ \text{kPa}} = 750\ \text{K} = \mathbf{477°C}$$

3-81 One side of a two-sided tank contains an ideal gas while the other side is evacuated. The partition is removed and the gas fills the entire tank. The gas is also heated to a final pressure. The final temperature is to be determined.

Assumptions The gas is specified as an ideal gas so that ideal gas relation can be used.

Analysis According to the ideal gas equation of state,

$$P_2 = P_1$$
$$V_2 = V_1 + 2V_1 = 3V_1$$



Applying these,

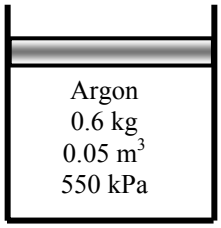
$$m_1 = m_1$$
$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$
$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$
$$T_2 = T_1 \frac{V_2}{V_1} = T_1 \frac{3V_1}{V_1} = 3T_1 = 3\left[927 + 273)\ \text{K}\right] = 3600\ \text{K} = \mathbf{3327°C}$$

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3-82 A piston-cylinder device containing argon undergoes an isothermal process. The final pressure is to be determined.

Assumptions At specified conditions, argon behaves as an ideal gas.

Properties The gas constant of argon is $R = 0.2081$ kJ/kg·K (Table A-1).



Analysis Since the temperature remains constant, the ideal gas equation gives

$$m = \frac{P_1 V_1}{RT} = \frac{P_2 V_2}{RT} \longrightarrow P_1 V_1 = P_2 V_2$$

which when solved for final pressure becomes

$$P_2 = P_1 \frac{V_1}{V_2} = P_1 \frac{V_1}{2V_1} = 0.5P_1 = 0.5(550 \text{ kPa}) = \mathbf{275\ kPa}$$

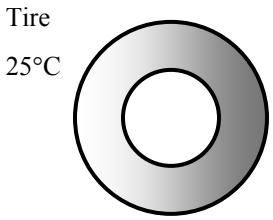
3-83 An automobile tire is inflated with air. The pressure rise of air in the tire when the tire is heated and the amount of air that must be bled off to reduce the temperature to the original value are to be determined.

Assumptions **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tire remains constant.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1).

Analysis Initially, the absolute pressure in the tire is

$$P_1 = P_g + P_{\text{atm}} = 210 + 100 = 310 \text{kPa}$$



Treating air as an ideal gas and assuming the volume of the tire to remain constant, the final pressure in the tire can be determined from

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{323 \text{ K}}{298 \text{ K}} (310 \text{ kPa}) = 336 \text{ kPa}$$

Thus the pressure rise is

$$\Delta P = P_2 - P_1 = 336 - 310 = \mathbf{26\ kPa}$$

The amount of air that needs to be bled off to restore pressure to its original value is

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = 0.0906 \text{ kg}$$

$$m_2 = \frac{P_1 V}{RT_2} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(323 \text{ K})} = 0.0836 \text{ kg}$$

$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070\ kg}$$

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Compressibility Factor

3-84C It represent the deviation from ideal gas behavior. The further away it is from 1, the more the gas deviates from ideal gas behavior.

3-85C All gases have the same compressibility factor Z at the same reduced temperature and pressure.

3-86C Reduced pressure is the pressure normalized with respect to the critical pressure; and reduced temperature is the temperature normalized with respect to the critical temperature.

3-87 The specific volume of steam is to be determined using the ideal gas relation, the compressibility chart, and the steam tables. The errors involved in the first two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$R = 0.4615$ kPa·m^3/kg·K, $T_{cr} = 647.1$ K, $P_{cr} = 22.06$ MPa

Analysis (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(623.15 \text{ K})}{15{,}000 \text{ kPa}} = \mathbf{0.01917 \ m^3/kg \ \ (67.0\% \ error)}$$

(*b*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{10 \text{ MPa}}{22.06 \text{ MPa}} = 0.453 \\ T_R &= \frac{T}{T_{cr}} = \frac{673 \text{ K}}{647.1 \text{ K}} = 1.04 \end{aligned}\right\} Z = 0.65$$

H_2O
15 MPa
350°C

Thus,

$$v = Zv_{\text{ideal}} = (0.65)(0.01917 \text{ m}^3/\text{kg}) = \mathbf{0.01246 \ m^3/kg \ \ (8.5\% \ error)}$$

(*c*) From the superheated steam table (Table A-6),

$$\left.\begin{aligned} P &= 15 \text{ MPa} \\ T &= 350°\text{C} \end{aligned}\right\} \ v = \mathbf{0.01148 \ m^3/kg}$$

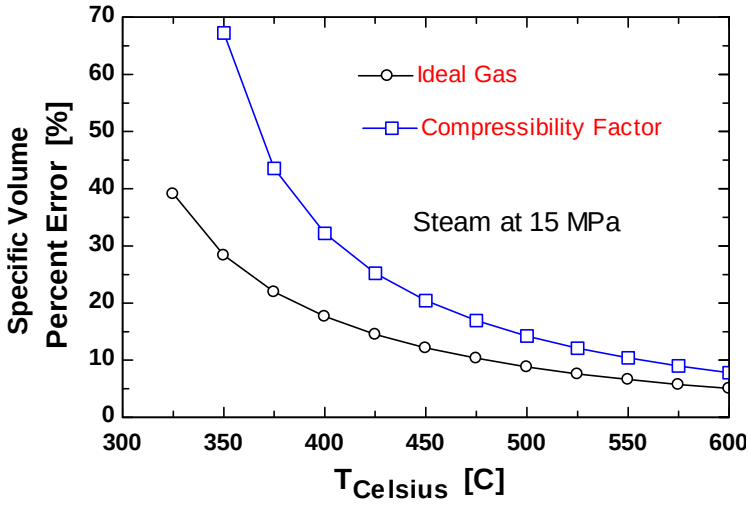
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3-88 Problem 3-87 is reconsidered. The problem is to be solved using the general compressibility factor feature of EES (or other) software. The specific volume of water for the three cases at 15 MPa over the temperature range of 350°C to 600°C in 25°C intervals is to be compared, and the % error involved in the ideal gas approximation is to be plotted against temperature.

Analysis The problem is solved using EES, and the solution is given below.

```
P=15 [MPa]*Convert(MPa,kPa)
{T_Celsius= 350 [C]}
T=T_Celsius+273 "[K]"
T_critical=T_CRIT(Steam_iapws)
P_critical=P_CRIT(Steam_iapws)
{v=Vol/m}
P_table=P; P_comp=P;P_idealgas=P
T_table=T; T_comp=T;T_idealgas=T
v_table=volume(Steam_iapws,P=P_table,T=T_table) "EES data for steam as a real gas"
{P_table=pressure(Steam_iapws, T=T_table,v=v)}
{T_sat=temperature(Steam_iapws,P=P_table,v=v)}
MM=MOLARMASS(water)
R_u=8.314 [kJ/kmol-K]  "Universal gas constant"
R=R_u/MM "[kJ/kg-K], Particular gas constant"
P_idealgas*v_idealgas=R*T_idealgas  "Ideal gas equation"
z = COMPRESS(T_comp/T_critical,P_comp/P_critical)
P_comp*v_comp=z*R*T_comp "generalized Compressibility factor"
Error_idealgas=Abs(v_table-v_idealgas)/v_table*Convert(, %)
Error_comp=Abs(v_table-v_comp)/v_table*Convert(, %)
```

$Error_{comp}$ [%]	$Error_{ideal\ gas}$ [%]	$T_{Celcius}$ [C]
9.447	67.22	350
2.725	43.53	375
0.4344	32.21	400
0.5995	25.23	425
1.101	20.44	450
1.337	16.92	475
1.428	14.22	500
1.437	12.1	525
1.397	10.39	550
1.329	8.976	575
1.245	7.802	600



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3-89 The specific volume of R-134a is to be determined using the ideal gas relation, the compressibility chart, and the R-134a tables. The errors involved in the first two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of refrigerant-134a are, from Table A-1,

$R = 0.08149$ kPa·m^3/kg·K, $T_{cr} = 374.2$ K, $P_{cr} = 4.059$ MPa

Analysis (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.08149 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(343 \text{ K})}{900 \text{ kPa}} = \mathbf{0.03105 \text{ m}^3/\text{kg}} \quad (\mathbf{13.3\% \text{ error}})$$

(*b*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{0.9 \text{ MPa}}{4.059 \text{ MPa}} = 0.222 \\ T_R &= \frac{T}{T_{cr}} = \frac{343 \text{ K}}{374.2 \text{ K}} = 0.917 \end{aligned}\right\} Z = 0.894$$

R-134a
0.9 MPa
70°C

Thus,

$$v = Zv_{\text{ideal}} = (0.894)(0.03105 \text{ m}^3/\text{kg}) = \mathbf{0.02776 \text{ m}^3/\text{kg}} \quad \mathbf{(1.3\% \text{error})}$$

(*c*) From the superheated refrigerant table (Table A-13),

$$\left.\begin{aligned} P &= 0.9 \text{ MPa} \\ T &= 70°\text{C} \end{aligned}\right\} v = \mathbf{0.027413 \text{ m}^3/\text{kg}}$$

3-90 The specific volume of steam is to be determined using the ideal gas relation, the compressibility chart, and the steam tables. The errors involved in the first two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$R = 0.4615$ kPa·m^3/kg·K, $T_{cr} = 647.1$ K, $P_{cr} = 22.06$ MPa

Analysis (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(723 \text{ K})}{3500 \text{ kPa}} = \mathbf{0.09533 \text{ m}^3/\text{kg}} \quad \mathbf{(3.7\% \text{ error})}$$

(*b*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{3.5 \text{ MPa}}{22.06 \text{ MPa}} = 0.159 \\ T_R &= \frac{T}{T_{cr}} = \frac{723 \text{ K}}{647.1 \text{ K}} = 1.12 \end{aligned}\right\} Z = 0.961$$

H_2O
3.5 MPa
450°C

Thus,

$$v = Zv_{\text{ideal}} = (0.961)(0.09533 \text{ m}^3/\text{kg}) = \mathbf{0.09161 \text{ m}^3/\text{kg}} \quad \mathbf{(0.4\% \text{ error})}$$

(*c*) From the superheated steam table (Table A-6),

$$\left.\begin{aligned} P &= 3.5 \text{ MPa} \\ T &= 450°\text{C} \end{aligned}\right\} v = \mathbf{0.09196 \text{ m}^3/\text{kg}}$$

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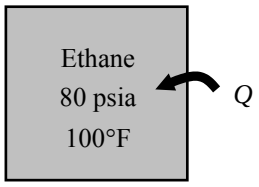
3-91E Ethane in a rigid vessel is heated. The final pressure is to be determined using the compressibility chart.

Properties The gas constant, the critical pressure, and the critical temperature of ethane are, from Table A-1E,

$$R = 0.3574 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}, \qquad T_{cr} = 549.8 \text{ R}, \qquad P_{cr} = 708 \text{ psia}$$

Analysis From the compressibility chart at the initial state (Fig. A-15),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{560 \text{ R}}{549.8 \text{ R}} = 1.019 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{80 \text{ psia}}{708 \text{ psia}} = 0.1130 \end{aligned}\right\} Z_1 = 0.963$$



The specific volume does not change during the process. Then,

$$v_1 = v_2 = \frac{Z_1 R T_1}{P_1} = \frac{(0.963)(0.3574 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(560 \text{ R})}{80 \text{ psia}} = 2.409 \text{ ft}^3/\text{lbm}$$

At the final state,

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{cr}} = \frac{1000 \text{ R}}{549.8 \text{ R}} = 1.819 \\ v_{R2} &= \frac{v_{2,\text{actual}}}{RT_{cr}/P_{cr}} = \frac{2.4091 \text{ft}^3/\text{lbm}}{(0.3574 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(549.8 \text{ R})/(708 \text{ psia})} = 8.68 \end{aligned}\right\} Z_2 = 1.0$$

Thus,

$$P_2 = \frac{Z_2 R T_2}{v_2} = \frac{(1.0)(0.3574 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1000 \text{ R})}{2.409 \text{ ft}^3/\text{lbm}} = \mathbf{148 \text{ psia}}$$

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3-92 Ethylene is heated at constant pressure. The specific volume change of ethylene is to be determined using the compressibility chart.

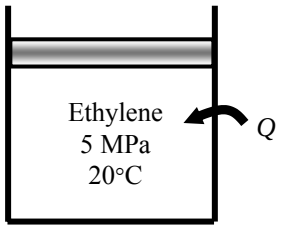
Properties The gas constant, the critical pressure, and the critical temperature of ethane are, from Table A-1,

$$R = 0.2964 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 282.4 \text{ K}, \qquad P_{cr} = 5.12 \text{ MPa}$$

Analysis From the compressibility chart at the initial and final states (Fig. A-15),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{293 \text{ K}}{282.4 \text{ K}} = 1.038 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{5 \text{ MPa}}{5.12 \text{ MPa}} = 0.977 \end{aligned}\right\} \; Z_1 = 0.56$$

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{cr}} = \frac{473 \text{ K}}{282.4 \text{ KR}} = 1.675 \\ P_{R2} &= P_{R1} = 0.977 \end{aligned}\right\} \; Z_1 = 0.961$$



The specific volume change is

$$\begin{aligned} \Delta v &= \frac{R}{P}(Z_2 T_2 - Z_1 T_1) \\ &= \frac{0.2964 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K}}{5000 \text{ kPa}}\left[(0.961)(473 \text{ K}) - (0.56)(293 \text{ K})\right] \\ &= \mathbf{0.0172 \text{ m}^3\text{/kg}} \end{aligned}$$

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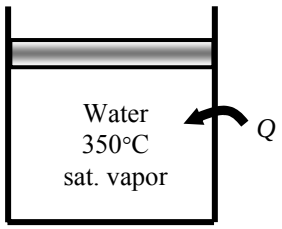
3-93 Water vapor is heated at constant pressure. The final temperature is to be determined using ideal gas equation, the compressibility charts, and the steam tables.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$R = 0.4615$ kPa·m^3/kg·K, $T_{cr} = 647.1$ K, $P_{cr} = 22.06$ MPa

Analysis (*a*) From the ideal gas equation,

$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (350 + 273\,\text{K})(2) = \mathbf{1246\ K}$$



(*b*) The pressure of the steam is

$$P_1 = P_2 = P_{\text{sat@350°C}} = 16{,}529\ \text{kPa}$$

From the compressibility chart at the initial state (Fig. A-15),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{623\,\text{K}}{647.1\,\text{KR}} = 0.963 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{16.529\,\text{MPa}}{22.06\,\text{MPa}} = 0.749 \end{aligned}\right\} \ Z_1 = 0.593,\ \nu_{R1} = 0.75$$

At the final state,

$$\left.\begin{aligned} P_{R2} &= P_{R1} = 0.749 \\ \nu_{R2} &= 2\nu_{R1} = 2(0.75) = 1.50 \end{aligned}\right\} \ Z_2 = 0.88$$

Thus,

$$T_2 = \frac{P_2\nu_2}{Z_2 R} = \frac{P_2}{Z_2}\frac{\nu_{R2}T_{cr}}{P_{cr}} = \frac{16{,}529\,\text{kPa}}{0.88}\frac{(1.50)(647.1\,\text{K})}{22{,}060\,\text{kPa}} = \mathbf{826\ K}$$

(*c*) From the superheated steam table,

$$\left.\begin{aligned} T_1 &= 350°\text{C} \\ x_1 &= 1 \end{aligned}\right\} \ \nu_1 = 0.008806\ \text{m}^3/\text{kg} \qquad \text{(Table A-4)}$$

$$\left.\begin{aligned} P_2 &= 16{,}529\ \text{kPa} \\ \nu_2 &= 2\nu_1 = 0.01761\ \text{m}^3/\text{kg} \end{aligned}\right\} \ T_2 = 477°\text{C} = \mathbf{750\ K} \qquad \text{(from Table A-6 or EES)}$$

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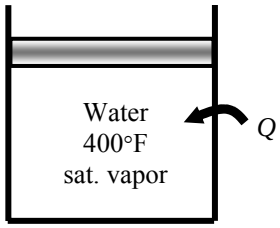
3-94E Water vapor is heated at constant pressure. The final temperature is to be determined using ideal gas equation, the compressibility charts, and the steam tables.

Properties The critical pressure and the critical temperature of water are, from Table A-1E,

$R = 0.5956$ psia·ft^3/lbm·R, $T_{cr} = 1164.8$ R, $P_{cr} = 3200$ psia

Analysis (*a*) From the ideal gas equation,

$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (400 + 460\ \text{R})(2) = \mathbf{1720\ R}$$



(*b*) The properties of steam are (Table A-4E)

$$P_1 = P_2 = P_{sat@400°F} = 247.26\ \text{psia}$$
$$\nu_1 = \nu_{g@400°F} = 1.8639\ \text{ft}^3/\text{lbm}$$
$$\nu_2 = 2\nu_1 = 3.7278\ \text{ft}^3/\text{lbm}$$

At the final state, from the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_{R2} &= \frac{P_2}{P_{cr}} = \frac{247.26\ \text{psia}}{3200\ \text{psia}} = 0.0773 \\ \nu_{R2} &= \frac{\nu_{2,\text{actual}}}{RT_{cr}/P_{cr}} = \frac{3.7278\ \text{ft}^3/\text{lbm}}{(0.5956\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1164.8\ \text{R})/(3200\ \text{psia})} = 17.19 \end{aligned}\right\} Z_2 = 0.985$$

Thus,

$$T_2 = \frac{P_2\nu_2}{Z_2 R} = \frac{(247.26\ \text{psia})(3.7278\ \text{ft}^3/\text{lbm})}{(0.985)(0.5956\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})} = \mathbf{1571R}$$

(*c*) From the superheated steam table,

$$\left.\begin{aligned} P_2 &= 247.26\ \text{psia} \\ \nu_2 &= 3.7278\ \text{ft}^3/\text{lbm} \end{aligned}\right\} T_2 = 1100°\text{F} = \mathbf{1560\ R} \qquad \text{(from Table A-6E or EES)}$$

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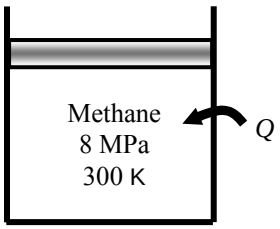
3-95 Methane is heated at constant pressure. The final temperature is to be determined using ideal gas equation and the compressibility charts.

Properties The gas constant, the critical pressure, and the critical temperature of methane are, from Table A-1,

$R = 0.5182$ kPa·m^3/kg·K, $T_{cr} = 191.1$ K, $P_{cr} = 4.64$ MPa

Analysis From the ideal gas equation,

$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (300\text{ K})(1.5) = \mathbf{450\ K}$$



From the compressibility chart at the initial state (Fig. A-15),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{300\text{ K}}{191.1\text{ K}} = 1.570 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{8\text{ MPa}}{4.64\text{ MPa}} = 1.724 \end{aligned}\right\} Z_1 = 0.88,\ \nu_{R1} = 0.80$$

At the final state,

$$\left.\begin{aligned} P_{R2} &= P_{R1} = 1.724 \\ \nu_{R2} &= 1.5\nu_{R1} = 1.5(0.80) = 1.2 \end{aligned}\right\} Z_2 = 0.975$$

Thus,

$$T_2 = \frac{P_2\nu_2}{Z_2 R} = \frac{P_2}{Z_2}\frac{\nu_{R2}T_{cr}}{P_{cr}} = \frac{8000\text{ kPa}}{0.975}\frac{(1.2)(191.1\text{ K})}{4640\text{ kPa}} = \mathbf{406\ K}$$

Of these two results, the accuracy of the second result is limited by the accuracy with which the charts may be read. Accepting the error associated with reading charts, the second temperature is the more accurate.

3-96 The percent error involved in treating CO_2 at a specified state as an ideal gas is to be determined.

Properties The critical pressure, and the critical temperature of CO_2 are, from Table A-1,

$T_{cr} = 304.2$K and $P_{cr} = 7.39$MPa

Analysis From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{5\text{ MPa}}{7.39\text{ MPa}} = 0.677 \\ T_R &= \frac{T}{T_{cr}} = \frac{298\text{ K}}{304.2\text{ K}} = 0.980 \end{aligned}\right\} Z = 0.69$$

CO_2
5 MPa
25°C

Then the error involved in treating CO_2 as an ideal gas is

$$\text{Error} = \frac{\nu - \nu_{\text{ideal}}}{\nu} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.69} = -0.44.9 \text{ or } \mathbf{44.9\%}$$

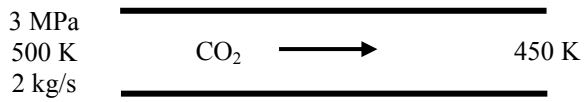
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3-97 CO_2 gas flows through a pipe. The volume flow rate and the density at the inlet and the volume flow rate at the exit of the pipe are to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of CO_2 are (Table A-1)

$R = 0.1889$ kPa·m^3/kg·K, $T_{cr} = 304.2$ K, $P_{cr} = 7.39$ MPa

Analysis



(*a*) From the ideal gas equation of state,

$$\dot{V}_1 = \frac{\dot{m}RT_1}{P_1} = \frac{(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})}{(3000\text{ kPa})} = \mathbf{0.06297\ m^3/kg}\quad\mathbf{(2.1\%\ error)}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{(3000\text{ kPa})}{(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})} = \mathbf{31.76\ kg/m^3}\quad\mathbf{(2.1\%\ error)}$$

$$\dot{V}_2 = \frac{\dot{m}RT_2}{P_2} = \frac{(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(450\text{ K})}{(3000\text{ kPa})} = \mathbf{0.05667\ m^3/kg}\quad\mathbf{(3.6\%\ error)}$$

(*b*) From the compressibility chart (EES function for compressibility factor is used)

$$\left.\begin{aligned} P_R &= \frac{P_1}{P_{cr}} = \frac{3\text{ MPa}}{7.39\text{ MPa}} = 0.407 \\ T_{R,1} &= \frac{T_1}{T_{cr}} = \frac{500\text{ K}}{304.2\text{ K}} = 1.64 \end{aligned}\right\} Z_1 = 0.9791$$

$$\left.\begin{aligned} P_R &= \frac{P_2}{P_{cr}} = \frac{3\text{ MPa}}{7.39\text{ MPa}} = 0.407 \\ T_{R,2} &= \frac{T_2}{T_{cr}} = \frac{450\text{ K}}{304.2\text{ K}} = 1.48 \end{aligned}\right\} Z_2 = 0.9656$$

Thus, $$\dot{V}_1 = \frac{Z_1\dot{m}RT_1}{P_1} = \frac{(0.9791)(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})}{(3000\text{ kPa})} = \mathbf{0.06165\ m^3/kg}$$

$$\rho_1 = \frac{P_1}{Z_1RT_1} = \frac{(3000\text{ kPa})}{(0.9791)(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})} = \mathbf{32.44\ kg/m^3}$$

$$\dot{V}_2 = \frac{Z_2\dot{m}RT_2}{P_2} = \frac{(0.9656)(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(450\text{ K})}{(3000\text{ kPa})} = \mathbf{0.05472\ m^3/kg}$$

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3-98 The specific volume of nitrogen gas is to be determined using the ideal gas relation and the compressibility chart. The errors involved in these two approaches are also to be determined.

Properties The gas constant, the critical pressure, and the critical temperature of nitrogen are, from Table A-1,

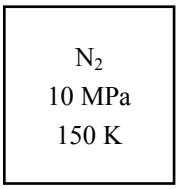
$R = 0.2968$ kPa·m^3/kg·K, $T_{cr} = 126.2$ K, $P_{cr} = 3.39$ MPa

Analysis (a) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(150 \text{ K})}{10{,}000 \text{ kPa}} = \mathbf{0.004452\,m^3/kg} \quad (86.4\% \text{ error})$$

(*b*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{10 \text{ MPa}}{3.39 \text{ MPa}} = 2.95 \\ T_R &= \frac{T}{T_{cr}} = \frac{150 \text{ K}}{126.2 \text{ K}} = 1.19 \end{aligned}\right\} Z = 0.54$$



Thus,

$$v = Zv_{\text{ideal}} = (0.54)(0.004452 \text{ m}^3/\text{kg}) = \mathbf{0.002404\,m^3/kg} \quad (0.7\% \text{ error})$$

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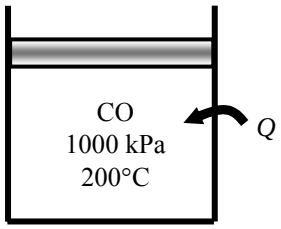
Other Equations of State

3-99C The constant *a* represents the increase in pressure as a result of intermolecular forces; the constant *b* represents the volume occupied by the molecules. They are determined from the requirement that the critical isotherm has an inflection point at the critical point.

3-100 Carbon monoxide is heated in a piston-cylinder device. The final volume of the carbon monoxide is to be determined using the ideal gas equation and the Benedict-Webb-Rubin equation of state.

Properties The gas constant and molar mass of CO are (Table A-1)

$R = 0.2968$ kPa·m^3/kg·K, $M = 28.011$ kg/kmol



Analysis (*a*) From the ideal gas equation of state,

$$V_2 = \frac{mRT_2}{P} = \frac{(0.100\,\text{kg})(0.2968\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(773\,\text{K})}{1000\,\text{kPa}} = \mathbf{0.02294\ m^3}$$

(*b*) Using the coefficients of Table 3-4 for carbon dioxide and the given data, the Benedict-Webb-Rubin equation of state for state 2 is

$$P_2 = \frac{R_u T_2}{\bar{v}_2} + \left(B_0 R_u T_2 - A_0 - \frac{C_0}{T_2^2}\right)\frac{1}{\bar{v}^2} + \frac{bR_u T_2 - a}{\bar{v}^3} + \frac{a\alpha}{\bar{v}^6} + \frac{c}{\bar{v}^3 T_2^2}\left(1 + \frac{\gamma}{\bar{v}^2}\right)\exp(-\gamma/\bar{v}^2)$$

$$1000 = \frac{(8.314)(773)}{\bar{v}_2} + \left(0.05454\times 8.314\times 773 - 135.9 - \frac{8.673\times 10^5}{773^2}\right)\frac{1}{\bar{v}^2} + \frac{0.002632\times 8.314\times 773 - 3.71}{\bar{v}^3}$$

$$+ \frac{3.71\times 0.000135}{\bar{v}^6} + \frac{1.054\times 10^5}{\bar{v}^3(773)^2}\left(1 + \frac{0.0060}{\bar{v}^2}\right)\exp(-0.0060/\bar{v}^2)$$

The solution of this equation by an equation solver such as EES gives

$\bar{v}_2 = 6.460$ m^3/kmol

Then,

$$v_2 = \frac{\bar{v}_2}{M} = \frac{6.460\,\text{m}^3/\text{kmol}}{28.011\,\text{kg/kmol}} = 0.2306\,\text{m}^3/\text{kg}$$

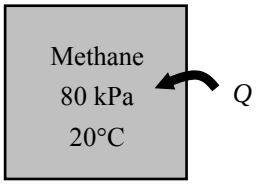
$$V_2 = mv_2 = (0.100\,\text{kg})(0.2306\,\text{m}^3/\text{kg}) = \mathbf{0.02306\ m^3}$$

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3-101 Methane is heated in a rigid container. The final pressure of the methane is to be determined using the ideal gas equation and the Benedict-Webb-Rubin equation of state.

Analysis (*a*) From the ideal gas equation of state,

$$P_2 = P_1 \frac{T_2}{T_1} = (80\ \text{kPa})\frac{573\ \text{K}}{293\ \text{K}} = \mathbf{156.5\ kPa}$$



(*b*) The specific molar volume of the methane is

$$\bar{v}_1 = \bar{v}_2 = \frac{R_u T_1}{P_1} = \frac{(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293\ \text{K})}{80\ \text{kPa}} = 30.45\ \text{m}^3/\text{kmol}$$

Using the coefficients of Table 3-4 for methane and the given data, the Benedict-Webb-Rubin equation of state for state 2 gives

$$P_2 = \frac{R_u T_2}{\bar{v}_2} + \left(B_0 R_u T_2 - A_0 - \frac{C_0}{T_2^2}\right)\frac{1}{\bar{v}^2} + \frac{bR_u T_2 - a}{\bar{v}^3} + \frac{a\alpha}{\bar{v}^6} + \frac{c}{\bar{v}^3 T_2^2}\left(1 + \frac{\gamma}{\bar{v}^2}\right)\exp(-\gamma/\bar{v}^2)$$

$$= \frac{(8.314)(573)}{30.45} + \left(0.04260\times 8.314\times 573 - 187.91 - \frac{2.286\times 10^6}{573^2}\right)\frac{1}{30.45^2} + \frac{0.003380\times 8.314\times 573 - 5.00}{30.45^3}$$

$$+ \frac{5.00\times 1.244\times 10^{-4}}{30.45^6} + \frac{2.578\times 10^5}{30.45^3(573)^2}\left(1 + \frac{0.0060}{30.45^2}\right)\exp(-0.0060/30.45^2)$$

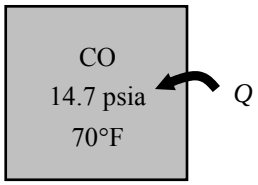
$$= \mathbf{156.5\ kPa}$$

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3-102E Carbon monoxide is heated in a rigid container. The final pressure of the CO is to be determined using the ideal gas equation and the Benedict-Webb-Rubin equation of state.

Properties The gas constant and molar mass of CO are (Table A-1)

$R = 0.2968$ kPa·m^3/kg·K, $M = 28.011$ kg/kmol



Analysis (*a*) From the ideal gas equation of state,

$$P_2 = P_1 \frac{T_2}{T_1} = (14.7 \text{ psia}) \frac{1260 \text{ R}}{530 \text{ R}} = \mathbf{34.95 \text{ psia}}$$

(*b*) The specific molar volume of the CO in SI units is

$$\bar{v}_1 = \bar{v}_2 = \frac{R_u T_1}{P_1} = \frac{(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(294 \text{ K})}{101 \text{ kPa}} = 24.20 \text{ m}^3/\text{kmol}$$

Using the coefficients of Table 3-4 for CO and the given data, the Benedict-Webb-Rubin equation of state for state 2 gives

$$P_2 = \frac{R_u T_2}{\bar{v}_2} + \left(B_0 R_u T_2 - A_0 - \frac{C_0}{T_2^2} \right) \frac{1}{\bar{v}^2} + \frac{bR_u T_2 - a}{\bar{v}^3} + \frac{a\alpha}{\bar{v}^6} + \frac{c}{\bar{v}^3 T_2^2} \left(1 + \frac{\gamma}{\bar{v}^2} \right) \exp(-\gamma / \bar{v}^2)$$

$$= \frac{(8.314)(700)}{24.20} + \left(0.05454 \times 8.314 \times 700 - 135.87 - \frac{8.673 \times 10^5}{700^2} \right) \frac{1}{24.20^2} + \frac{0.002632 \times 8.314 \times 700 - 3.71}{24.20^3}$$

$$+ \frac{3.71 \times 1.350 \times 10^{-4}}{24.20^6} + \frac{1.054 \times 10^5}{24.20^3 (700)^2} \left(1 + \frac{0.0060}{24.20^2} \right) \exp(-0.0060 / 24.20^2)$$

$$= 240.8 \text{ kPa}$$

The pressure in English unit is

$$P_2 = (240.8 \text{ kPa}) \left(\frac{1 \text{ psia}}{6.8948 \text{ kPa}} \right) = \mathbf{34.92 \text{ psia}}$$

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3-103E The temperature of R-134a in a tank at a specified state is to be determined using the ideal gas relation, the van der Waals equation, and the refrigerant tables.

Properties The gas constant, critical pressure, and critical temperature of R-134a are (Table A-1E)

$R = 0.1052$ psia·ft^3/lbm·R, $T_{cr} = 673.6$ R, $P_{cr} = 588.7$ psia

Analysis (a) From the ideal gas equation of state,

$$T = \frac{P\nu}{R} = \frac{(160 \text{ psia})(0.3479 \text{ ft}^3/\text{lbm})}{0.1052 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}} = \mathbf{529\ R}$$

(*b*) The van der Waals constants for the refrigerant are determined from

$$a = \frac{27R^2T_{cr}^2}{64P_{cr}} = \frac{(27)(0.1052 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})^2(673.6 \text{ R})^2}{(64)(588.7 \text{ psia})} = 3.591 \text{ ft}^6 \cdot \text{psia/lbm}^2$$

$$b = \frac{RT_{cr}}{8P_{cr}} = \frac{(0.1052 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(673.6 \text{ R})}{8 \times 588.7 \text{ psia}} = 0.0150 \text{ ft}^3/\text{lbm}$$

Then,

$$T = \frac{1}{R}\left(P + \frac{a}{\nu^2}\right)(\nu - b) = \frac{1}{0.1052}\left(160 + \frac{3.591}{(0.3479)^2}\right)(0.3479 - 0.0150) = \mathbf{600\ R}$$

(c) From the superheated refrigerant table (Table A-13E),

$$\left.\begin{array}{l} P = 160 \text{ psia} \\ \nu = 0.3479 \text{ ft}^3/\text{lbm} \end{array}\right\} T = \mathbf{160°F} \quad (620 \text{ R})$$

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3-104 The pressure of nitrogen in a tank at a specified state is to be determined using the ideal gas relation and the Beattie-Bridgeman equation. The error involved in each case is to be determined.

Properties The gas constant and molar mass of nitrogen are (Table A-1)

$R = 0.2968$ kPa·m^3/kg·K and $M = 28.013$ kg/kmol

N_2
0.041884 m^3/kg
150 K

Analysis (*a*) From the ideal gas equation of state,

$$P = \frac{RT}{v} = \frac{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(150\ \text{K})}{0.041884\ \text{m}^3/\text{kg}} = \mathbf{1063\ kPa}\ \ (6.3\%\ \text{error})$$

(*b*) The constants in the Beattie-Bridgeman equation are

$$A = A_o\left(1-\frac{a}{\bar{v}}\right) = 136.2315\left(1-\frac{0.02617}{1.1733}\right) = 133.193$$

$$B = B_o\left(1-\frac{b}{\bar{v}}\right) = 0.05046\left(1-\frac{-0.00691}{1.1733}\right) = 0.05076$$

$$c = 4.2\times10^4\ \text{m}^3\cdot\text{K}^3/\text{kmol}$$

since

$$\bar{v} = Mv = (28.013\ \text{kg/kmol})(0.041884\ \text{m}^3/\text{kg}) = 1.1733\ \text{m}^3/\text{kmol}.$$

Substituting,

$$P = \frac{R_uT}{\bar{v}^2}\left(1-\frac{c}{\bar{v}T^3}\right)(\bar{v}+B) - \frac{A}{\bar{v}^2} = \frac{8.314\times150}{(1.1733)^2}\left(1-\frac{4.2\times10^4}{1.1733\times150^3}\right)(1.1733+0.05076) - \frac{133.193}{(1.1733)^2}$$

$$= \mathbf{1000.4\ kPa}\ \ (\text{negligible error})$$

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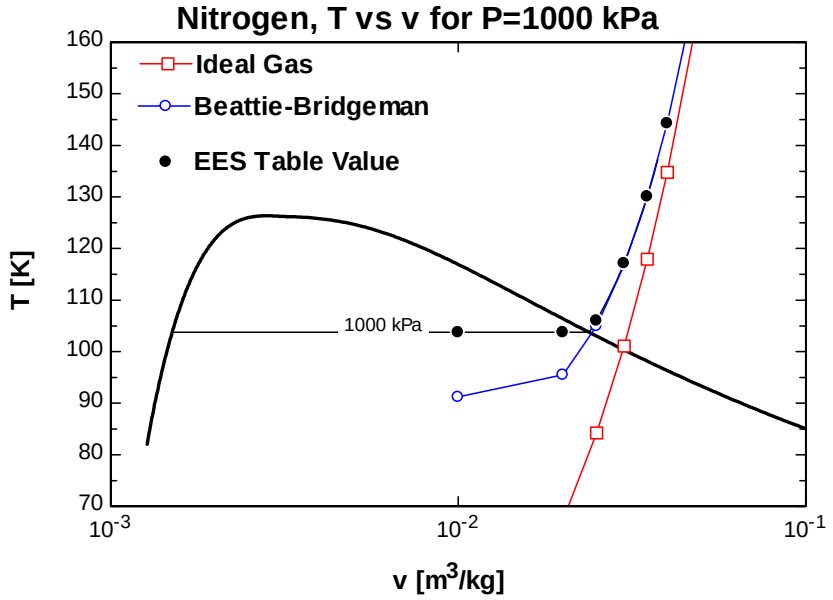
3-105 Problem 3-104 is reconsidered. Using EES (or other) software, the pressure results of the ideal gas and Beattie-Bridgeman equations with nitrogen data supplied by EES are to be compared. The temperature is to be plotted versus specific volume for a pressure of 1000 kPa with respect to the saturated liquid and saturated vapor lines of nitrogen over the range of 110 K < *T* < 150 K.

Analysis The problem is solved using EES, and the solution is given below.

```
Function BeattBridg(T,v,M,R_u)
v_bar=v*M  "Conversion from m^3/kg to m^3/kmol"
"The constants for the Beattie-Bridgeman equation of state are found in text"
Ao=136.2315; aa=0.02617; Bo=0.05046; bb=-0.00691; cc=4.20*1E4
B=Bo*(1-bb/v_bar)
A=Ao*(1-aa/v_bar)
"The Beattie-Bridgeman equation of state is"
BeattBridg:=R_u*T/(v_bar**2)*(1-cc/(v_bar*T**3))*(v_bar+B)-A/v_bar**2
End

T=150  [K]
v=0.041884 [m^3/kg]
P_exper=1000 [kPa]
T_table=T; T_BB=T;T_idealgas=T
P_table=PRESSURE(Nitrogen,T=T_table,v=v)  "EES data for nitrogen as a real gas"
{T_table=temperature(Nitrogen, P=P_table,v=v)}
M=MOLARMASS(Nitrogen)
R_u=8.314 [kJ/kmol-K] "Universal gas constant"
R=R_u/M  "Particular gas constant"
P_idealgas=R*T_idealgas/v  "Ideal gas equation"
P_BB=BeattBridg(T_BB,v,M,R_u) "Beattie-Bridgeman equation of state Function"
```

P_{BB} [kPa]	P_{table} [kPa]	$P_{idealgas}$ [kPa]	v [m3/kg]	T_{BB} [K]	$T_{ideal\ gas}$ [K]	T_{table} [K]
1000	1000	1000	0.01	91.23	33.69	103.8
1000	1000	1000	0.02	95.52	67.39	103.8
1000	1000	1000	0.025	105	84.23	106.1
1000	1000	1000	0.03	116.8	101.1	117.2
1000	1000	1000	0.035	130.1	117.9	130.1
1000	1000	1000	0.04	144.4	134.8	144.3
1000	1000	1000	0.05	174.6	168.5	174.5



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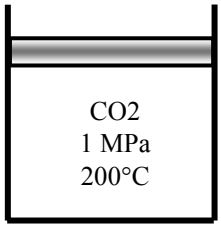
3-106 Carbon dioxide is compressed in a piston-cylinder device in a polytropic process. The final temperature is to be determined using the ideal gas and van der Waals equations.

Properties The gas constant, molar mass, critical pressure, and critical temperature of carbon dioxide are (Table A-1)

$R = 0.1889$ kPa·m^3/kg·K, $M = 44.01$ kg/kmol, $T_{cr} = 304.2$ K, $P_{cr} = 7.39$ MPa

Analysis (*a*) The specific volume at the initial state is

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\,\text{K})}{1000\,\text{kPa}} = 0.08935\,\text{m}^3/\text{kg}$$



According to process specification,

$$\nu_2 = \nu_1\left(\frac{P_1}{P_2}\right)^{1/n} = (0.08935\,\text{m}^3/\text{kg})\left(\frac{1000\,\text{kPa}}{3000\,\text{kPa}}\right)^{1/1.2} = 0.03577\,\text{m}^3/\text{kg}$$

The final temperature is then

$$T_2 = \frac{P_2\nu_2}{R} = \frac{(3000\,\text{kPa})(0.03577\,\text{m}^3/\text{kg})}{0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}} = \mathbf{568\,K}$$

(*b*) The van der Waals constants for carbon dioxide are determined from

$$a = \frac{27R^2T_{cr}^2}{64P_{cr}} = \frac{(27)(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})^2(304.2\,\text{K})^2}{(64)(7390\,\text{kPa})} = 0.1885\,\text{m}^6\cdot\text{kPa/kg}^2$$

$$b = \frac{RT_{cr}}{8P_{cr}} = \frac{(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(304.2\,\text{K})}{8\times 7390\,\text{kPa}} = 0.0009720\,\text{m}^3/\text{kg}$$

Applying the van der Waals equation to the initial state,

$$\left(P + \frac{a}{\nu^2}\right)(\nu - b) = RT$$

$$\left(1000 + \frac{0.1885}{\nu^2}\right)(\nu - 0.0009720) = (0.1889)(473)$$

Solving this equation by trial-error or by EES gives

$$\nu_1 = 0.08821\,\text{m}^3/\text{kg}$$

According to process specification,

$$\nu_2 = \nu_1\left(\frac{P_1}{P_2}\right)^{1/n} = (0.08821\,\text{m}^3/\text{kg})\left(\frac{1000\,\text{kPa}}{3000\,\text{kPa}}\right)^{1/1.2} = 0.03531\,\text{m}^3/\text{kg}$$

Applying the van der Waals equation to the final state,

$$\left(P + \frac{a}{\nu^2}\right)(\nu - b) = RT$$

$$\left(3000 + \frac{0.1885}{0.03531^2}\right)(0.03531 - 0.0009720) = (0.1889)T$$

Solving for the final temperature gives

$$T_2 = \mathbf{573\,K}$$

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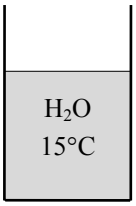
Special Topic: Vapor Pressure and Phase Equilibrium

3-107 A glass of water is left in a room. The vapor pressures at the free surface of the water and in the room far from the glass are to be determined.

Assumptions The water in the glass is at a uniform temperature.

Properties The saturation pressure of water is 2.339 kPa at 20°C, and 1.706 kPa at 15°C (Table A-4).

Analysis The vapor pressure at the water surface is the saturation pressure of water at the water temperature,



$$P_{v,\,\text{water surface}} = P_{\text{sat}\,@\,T_{\text{water}}} = P_{\text{sat@15°C}} = \mathbf{1.706\ kPa}$$

Noting that the air in the room is not saturated, the vapor pressure in the room far from the glass is

$$P_{v,\,\text{air}} = \phi P_{\text{sat}\,@\,T_{\text{air}}} = \phi P_{\text{sat@20°C}} = (0.4)(2.339\ \text{kPa}) = \mathbf{0.936\ kPa}$$

3-108 The vapor pressure in the air at the beach when the air temperature is 30°C is claimed to be 5.2 kPa. The validity of this claim is to be evaluated.

Properties The saturation pressure of water at 30°C is 4.247 kPa (Table A-4).

Analysis The maximum vapor pressure in the air is the saturation pressure of water at the given temperature, which is



$$P_{v,\,\text{max}} = P_{\text{sat}\,@\,T_{\text{air}}} = P_{\text{sat@30°C}} = \mathbf{4.247\ kPa}$$

which is less than the claimed value of 5.2 kPa. Therefore, the claim is **false**.

3-109 The temperature and relative humidity of air over a swimming pool are given. The water temperature of the swimming pool when phase equilibrium conditions are established is to be determined.

Assumptions The temperature and relative humidity of air over the pool remain constant.

Properties The saturation pressure of water at 20°C is 2.339 kPa (Table A-4).

Analysis The vapor pressure of air over the swimming pool is



$$P_{v,\,\text{air}} = \phi P_{\text{sat}\,@\,T_{\text{air}}} = \phi P_{\text{sat@20°C}} = (0.4)(2.339\ \text{kPa}) = 0.9357\ \text{kPa}$$

Phase equilibrium will be established when the vapor pressure at the water surface equals the vapor pressure of air far from the surface. Therefore,

$$P_{v,\,\text{water surface}} = P_{v,\,\text{air}} = 0.9357\ \text{kPa}$$

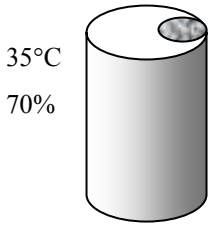
and $$T_{\text{water}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,0.9357\ \text{kPa}} = \mathbf{6.0°C}$$

Discussion Note that the water temperature drops to 6.0°C in an environment at 20°C when phase equilibrium is established.

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3-110 A person buys a supposedly cold drink in a hot and humid summer day, yet no condensation occurs on the drink. The claim that the temperature of the drink is below 10°C is to be evaluated.

Properties The saturation pressure of water at 35°C is 5.629 kPa (Table A-4).



Analysis The vapor pressure of air is

$$P_{v,\text{air}} = \phi P_{\text{sat}@T_{\text{air}}} = \phi P_{\text{sat@35°C}} = (0.7)(5.629\ \text{kPa}) = 3.940\ \text{kPa}$$

The saturation temperature corresponding to this pressure (called the dew-point temperature) is

$$T_{\text{sat}} = T_{\text{sat}@P_v} = T_{\text{sat@3.940 kPa}} = \mathbf{28.7°C}$$

That is, the vapor in the air will condense at temperatures below 28.7°C. Noting that no condensation is observed on the can, the claim that the drink is at 10°C is **false**.

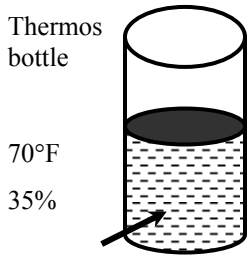
3-111E A thermos bottle half-filled with water is left open to air in a room at a specified temperature and pressure. The temperature of water when phase equilibrium is established is to be determined.

Assumptions The temperature and relative humidity of air over the bottle remain constant.

Properties The saturation pressure of water at 70°F is 0.3633 psia (Table A-4E).

Analysis The vapor pressure of air in the room is

$$P_{v,\text{air}} = \phi P_{\text{sat}@T_{\text{air}}} = \phi P_{\text{sat@70°F}} = (0.35)(0.3633\ \text{psia}) = 0.1272\ \text{psia}$$



Phase equilibrium will be established when the vapor pressure at the water surface equals the vapor pressure of air far from the surface. Therefore,

$$P_{v,\text{water surface}} = P_{v,\text{air}} = 0.1272\ \text{psia}$$

and

$$T_{\text{water}} = T_{\text{sat}@P_v} = T_{\text{sat}@0.1272\ \text{psia}} = \mathbf{41.1°F}$$

Discussion Note that the water temperature drops to 41°F in an environment at 70°F when phase equilibrium is established.

3-112 Two rooms are identical except that they are maintained at different temperatures and relative humidities. The room that contains more moisture is to be determined.

Properties The saturation pressure of water is 2.339 kPa at 20°C, and 3.17 kPa at 25°C (Table A-4).

Analysis The vapor pressures in the two rooms are

Room 1: $$P_{v1} = \phi_1 P_{\text{sat}@T_1} = \phi_1 P_{\text{sat@25°C}} = (0.4)(3.17\ \text{kPa}) = \mathbf{1.27\ kPa}$$

Room 2: $$P_{v2} = \phi_2 P_{\text{sat}@T_2} = \phi_2 P_{\text{sat@20°C}} = (0.55)(2.339\ \text{kPa}) = \mathbf{1.29\ kPa}$$

Therefore, room 1 at 30°C and 40% relative humidity contains more moisture.

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Review Problems

3-113 Nitrogen gas in a rigid tank is heated to a final gage pressure. The final temperature is to be determined.

Assumptions At specified conditions, nitrogen behaves as an ideal gas.

Analysis According to the ideal gas equation of state at constant volume,

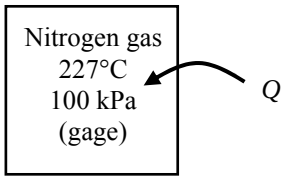
$$m_1 = m_1$$

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

Since $V_1 = V_2$

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

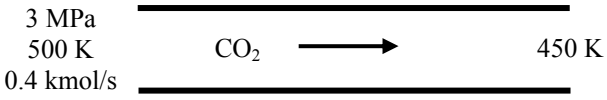
$$T_2 = T_1 \frac{P_2}{P_2} = [(227 + 273)\text{ K}]\frac{(250 + 100)\text{ kPa}}{(100 + 100)\text{ kPa}} = 875\text{ K} = \mathbf{602°C}$$



$P_{atm} = 100$ kPa

3-114 Carbon dioxide flows through a pipe at a given state. The volume and mass flow rates and the density of CO_2 at the given state and the volume flow rate at the exit of the pipe are to be determined.

Analysis



(*a*) The volume and mass flow rates may be determined from ideal gas relation as

$$\dot{V}_1 = \frac{\dot{N} R_u T_1}{P} = \frac{(0.4\text{ kmol/s})(8.314\text{ kPa.m}^3\text{/kmol.K})(500\text{ K})}{3000\text{ kPa}} = \mathbf{0.5543\ m^3/s}$$

$$\dot{m}_1 = \frac{P_1 \dot{V}_1}{R T_1} = \frac{(3000\text{ kPa})(0.5543\text{ m}^3/s)}{(0.1889\text{ kPa.m}^3\text{/kg.K})(500\text{ K})} = \mathbf{17.60\ kg/s}$$

The density is

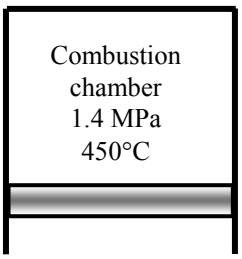
$$\rho_1 = \frac{\dot{m}_1}{\dot{V}_1} = \frac{(17.60\text{ kg/s})}{(0.5543\text{ m}^3\text{/s})} = \mathbf{31.76\ kg/m^3}$$

(b) The volume flow rate at the exit is

$$\dot{V}_2 = \frac{\dot{N} R_u T_2}{P} = \frac{(0.4\text{ kmol/s})(8.314\text{ kPa.m}^3\text{/kmol.K})(450\text{ K})}{3000\text{ kPa}} = \mathbf{0.4988\ m^3/s}$$

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3-115 The cylinder conditions before the heat addition process is specified. The pressure after the heat addition process is to be determined.

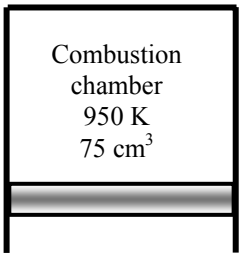


Assumptions **1** The contents of cylinder are approximated by the air properties. **2** Air is an ideal gas.

Analysis The final pressure may be determined from the ideal gas relation

$$P_2 = \frac{T_2}{T_1} P_1 = \left(\frac{1600 + 273\,\text{K}}{450 + 273\,\text{K}}\right)(1400\,\text{kPa}) = \mathbf{3627\ kPa}$$

3-116 The cylinder conditions before the heat addition process is specified. The temperature after the heat addition process is to be determined.



Assumptions **1** The contents of cylinder is approximated by the air properties. **2** Air is an ideal gas.

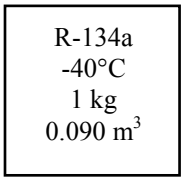
Analysis The ratio of the initial to the final mass is

$$\frac{m_1}{m_2} = \frac{\text{AF}}{\text{AF} + 1} = \frac{22}{22 + 1} = \frac{22}{23}$$

The final temperature may be determined from ideal gas relation

$$T_2 = \frac{m_1}{m_2}\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} T_1 = \left(\frac{22}{23}\right)\left(\frac{150\,\text{cm}^3}{75\,\text{cm}^3}\right)(950\,\text{K}) = \mathbf{1817\ K}$$

3-117 A rigid container that is filled with R-13a is heated. The initial pressure and the final temperature are to be determined.

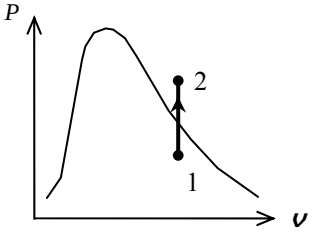


Analysis The initial specific volume is 0.090 m^3/kg. Using this with the initial temperature reveals that the initial state is a mixture. The initial pressure is then the saturation pressure,

$$\left.\begin{array}{l} T_1 = -40°\text{C} \\ \boldsymbol{v}_1 = 0.090\,\text{m}^3/\text{kg} \end{array}\right\} P_1 = P_{\text{sat @ -40°C}} = \mathbf{51.25\ kPa} \quad \text{(Table A-11)}$$

This is a constant volume cooling process ($\boldsymbol{v} = \boldsymbol{V}/m$ = constant). The final state is superheated vapor and the final temperature is then

$$\left.\begin{array}{l} P_2 = 280\,\text{kPa} \\ \boldsymbol{v}_2 = \boldsymbol{v}_1 = 0.090\,\text{m}^3/\text{kg} \end{array}\right\} T_2 = \mathbf{50°C} \quad \text{(Table A-13)}$$

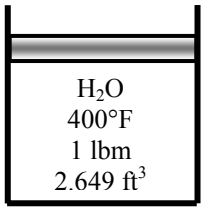


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3-118E A piston-cylinder device that is filled with water is cooled. The final pressure and volume of the water are to be determined.

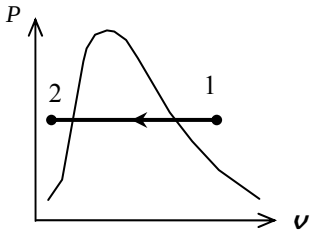
Analysis The initial specific volume is

$$v_1 = \frac{V_1}{m} = \frac{2.649\ \text{ft}^3}{1\ \text{lbm}} = 2.649\ \text{ft}^3/\text{lbm}$$



This is a constant-pressure process. The initial state is determined to be superheated vapor and thus the pressure is determined to be

$$\left.\begin{array}{l} T_1 = 400°\text{F} \\ v_1 = 2.649\ \text{ft}^3/\text{lbm} \end{array}\right\} \quad P_1 = P_2 = \mathbf{180\ psia} \quad \text{(Table A - 6E)}$$



The saturation temperature at 180 psia is 373.1°F. Since the final temperature is less than this temperature, the final state is compressed liquid. Using the incompressible liquid approximation,

$$v_2 = v_{f\,@\,100°\text{F}} = 0.01613\ \text{ft}^3/\text{lbm} \quad \text{(Table A - 4E)}$$

The final volume is then

$$V_2 = mv_2 = (1\,\text{lbm})(0.01613\,\text{ft}^3/\text{lbm}) = \mathbf{0.01613\ ft^3}$$

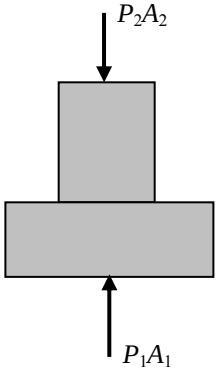
3-119 The volume of chamber 1 of the two-piston cylinder shown in the figure is to be determined.

Assumptions At specified conditions, helium behaves as an ideal gas.

Properties The gas constant of helium is $R = 2.0769$ kJ/kg·K (Table A-1).

Analysis Since the water vapor in chamber 2 is condensing, the pressure in this chamber is the saturation pressure,

$$P_2 = P_{\text{sat}\,@\,200°\text{C}} = 1555\,\text{kPa} \quad \text{(Table A-4)}$$



Summing the forces acting on the piston in the vertical direction gives

$$P_1 = P_2 \frac{A_2}{A_1} = P_2\left(\frac{D_2}{D_1}\right)^2 = (1555\,\text{kPa})\left(\frac{4}{10}\right)^2 = 248.8\,\text{kPa}$$

According to the ideal gas equation of state,

$$V_1 = \frac{mRT}{P_1} = \frac{(1\,\text{kg})(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(200+273\,\text{K})}{248.8\,\text{kPa}} = \mathbf{3.95\ m^3}$$

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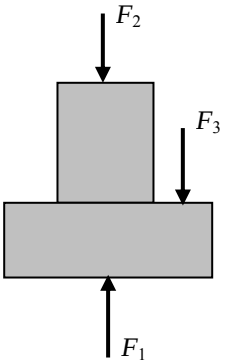
3-120E The volume of chamber 1 of the two-piston cylinder shown in the figure is to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.3704$ psia·ft^3/lbm·R (Table A-1E).

Analysis Since R-134a in chamber 2 is condensing, the pressure in this chamber is the saturation pressure,

$$P_2 = P_{\text{sat}@120°\text{F}} = 186.0 \text{ psia} \quad \text{(Table A-11E)}$$



Summing the forces acting on the piston in the vertical direction gives

$$F_2 + F_3 = F_1$$
$$P_2 A_2 + P_3 (A_1 - A_2) = P_1 A_1$$

which when solved for P_1 gives

$$P_1 = P_2 \frac{A_2}{A_1} + P_3 \left(1 - \frac{A_2}{A_1}\right)$$

since the areas of the piston faces are given by $A = \pi D^2 / 4$ the above equation becomes

$$P_1 = P_2 \left(\frac{D_2}{D_1}\right)^2 + P_3 \left[1 - \left(\frac{D_2}{D_1}\right)^2\right]$$
$$= (186.0 \text{ psia}) \left(\frac{2}{3}\right)^2 + (30 \text{ psia}) \left[1 - \left(\frac{2}{3}\right)^2\right]$$
$$= 99.33 \text{ psia}$$

According to the ideal gas equation of state,

$$V_1 = \frac{mRT}{P_1} = \frac{(0.5 \text{ lbm})(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(120 + 460 \text{ R})}{99.33 \text{ psia}} = \mathbf{1.08 \text{ ft}^3}$$

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3-121E The difference in the volume of chamber 1 for two cases of pressure in chamber 3 is to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

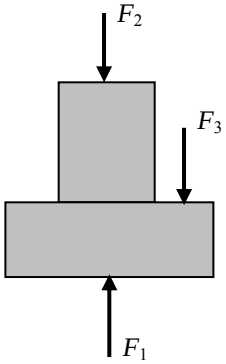
Properties The gas constant of air is $R = 0.3704$ psia·ft^3/lbm·R (Table A-1).

Analysis Since R-134a in chamber 2 is condensing, the pressure in this chamber is the saturation pressure,

$$P_2 = P_{\text{sat}@120°\text{F}} = 186.0 \text{ psia} \quad \text{(Table A-11E)}$$

Summing the forces acting on the piston in the vertical direction gives

$$F_2 + F_3 = F_1$$
$$P_2 A_2 + P_3 (A_1 - A_2) = P_1 A_1$$



which when solved for P_1 gives

$$P_1 = P_2 \frac{A_2}{A_1} + P_3 \left(1 - \frac{A_2}{A_1}\right)$$

since the areas of the piston faces are given by $A = \pi D^2 / 4$ the above equation becomes

$$P_1 = P_2 \left(\frac{D_2}{D_1}\right)^2 + P_3 \left[1 - \left(\frac{D_2}{D_1}\right)^2\right]$$
$$= (186.0 \text{ psia}) \left(\frac{2}{3}\right)^2 + (60 \text{ kPa}) \left[1 - \left(\frac{2}{3}\right)^2\right]$$
$$= 116 \text{ psia}$$

According to the ideal gas equation of state,

$$V_1 = \frac{mRT}{P_1} = \frac{(0.5 \text{ lbm})(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(120 + 460 \text{ R})}{116 \text{ psia}} = 0.926 \text{ ft}^3$$

For a chamber 3 pressure of 30 psia, the volume of chamber 1 was determined to be 1.08 ft^3. Then the change in the volume of chamber 1 is

$$\Delta V = V_2 - V_1 = 1.08 - 0.926 = \mathbf{0.154 \text{ ft}^3}$$

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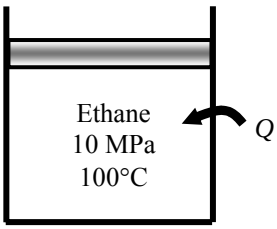
3-122 Ethane is heated at constant pressure. The final temperature is to be determined using ideal gas equation and the compressibility charts.

Properties The gas constant, the critical pressure, and the critical temperature of ethane are, from Table A-1,

$R = 0.2765$ kPa·m^3/kg·K, $T_{cr} = 305.5$ K, $P_{cr} = 4.48$ MPa

Analysis From the ideal gas equation,

$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (373\text{ K})(1.6) = \mathbf{596.8\ K}$$



From the compressibility chart at the initial state (Fig. A-15),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{373\text{ K}}{305.5\text{ K}} = 1.221 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{10\text{ MPa}}{4.48\text{ MPa}} = 2.232 \end{aligned}\right\} \quad Z_1 = 0.61,\ \nu_{R1} = 0.35$$

At the final state,

$$\left.\begin{aligned} P_{R2} &= P_{R1} = 2.232 \\ \nu_{R2} &= 1.6\nu_{R1} = 1.6(0.35) = 0.56 \end{aligned}\right\} \quad Z_2 = 0.83$$

Thus,

$$T_2 = \frac{P_2\nu_2}{Z_2 R} = \frac{P_2}{Z_2}\frac{\nu_{R2}T_{cr}}{P_{cr}} = \frac{10{,}000\text{ kPa}}{0.83}\frac{(0.56)(305.5\text{ K})}{4480\text{ kPa}} = \mathbf{460\ K}$$

Of these two results, the accuracy of the second result is limited by the accuracy with which the charts may be read. Accepting the error associated with reading charts, the second temperature is the more accurate.

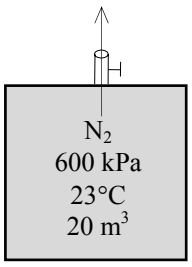
3-123 A large tank contains nitrogen at a specified temperature and pressure. Now some nitrogen is allowed to escape, and the temperature and pressure of nitrogen drop to new values. The amount of nitrogen that has escaped is to be determined.

Properties The gas constant for nitrogen is 0.2968 kPa·m^3/kg·K (Table A-1).

Analysis Treating N_2 as an ideal gas, the initial and the final masses in the tank are determined to be

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(600\text{ kPa})(20\text{ m}^3)}{(0.2968\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\text{ K})} = 136.6\text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(400\text{ kPa})(20\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 92.0\text{ kg}$$



Thus the amount of N_2 that escaped is

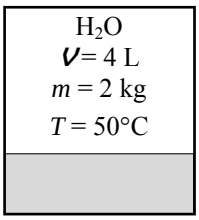
$$\Delta m = m_1 - m_2 = 136.6 - 92.0 = \mathbf{44.6\ kg}$$

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3-124 The rigid tank contains saturated liquid-vapor mixture of water. The mixture is heated until it exists in a single phase. For a given tank volume, it is to be determined if the final phase is a liquid or a vapor.

Analysis This is a constant volume process ($v = V/m$ = constant), and thus the final specific volume will be equal to the initial specific volume,

$$v_2 = v_1$$



The critical specific volume of water is 0.003106 m^3/kg. Thus if the final specific volume is smaller than this value, the water will exist as a liquid, otherwise as a vapor.

$$V = 4L \longrightarrow v = \frac{V}{m} = \frac{0.004\ \text{m}^3}{2\ \text{kg}} = 0.002\ \text{m}^3/\text{kg} < v_{cr}$$ Thus, **liquid**.

$$V = 400L \longrightarrow v = \frac{V}{m} = \frac{0.4\ \text{m}^3}{2\ \text{kg}} = 0.2\ \text{m}^3/\text{kg} > v_{cr}.$$ Thus, **vapor**.

3-125 Two rigid tanks that contain hydrogen at two different states are connected to each other. Now a valve is opened, and the two gases are allowed to mix while achieving thermal equilibrium with the surroundings. The final pressure in the tanks is to be determined.

Properties The gas constant for hydrogen is 4.124 kPa·m^3/kg·K (Table A-1).

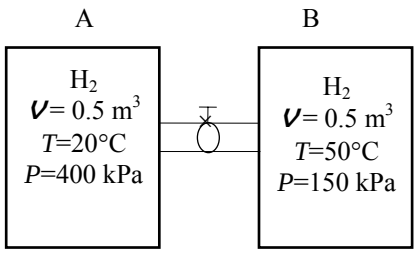
Analysis Let's call the first and the second tanks A and B. Treating H_2 as an ideal gas, the total volume and the total mass of H_2 are

$$V = V_A + V_B = 0.5 + 0.5 = 1.0\ \text{m}^3$$

$$m_A = \left(\frac{P_1 V}{RT_1}\right)_A = \frac{(400\ \text{kPa})(0.5\ \text{m}^3)}{(4.124\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 0.1655\ \text{kg}$$

$$m_B = \left(\frac{P_1 V}{RT_1}\right)_B = \frac{(150\ \text{kPa})(0.5\ \text{m}^3)}{(4.124\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\ \text{K})} = 0.0563\ \text{kg}$$

$$m = m_A + m_B = 0.1655 + 0.0563 = 0.2218\ \text{kg}$$



Then the final pressure can be determined from

$$P = \frac{mRT_2}{V} = \frac{(0.2218\ \text{kg})(4.124\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{1.0\ \text{m}^3} = \mathbf{264\ kPa}$$

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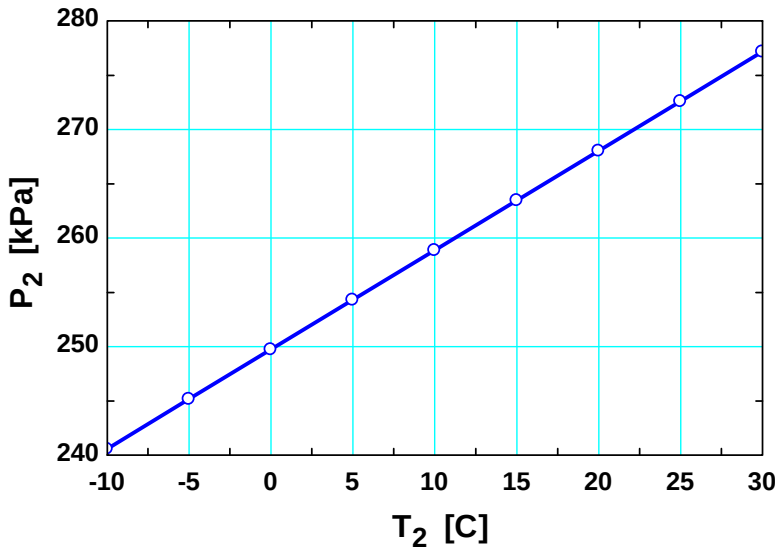
3-126 Problem 3-125 is reconsidered. The effect of the surroundings temperature on the final equilibrium pressure in the tanks is to be investigated. The final pressure in the tanks is to be plotted versus the surroundings temperature, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given Data"
V_A=0.5 [m^3]
T_A=20 [C]
P_A=400 [kPa]
V_B=0.5 [m^3]
T_B=50 [C]
P_B=150 [kPa]
{T_2=15 [C]}

"Solution"
R=R_u/MOLARMASS(H2)
R_u=8.314 [kJ/kmol-K]
V_total=V_A+V_B
m_total=m_A+m_B
P_A*V_A=m_A*R*(T_A+273)
P_B*V_B=m_B*R*(T_B+273)
P_2*V_total=m_total*R*(T_2+273)
```

P_2 [kPa]	T_2 [C]
240.6	-10
245.2	-5
249.7	0
254.3	5
258.9	10
263.5	15
268	20
272.6	25
277.2	30



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3-127 The pressure in an automobile tire increases during a trip while its volume remains constant. The percent increase in the absolute temperature of the air in the tire is to be determined.

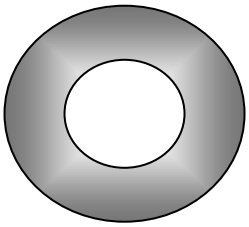
Assumptions **1** The volume of the tire remains constant. **2** Air is an ideal gas.

Properties The local atmospheric pressure is 90 kPa.

Analysis The absolute pressures in the tire before and after the trip are

$$P_1 = P_{\text{gage},1} + P_{\text{atm}} = 200 + 90 = 290 \text{ kPa}$$
$$P_2 = P_{\text{gage},2} + P_{\text{atm}} = 220 + 90 = 310 \text{ kPa}$$

TIRE
200 kPa
0.035 m^3



Noting that air is an ideal gas and the volume is constant, the ratio of absolute temperatures after and before the trip are

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \rightarrow \frac{T_2}{T_1} = \frac{P_2}{P_1} = \frac{310 \text{ kPa}}{290 \text{ kPa}} = 1.069$$

Therefore, the absolute temperature of air in the tire will increase by **6.9%** during this trip.

3-128 The temperature of steam in a tank at a specified state is to be determined using the ideal gas relation, the generalized chart, and the steam tables.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \quad T_{\text{cr}} = 647.1 \text{ K}, \quad P_{\text{cr}} = 22.06 \text{ MPa}$$

Analysis (*a*) From the ideal gas equation of state,

$$P = \frac{RT}{v} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(673 \text{ K})}{0.02 \text{ m}^3/\text{kg}} = \mathbf{15{,}529 \text{ kPa}}$$

H_2O
0.02 m^3/kg
400°C

(*b*) From the compressibility chart (Fig. A-15a),

$$\left.\begin{aligned} T_R &= \frac{T}{T_{\text{cr}}} = \frac{673 \text{ K}}{647.1 \text{ K}} = 1.040 \\ v_R &= \frac{v_{\text{actual}}}{RT_{\text{cr}}/P_{\text{cr}}} = \frac{(0.02 \text{ m}^3/\text{kg})(22{,}060 \text{ kPa})}{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(647.1 \text{ K})} = 1.48 \end{aligned}\right\} P_R = 0.57$$

Thus,

$$P = P_R P_{\text{cr}} = 0.57 \times 22{,}060 = \mathbf{12{,}574 \text{ kPa}}$$

(*c*) From the superheated steam table,

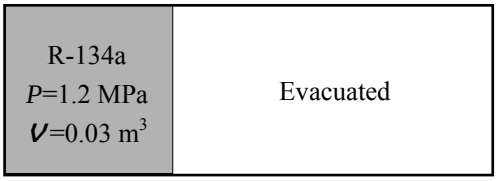
$$\left.\begin{aligned} T &= 400°\text{C} \\ v &= 0.02 \text{ m}^3/\text{kg} \end{aligned}\right\} P = \mathbf{12{,}515 \text{ kPa}} \quad \text{(from EES)}$$

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3-129 One section of a tank is filled with saturated liquid R-134a while the other side is evacuated. The partition is removed, and the temperature and pressure in the tank are measured. The volume of the tank is to be determined.

Analysis The mass of the refrigerant contained in the tank is

$$m = \frac{V_1}{v_1} = \frac{0.03 \text{ m}^3}{0.0008934 \text{ m}^3/\text{kg}} = 33.58 \text{ kg}$$



since

$$v_1 = v_{f@1.4 \text{ MPa}} = 0.0008934 \text{ m}^3/\text{kg}$$

At the final state (Table A-13),

$$\left.\begin{array}{l} P_2 = 400 \text{ kPa} \\ T_2 = 30°\text{C} \end{array}\right\} \; v_2 = 0.05680 \text{ m}^3/\text{kg}$$

Thus,

$$V_{\text{tank}} = V_2 = mv_2 = (33.58 \text{ kg})(0.05680 \text{ m}^3/\text{kg}) = \mathbf{1.91 \text{ m}^3}$$

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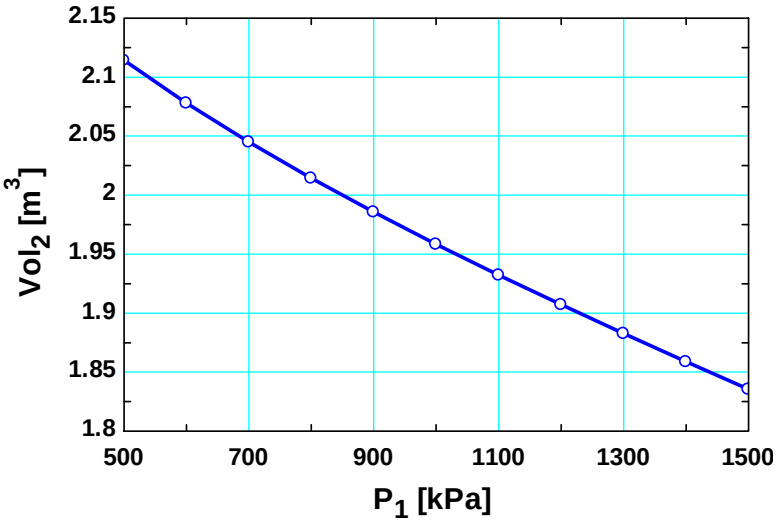
3-130 Problem 3-129 is reconsidered. The effect of the initial pressure of refrigerant-134 on the volume of the tank is to be investigated as the initial pressure varies from 0.5 MPa to 1.5 MPa. The volume of the tank is to be plotted versus the initial pressure, and the results are to be discussed.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given Data"
x_1=0.0
Vol_1=0.03 [m^3]
P_1=1200 [kPa]
T_2=30 [C]
P_2=400 [kPa]

"Solution"
v_1=volume(R134a,P=P_1,x=x_1)
Vol_1=m*v_1
v_2=volume(R134a,P=P_2,T=T_2)
Vol_2=m*v_2
```

P_1 [kPa]	Vol_2 [m^3]	m [kg]
500	2.114	37.23
600	2.078	36.59
700	2.045	36.01
800	2.015	35.47
900	1.986	34.96
1000	1.958	34.48
1100	1.932	34.02
1200	1.907	33.58
1300	1.883	33.15
1400	1.859	32.73
1500	1.836	32.32



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3-131 A propane tank contains 5 L of liquid propane at the ambient temperature. Now a leak develops at the top of the tank and propane starts to leak out. The temperature of propane when the pressure drops to 1 atm and the amount of heat transferred to the tank by the time the entire propane in the tank is vaporized are to be determined.

Properties The properties of propane at 1 atm are T_{sat} = -42.1°C, $\rho = 581\ \text{kg/m}^3$, and h_{fg} = 427.8 kJ/kg (Table A-3).

Analysis The temperature of propane when the pressure drops to 1 atm is simply the saturation pressure at that temperature,

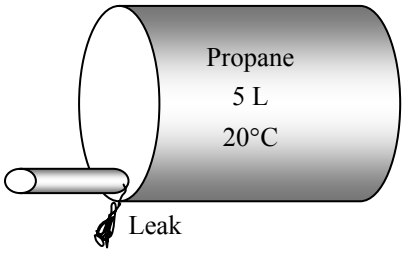
$$T = T_{\text{sat}@1\,\text{atm}} = \mathbf{-42.1^\circ C}$$

The initial mass of liquid propane is

$$m = \rho V = (581\,\text{kg/m}^3)(0.005\,\text{m}^3) = 2.905\,\text{kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

$$Q_{\text{absorbed}} = mh_{fg} = (2.905\ \text{kg})(427.8\ \text{kJ/kg}) = \mathbf{1243\ kJ}$$



3-132 An isobutane tank contains 5 L of liquid isobutane at the ambient temperature. Now a leak develops at the top of the tank and isobutane starts to leak out. The temperature of isobutane when the pressure drops to 1 atm and the amount of heat transferred to the tank by the time the entire isobutane in the tank is vaporized are to be determined.

Properties The properties of isobutane at 1 atm are T_{sat} = -11.7°C, $\rho = 593.8\ \text{kg/m}^3$, and h_{fg} = 367.1 kJ/kg (Table A-3).

Analysis The temperature of isobutane when the pressure drops to 1 atm is simply the saturation pressure at that temperature,

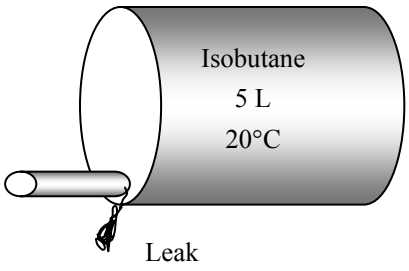
$$T = T_{\text{sat}@1\,\text{atm}} = \mathbf{-11.7^\circ C}$$

The initial mass of liquid isobutane is

$$m = \rho V = (593.8\,\text{kg/m}^3)(0.005\,\text{m}^3) = 2.969\text{kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

$$Q_{\text{absorbed}} = mh_{fg} = (2.969\ \text{kg})(367.1\ \text{kJ/kg}) = \mathbf{1090\ kJ}$$



3-133 A tank contains helium at a specified state. Heat is transferred to helium until it reaches a specified temperature. The final gage pressure of the helium is to be determined.

Assumptions **1** Helium is an ideal gas.

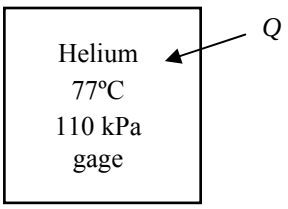
Properties The local atmospheric pressure is given to be 100 kPa.

Analysis Noting that the specific volume of helium in the tank remains constant, from ideal gas relation, we have

$$P_2 = P_1 \frac{T_2}{T_1} = (110 + 100\ \text{kPa})\frac{(300+273)\text{K}}{(77+273)\text{K}} = 343.8\ \text{kPa}$$

Then the gage pressure becomes

$$P_{\text{gage},2} = P_2 - P_{\text{atm}} = 343.8 - 100 = \mathbf{244\ kPa}$$



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3-134 The first eight virial coefficients of a Benedict-Webb-Rubin gas are to be obtained.

Analysis The Benedict-Webb-Rubin equation of state is given by

$$P = \frac{R_u T}{\bar{v}} + \left(B_0 R_u T - A_0 - \frac{C_0}{T^2} \right) \frac{1}{\bar{v}^2} + \frac{bR_u T - a}{\bar{v}^3} + \frac{a\alpha}{\bar{v}^6} + \frac{c}{\bar{v}^3 T^2} \left(1 + \frac{\gamma}{\bar{v}^2} \right) \exp(-\gamma / \bar{v}^2)$$

Expanding the last term in a series gives

$$\exp(-\gamma / \bar{v}^2) = 1 - \frac{\gamma}{\bar{v}^2} + \frac{1}{2!} \frac{\gamma^2}{\bar{v}^4} - \frac{1}{3!} \frac{\gamma^3}{\bar{v}^6} + ...$$

Substituting this into the Benedict-Webb-Rubin equation of state and rearranging the first terms gives

$$P = \frac{R_u T}{\bar{v}} + \frac{R_u T B_0 - A_0 - C_0 / T^2}{\bar{v}^2} + \frac{bR_u T - a}{\bar{v}^3} + \frac{c(1+\gamma)}{\bar{v}^5 T^2} + \frac{a\alpha}{\bar{v}^6} - \frac{c\gamma(1+\gamma)}{\bar{v}^7 T^2} + \frac{1}{2!} \frac{c\gamma^2 (1+\gamma)}{\bar{v}^9 T^2}$$

The virial equation of state is

$$P = \frac{R_u T}{\bar{v}} + \frac{a(T)}{\bar{v}^2} + \frac{b(T)}{\bar{v}^3} + \frac{c(T)}{\bar{v}^4} + \frac{d(T)}{\bar{v}^5} + \frac{e(T)}{\bar{v}^6} + \frac{f(T)}{\bar{v}^7} + \frac{g(T)}{\bar{v}^8} + \frac{h(T)}{\bar{v}^9} ...$$

Comparing the Benedict-Webb-Rubin equation of state to the virial equation of state, the virial coefficients are

$$a(T) = R_u T B_0 - A_0 - C_0 / T^2$$
$$b(T) = bR_u T - a$$
$$c(T) = 0$$
$$d(T) = c(1+\gamma) / T^2$$
$$e(T) = a\alpha$$
$$f(T) = c\gamma(1+\gamma) / T^2$$
$$g(T) = 0$$
$$h(T) = \frac{1}{2!} \frac{c\gamma^2 (1+\gamma)}{T^2}$$

3-135 The table is completed as follows:

P, kPa	T, °C	v, m³/kg	u, kJ/kg	Condition description and quality, if applicable
300	250	**0.7921**	2728.9	**Superheated vapor**
300	**133.52**	**0.3058**	1560.0	***x* = 0.504, Two-phase mixture**
101.42	100	-	-	**Insufficient information**
3000	180	**0.001127***	**761.92***	**Compressed liquid**

* Approximated as saturated liquid at the given temperature of 180°C

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3-136 The table is completed as follows:

P, kPa	T, °C	v, m³/kg	u, kJ/kg	Condition description and quality, if applicable
200	**120.2**	**0.8858**	2529.1	**Saturated vapor**
232.23	125	**0.5010**	**1831.0**	***x* = 0.650, Two-phase mixture**
7829	400	**0.0352**	2967.2	**Superheated vapor**
1000	30	**0.001004***	**125.73***	**Compressed liquid**
120.90	105	-	-	**Insufficient information**

* Approximated as saturated liquid at the given temperature of 30°C

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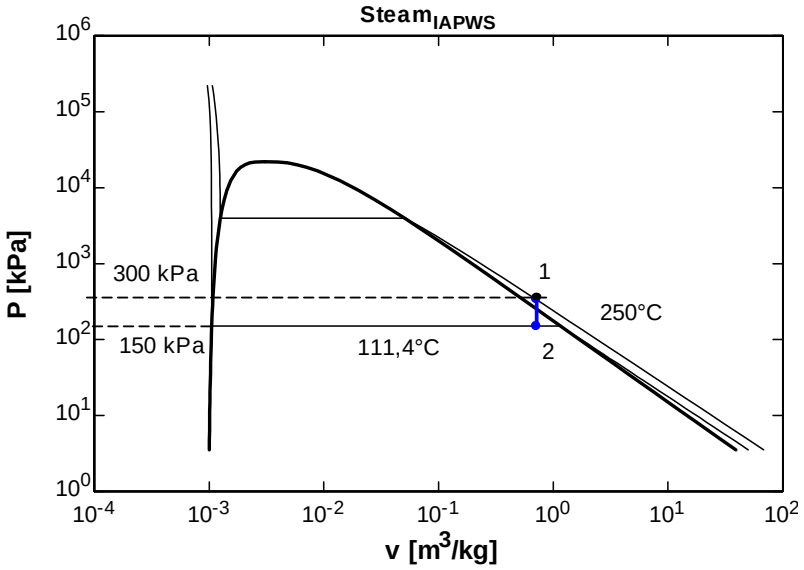
3-137 Water at a specified state is contained in a tank. It is now cooled. The process will be indicated on the P-v and T- v diagrams.

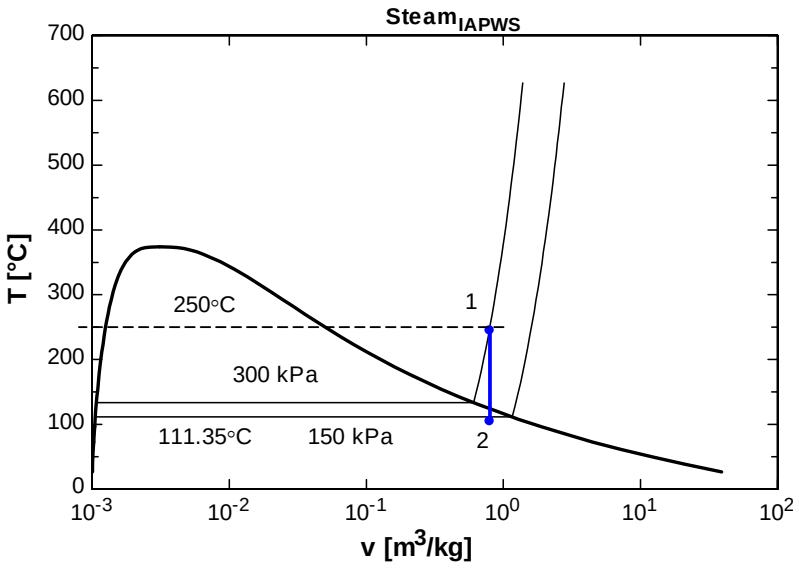
Analysis The properties at the initial and final states are

$$\left.\begin{array}{l} P_1 = 300\ \text{kPa} \\ T_1 = 250°\text{C} \end{array}\right\} v_1 = 0.7964\ \text{m}^3/\text{kg} \quad \text{(Table A - 6)}$$

$$\left.\begin{array}{l} P_2 = 150\ \text{kPa} \\ v_2 = v_1 = 0.7964\ \text{m}^3/\text{kg} \end{array}\right\} T_2 = 111.35°\text{C} \quad \text{(Table A - 5)}$$

Using Property Plot feature of EES, and by adding state points we obtain following diagrams.





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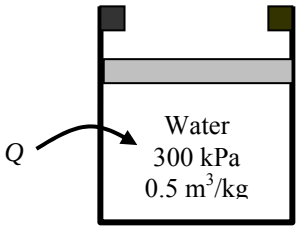
3-138 Water at a specified state is contained in a piston-cylinder device fitted with stops. Water is now heated until a final pressure. The process will be indicated on the P-v and T- v diagrams.

Analysis The properties at the three states are

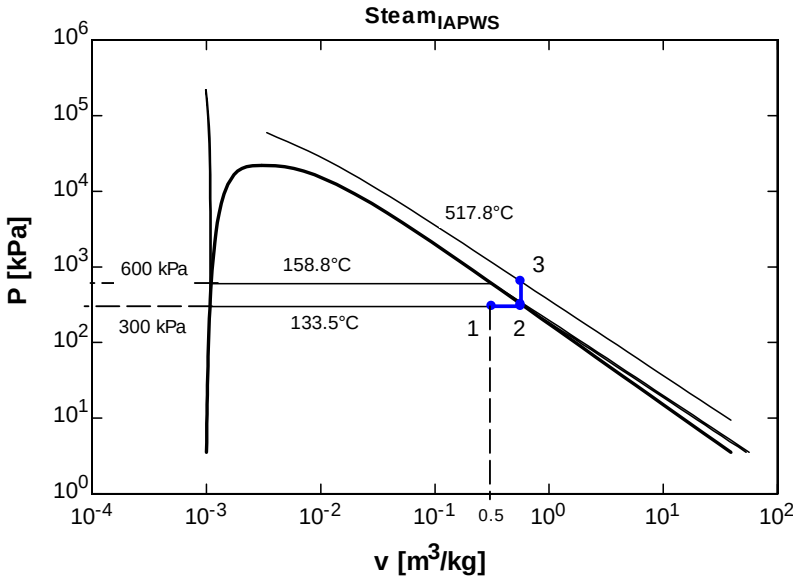
$$\left.\begin{matrix} P_1 = 300\ \text{kPa} \\ v_1 = 0.5\ \text{m}^3/\text{kg} \end{matrix}\right\} T_1 = 133.5°\text{C} \quad \text{(Table A - 5)}$$

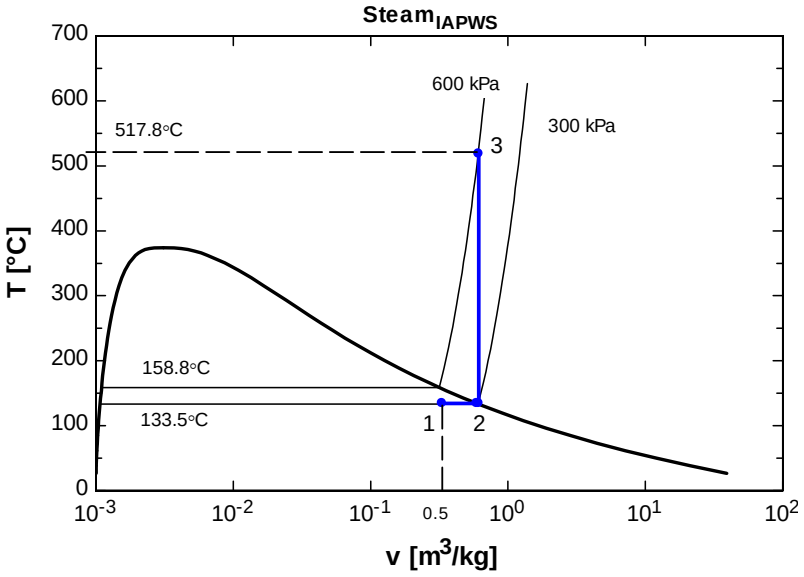
$$\left.\begin{matrix} P_2 = 300\ \text{kPa} \\ x_2 = 1\ (\text{sat. vap.}) \end{matrix}\right\} v_2 = 0.6058\ \text{m}^3/\text{kg}, T_2 = 133.5°\text{C} \quad \text{(Table A - 5)}$$

$$\left.\begin{matrix} P_2 = 600\ \text{kPa} \\ v_3 = 0.6058\ \text{m}^3/\text{kg} \end{matrix}\right\} T_2 = 517.8°\text{C} \quad \text{(Table A - 6)}$$



Using Property Plot feature of EES, and by adding state points we obtain following diagrams.



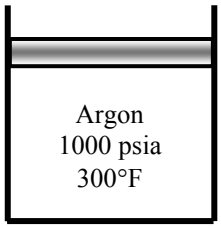


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3-139E Argon contained in a piston-cylinder device at a given state undergoes a polytropic process. The final temperature is to be determined using the ideal gas relation and the Beattie-Bridgeman equation.

Analysis (*a*) The polytropic relations for an ideal gas give

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{n-1/n} = (300 + 460\ \text{R})\left(\frac{2000\ \text{psia}}{1000\ \text{psia}}\right)^{0.6/1.6} = \mathbf{986\ R}$$



(*b*) The constants in the Beattie-Bridgeman equation are expressed as

$$A = A_o\left(1 - \frac{a}{\bar{v}}\right) = 130.7802\left(1 - \frac{0.02328}{\bar{v}}\right)$$

$$B = B_o\left(1 - \frac{b}{\bar{v}}\right) = 0.03931\left(1 - \frac{0}{\bar{v}}\right)$$

$$c = 5.99 \times 10^4\ \text{m}^3 \cdot \text{K}^3/\text{kmol}$$

Substituting these coefficients into the Beattie-Bridgeman equation and using data in SI units (P = 1000 psia = 6895 kPa, T=760 R = 422.2 K, R_u = 8.314 kJ/kmol·K)

$$P = \frac{R_u T}{\bar{v}^2}\left(1 - \frac{c}{\bar{v}T^3}\right)(\bar{v} + B) - \frac{A}{\bar{v}^2}$$

and solving using an equation solver such as EES gives

$$\bar{v} = 0.5120\ \text{m}^3/\text{kmol} = 8.201\ \text{ft}^3/\text{lbmol}$$

From the polytropic equation

$$\bar{v}_2 = \bar{v}_1\left(\frac{P_1}{P_2}\right)^{1/n} = (0.5120\ \text{m}^3/\text{kmol})\left(\frac{1}{2}\right)^{1/1.6} = 0.3319\ \text{m}^3/\text{kmol}$$

Substituting this value into the Beattie-Bridgeman equation and using data in SI units (P = 2000 psia = 13790 kPa and R_u = 8.314 kJ/kmol·K),

$$P = \frac{R_u T}{\bar{v}^2}\left(1 - \frac{c}{\bar{v}T^3}\right)(\bar{v} + B) - \frac{A}{\bar{v}^2}$$

and solving using an equation solver such as EES gives

$$T_2 = 532.2\ \text{K} = \mathbf{958\ R}$$

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3-140E The specific volume of nitrogen at a given state is to be determined using the ideal gas relation, the Benedict-Webb-Rubin equation, and the compressibility factor.

Properties The properties of nitrogen are (Table A-1E)

$R = 0.3830$ psia·ft^3/lbm·R, $M = 28.013$ lbm/lbmol, $T_{cr} = 227.1$ R, $P_{cr} = 492$ psia

Analysis (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.3830\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(360\,\text{R})}{400\,\text{psia}} = \mathbf{0.3447\ ft^3/lbm}$$

Nitrogen
400 psia, -100°F

(*b*) Using the coefficients of Table 3-4 for nitrogen and the given data in SI units, the Benedict-Webb-Rubin equation of state is

$$P = \frac{R_u T}{\bar{v}} + \left(B_0 R_u T - A_0 - \frac{C_0}{T^2}\right)\frac{1}{\bar{v}} + \frac{bR_u T - a}{\bar{v}^3} + \frac{a\alpha}{\bar{v}^6} + \frac{c}{\bar{v}^3 T^2}\left(1 + \frac{\gamma}{\bar{v}^2}\right)\exp(-\gamma/\bar{v}^2)$$

$$2758 = \frac{(8.314)(200)}{\bar{v}_2} + \left(0.04074\times 8.314\times 200 - 106.73 - \frac{8.164\times 10^5}{200^2}\right)\frac{1}{\bar{v}^2} + \frac{0.002328\times 8.314\times 200 - 2.54}{\bar{v}^3}$$
$$+ \frac{2.54\times 1.272\times 10^{-4}}{\bar{v}^6} + \frac{7.379\times 10^4}{\bar{v}^3(200)^2}\left(1 + \frac{0.0053}{\bar{v}^2}\right)\exp(-0.0053/\bar{v}^2)$$

The solution of this equation by an equation solver such as EES gives

$$\bar{v} = 0.5666\ \text{m}^3/\text{kmol}$$

Then,

$$v = \frac{\bar{v}}{M} = \frac{0.5666\,\text{m}^3/\text{kmol}}{28.013\,\text{kg/kmol}}\left(\frac{16.02\,\text{ft}^3/\text{lbm}}{1\,\text{m}^3/\text{kg}}\right) = \mathbf{0.3240\ ft^3/lbm}$$

(*c*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} T_R &= \frac{T}{T_{cr}} = \frac{360\,\text{R}}{227.1\,\text{R}} = 1.585 \\ P_R &= \frac{P}{P_{cr}} = \frac{400\,\text{psia}}{492\,\text{psia}} = 0.813 \end{aligned}\right\} \ Z = 0.94$$

Thus,

$$v = Zv_{\text{ideal}} = (0.94)(0.3447\,\text{ft}^3/\text{lbm}) = \mathbf{0.3240\ ft^3/lbm}$$

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Fundamentals of Engineering (FE) Exam Problems

3-141 A rigid tank contains 2 kg of an ideal gas at 4 atm and 40°C. Now a valve is opened, and half of mass of the gas is allowed to escape. If the final pressure in the tank is 2.2 atm, the final temperature in the tank is

(a) 71°C (b) 44°C (c) -100°C (d) 20°C (e) 172°C

Answer (a) 71°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"When R=constant and V= constant, P1/P2=m1*T1/m2*T2"
m1=2 "kg"
P1=4 "atm"
P2=2.2 "atm"
T1=40+273 "K"
m2=0.5*m1 "kg"
P1/P2=m1*T1/(m2*T2)
T2_C=T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
P1/P2=m1*(T1-273)/(m2*W1_T2) "Using C instead of K"
P1/P2=m1*T1/(m1*(W2_T2+273)) "Disregarding the decrease in mass"
P1/P2=m1*T1/(m1*W3_T2) "Disregarding the decrease in mass, and not converting to deg. C"
W4_T2=(T1-273)/2 "Taking T2 to be half of T1 since half of the mass is discharged"
```

3-142 The pressure of an automobile tire is measured to be 190 kPa (gage) before a trip and 215 kPa (gage) after the trip at a location where the atmospheric pressure is 95 kPa. If the temperature of air in the tire before the trip is 25°C, the air temperature after the trip is

(a) 51.1°C (b) 64.2°C (c) 27.2°C (d) 28.3°C (e) 25.0°C

Answer (a) 51.1°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"When R, V, and m are constant, P1/P2=T1/T2"
Patm=95
P1=190+Patm "kPa"
P2=215+Patm "kPa"
T1=25+273 "K"
P1/P2=T1/T2
T2_C=T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
P1/P2=(T1-273)/W1_T2 "Using C instead of K"
(P1-Patm)/(P2-Patm)=T1/(W2_T2+273) "Using gage pressure instead of absolute pressure"
(P1-Patm)/(P2-Patm)=(T1-273)/W3_T2 "Making both of the mistakes above"
W4_T2=T1-273 "Assuming the temperature to remain constant"
```

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3-143 A 300-m^3 rigid tank is filled with saturated liquid-vapor mixture of water at 200 kPa. If 25% of the mass is liquid and the 75% of the mass is vapor, the total mass in the tank is

(a) 451 kg (b) 556 kg (c) 300 kg (d) 331 kg (e) 195 kg

Answer (a) 451 kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_tank=300 "m3"
P1=200 "kPa"
x=0.75
v_f=VOLUME(Steam_IAPWS, x=0,P=P1)
v_g=VOLUME(Steam_IAPWS, x=1,P=P1)
v=v_f+x*(v_g-v_f)
m=V_tank/v "kg"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
T=TEMPERATURE(Steam_IAPWS,x=0,P=P1)
P1*V_tank=W1_m*R*(T+273) "Treating steam as ideal gas"
P1*V_tank=W2_m*R*T "Treating steam as ideal gas and using deg.C"
W3_m=V_tank "Taking the density to be 1 kg/m^3"
```

3-144 Water is boiled at 1 atm pressure in a coffee maker equipped with an immersion-type electric heating element. The coffee maker initially contains 1 kg of water. Once boiling started, it is observed that half of the water in the coffee maker evaporated in 10 minutes. If the heat loss from the coffee maker is negligible, the power rating of the heating element is

(a) 3.8 kW (b) 2.2 kW (c) 1.9 kW (d) 1.6 kW (e) 0.8 kW

Answer (c) 1.9 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_1=1 "kg"
P=101.325 "kPa"
time=10*60 "s"
m_evap=0.5*m_1
Power*time=m_evap*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,P=P)
h_g=ENTHALPY(Steam_IAPWS, x=1,P=P)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
W1_Power*time=m_evap*h_g "Using h_g"
W2_Power*time/60=m_evap*h_g "Using minutes instead of seconds for time"
W3_Power=2*Power  "Assuming all the water evaporates"
```

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3-145 A 1-m^3 rigid tank contains 10 kg of water (in any phase or phases) at 160°C. The pressure in the tank is

(a) 738 kPa (b) 618 kPa (c) 370 kPa (d) 2000 kPa (e) 1618 kPa

Answer (b) 618 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_tank=1 "m^3"
m=10 "kg"
v=V_tank/m
T=160 "C"
P=PRESSURE(Steam_IAPWS,v=v,T=T)

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
W1_P*V_tank=m*R*(T+273) "Treating steam as ideal gas"
W2_P*V_tank=m*R*T "Treating steam as ideal gas and using deg.C"
```

3-146 Water is boiling at 1 atm pressure in a stainless steel pan on an electric range. It is observed that 2 kg of liquid water evaporates in 30 minutes. The rate of heat transfer to the water is

(a) 2.51 kW (b) 2.32 kW (c) 2.97 kW (d) 0.47 kW (e) 3.12 kW

Answer (a) 2.51 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_evap=2 "kg"
P=101.325 "kPa"
time=30*60 "s"
Q*time=m_evap*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,P=P)
h_g=ENTHALPY(Steam_IAPWS, x=1,P=P)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
W1_Q*time=m_evap*h_g "Using h_g"
W2_Q*time/60=m_evap*h_g "Using minutes instead of seconds for time"
W3_Q*time=m_evap*h_f "Using h_f"
```

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3-147 Water is boiled in a pan on a stove at sea level. During 10 min of boiling, its is observed that 200 g of water has evaporated. Then the rate of heat transfer to the water is

(a) 0.84 kJ/min (b) 45.1 kJ/min (c) 41.8 kJ/min (d) 53.5 kJ/min (e) 225.7 kJ/min

Answer (b) 45.1 kJ/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_evap=0.2 "kg"
P=101.325 "kPa"
time=10 "min"
Q*time=m_evap*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,P=P)
h_g=ENTHALPY(Steam_IAPWS, x=1,P=P)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
W1_Q*time=m_evap*h_g "Using h_g"
W2_Q*time*60=m_evap*h_g "Using seconds instead of minutes for time"
W3_Q*time=m_evap*h_f "Using h_f"
```

3-148 A rigid 3-m^3 rigid vessel contains steam at 4 MPa and 500°C. The mass of the steam is

(a) 3 kg (b) 9 kg (c) 26 kg (d) 35 kg (e) 52 kg

Answer (d) 35 kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=3 "m^3"
m=V/v1 "m^3/kg"
P1=4000 "kPa"
T1=500 "C"
 v1=VOLUME(Steam_IAPWS,T=T1,P=P1)

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
P1*V=W1_m*R*(T1+273) "Treating steam as ideal gas"
P1*V=W2_m*R*T1 "Treating steam as ideal gas and using deg.C"
```

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3-149 Consider a sealed can that is filled with refrigerant-134a. The contents of the can are at the room temperature of 25°C. Now a leak developes, and the pressure in the can drops to the local atmospheric pressure of 90 kPa. The temperature of the refrigerant in the can is expected to drop to (rounded to the nearest integer)

(a) 0°C (b) -29°C (c) -16°C (d) 5°C (e) 25°C

Answer (b) -29°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P2=90 "kPa"
T2=TEMPERATURE(R134a,x=0,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
```

3-150 … 3-152 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 4
ENERGY ANALYSIS OF CLOSED SYSTEMS

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Moving Boundary Work

4-1C Yes.

4-2C The area under the process curve, and thus the boundary work done, is greater in the constant pressure case.

4-3 $1\ \text{kPa}\cdot\text{m}^3 = 1\ \text{k(N/m}^2)\cdot\text{m}^3 = 1\ \text{kN}\cdot\text{m} = 1\ \text{kJ}$

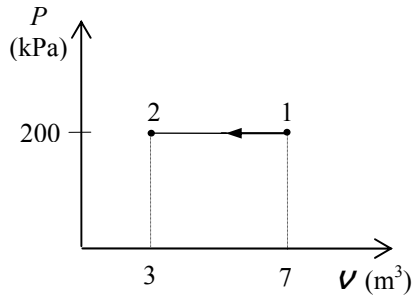
4-4 Helium is compressed in a piston-cylinder device. The initial and final temperatures of helium and the work required to compress it are to be determined.

Assumptions The process is quasi-equilibrium.

Properties The gas constant of helium is R = 2.0769 kJ/kg·K (Table A-1).

Analysis The initial specific volume is

$$v_1 = \frac{V_1}{m} = \frac{7\ \text{m}^3}{1\ \text{kg}} = 7\ \text{m}^3/\text{kg}$$



Using the ideal gas equation,

$$T_1 = \frac{P_1 v_1}{R} = \frac{(150\ \text{kPa})(7\ \text{m}^3/\text{kg})}{2.0769\ \text{kJ/kg}\cdot\text{K}} = \mathbf{505.1\ K}$$

Since the pressure stays constant,

$$T_2 = \frac{V_2}{V_1} T_1 = \frac{3\ \text{m}^3}{7\ \text{m}^3}(505.1\ \text{K}) = \mathbf{216.5\ K}$$

and the work integral expression gives

$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = (150\ \text{kPa})(3-7)\ \text{m}^3\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = -600\ \text{kJ}$$

That is,

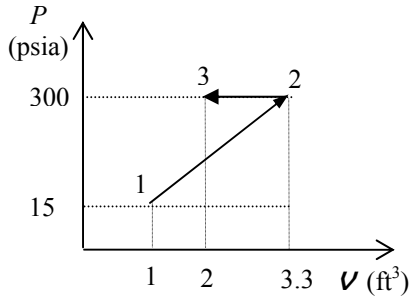
$$W_{b,\text{in}} = \mathbf{600\ kJ}$$

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4-5E The boundary work done during the process shown in the figure is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The work done is equal to the the sumof the areas under the process lines 1-2 and 2-3:



$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(v_2 - v_1) + P_2(v_3 - v_2)$$

$$= \frac{(300+15)\text{psia}}{2}(3.3-1)\,\text{ft}^3)\left(\frac{1\,\text{Btu}}{5.404\,\text{psia}\cdot\text{ft}^3}\right)$$

$$+ (300\,\text{psia})(2-3.3)\text{ft}^3\left(\frac{1\,\text{Btu}}{5.404\,\text{psia}\cdot\text{ft}^3}\right)$$

$$= \mathbf{-5.14\,Btu}$$

The negative sign shows that the work is done on the system.

4-6 The work done during the isothermal process shown in the figure is to be determined.

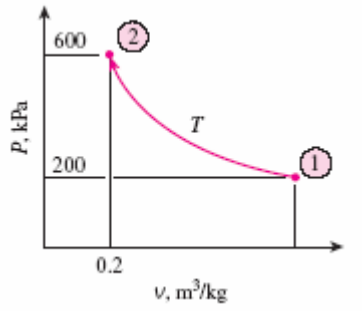
Assumptions The process is quasi-equilibrium.

Analysis From the ideal gas equation,

$$P = \frac{RT}{v}$$

For an isothermal process,

$$v_1 = v_2\,\frac{P_2}{P_1} = (0.2\ \text{m}^3/\text{kg})\frac{600\,\text{kPa}}{200\,\text{kPa}} = 0.6\ \text{m}^3/\text{kg}$$



Substituting ideal gas equation and this result into the boundary work integral produces

$$W_{b,\text{out}} = \int_1^2 P\,dv = mRT\int_1^2 \frac{dv}{v}$$

$$= mP_1v_1\ln\frac{v_2}{v_1} = (3\ \text{kg})(200\ \text{kPa})(0.6\,\text{m}^3)\ln\frac{0.2\,\text{m}^3}{0.6\,\text{m}^3}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$

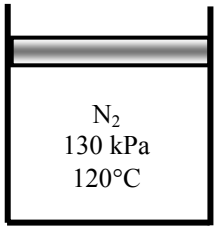
$$= \mathbf{-395.5\,kJ}$$

The negative sign shows that the work is done on the system.

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4-7 A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the polytropic expansion of nitrogen.

Properties The gas constant for nitrogen is 0.2968 kJ/kg.K (Table A-2).



Analysis The mass and volume of nitrogen at the initial state are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(130\ \text{kPa})(0.07\ \text{m}^3)}{(0.2968\ \text{kJ/kg.K})(120+273\ \text{K})} = 0.07802\ \text{kg}$$

$$V_2 = \frac{mRT_2}{P_2} = \frac{(0.07802\ \text{kg})(0.2968\ \text{kPa.m}^3/\text{kg.K})(100+273\ \text{K})}{100\ \text{kPa}} = 0.08637\ \text{m}^3$$

The polytropic index is determined from

$$P_1 V_1^n = P_2 V_2^n \longrightarrow (130\,\text{kPa})(0.07\,\text{m}^3)^n = (100\,\text{kPa})(0.08637\,\text{m}^3)^n \longrightarrow n = 1.249$$

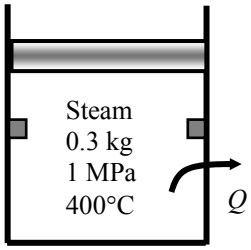
The boundary work is determined from

$$W_b = \frac{P_2 V_2 - P_1 V_1}{1-n} = \frac{(100\,\text{kPa})(0.08637\,\text{m}^3) - (130\,\text{kPa})(0.07\,\text{m}^3)}{1-1.249} = \mathbf{1.86\ kJ}$$

4-8 A piston-cylinder device with a set of stops contains steam at a specified state. Now, the steam is cooled. The compression work for two cases and the final temperature are to be determined.

Analysis (a) The specific volumes for the initial and final states are (Table A-6)

$$\left.\begin{matrix} P_1 = 1\,\text{MPa} \\ T_1 = 400°\text{C} \end{matrix}\right\} v_1 = 0.30661\ \text{m}^3/\text{kg} \qquad \left.\begin{matrix} P_2 = 1\,\text{MPa} \\ T_2 = 250°\text{C} \end{matrix}\right\} v_2 = 0.23275\ \text{m}^3/\text{kg}$$



Noting that pressure is constant during the process, the boundary work is determined from

$$W_b = mP(v_1 - v_2) = (0.3\,\text{kg})(1000\,\text{kPa})(0.30661 - 0.23275)\text{m}^3/\text{kg} = \mathbf{22.16\ kJ}$$

(b) The volume of the cylinder at the final state is 60% of initial volume. Then, the boundary work becomes

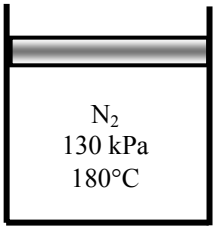
$$W_b = mP(v_1 - 0.60v_1) = (0.3\,\text{kg})(1000\,\text{kPa})(0.30661 - 0.60\times 0.30661)\text{m}^3/\text{kg} = \mathbf{36.79\ kJ}$$

The temperature at the final state is

$$\left.\begin{matrix} P_2 = 0.5\,\text{MPa} \\ v_2 = (0.60\times 0.30661)\,\text{m}^3/\text{kg} \end{matrix}\right\} T_2 = \mathbf{151.8°C} \quad \text{(Table A-5)}$$

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4-9 A piston-cylinder device contains nitrogen gas at a specified state. The final temperature and the boundary work are to be determined for the isentropic expansion of nitrogen.



Properties The properties of nitrogen are R = 0.2968 kJ/kg.K , k = 1.395 (Tables A-2a, A-2b)

Analysis The mass and the final volume of nitrogen are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(130\,\text{kPa})(0.07\,\text{m}^3)}{(0.2968\,\text{kJ/kg.K})(180+273\,\text{K})} = 0.06768\,\text{kg}$$

$$P_1 V_1^k = P_2 V_2^k \longrightarrow (130\,\text{kPa})(0.07\,\text{m}^3)^{1.395} = (80\,\text{kPa}) V_2^{1.395} \longrightarrow V_2 = 0.09914\,\text{m}^3$$

The final temperature and the boundary work are determined as

$$T_2 = \frac{P_2 V_2}{mR} = \frac{(80\,\text{kPa})(0.09914\,\text{m}^3)}{(0.06768\,\text{kg})(0.2968\,\text{kPa.m}^3/\text{kg.K})} = \mathbf{395\ K}$$

$$W_b = \frac{P_2 V_2 - P_1 V_1}{1-k} = \frac{(80\,\text{kPa})(0.09914\,\text{m}^3) - (130\,\text{kPa})(0.07\,\text{m}^3)}{1-1.395} = \mathbf{2.96\ kJ}$$

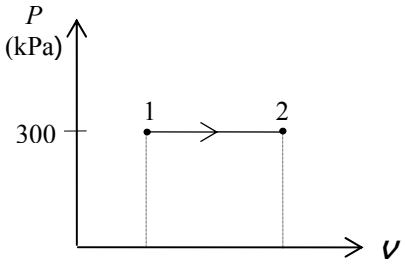
4-10 Saturated water vapor in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Properties Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 300\,\text{kPa} \\ \text{Sat. vapor} \end{array}\right\} v_1 = v_{g@300\,\text{kPa}} = 0.60582\,\text{m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 300\,\text{kPa} \\ T_2 = 200°\text{C} \end{array}\right\} v_2 = 0.71643\,\text{m}^3/\text{kg}$$



Analysis The boundary work is determined from its definition to be

$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = mP(v_2 - v_1)$$

$$= (5\,\text{kg})(300\,\text{kPa})(0.71643 - 0.60582)\,\text{m}^3/\text{kg}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{165.9\ kJ}$$

Discussion The positive sign indicates that work is done by the system (work output).

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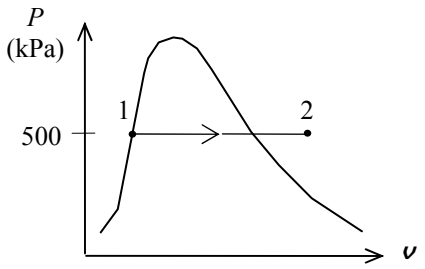
4-11 Refrigerant-134a in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Properties Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 500 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} v_1 = v_{f@500\text{ kPa}} = 0.0008059 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ T_2 = 70°\text{C} \end{array}\right\} v_2 = 0.052427 \text{ m}^3/\text{kg}$$



Analysis The boundary work is determined from its definition to be

$$m = \frac{V_1}{v_1} = \frac{0.05 \text{ m}^3}{0.0008059 \text{ m}^3/\text{kg}} = 62.04 \text{ kg}$$

and

$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = mP(v_2 - v_1)$$

$$= (62.04 \text{ kg})(500 \text{ kPa})(0.052427 - 0.0008059)\text{m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= \mathbf{1600 \text{ kJ}}$$

Discussion The positive sign indicates that work is done by the system (work output).

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4-12 Problem 4-11 is reconsidered. The effect of pressure on the work done as the pressure varies from 400 kPa to 1200 kPa is to be investigated. The work done is to be plotted versus the pressure.

Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns"
Vol_1L=200 [L]
x_1=0 "saturated liquid state"
P=900 [kPa]
T_2=70 [C]

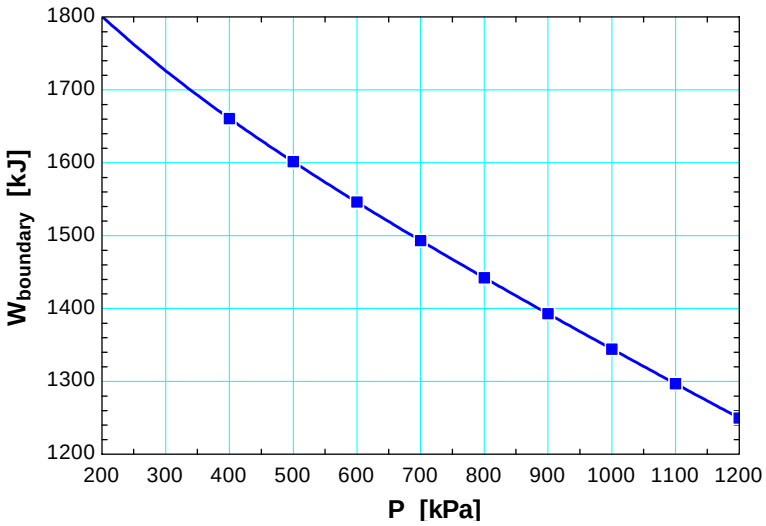
"Solution"
Vol_1=Vol_1L*convert(L,m^3)
"The work is the boundary work done by the R-134a during the constant pressure process."

W_boundary=P*(Vol_2-Vol_1)

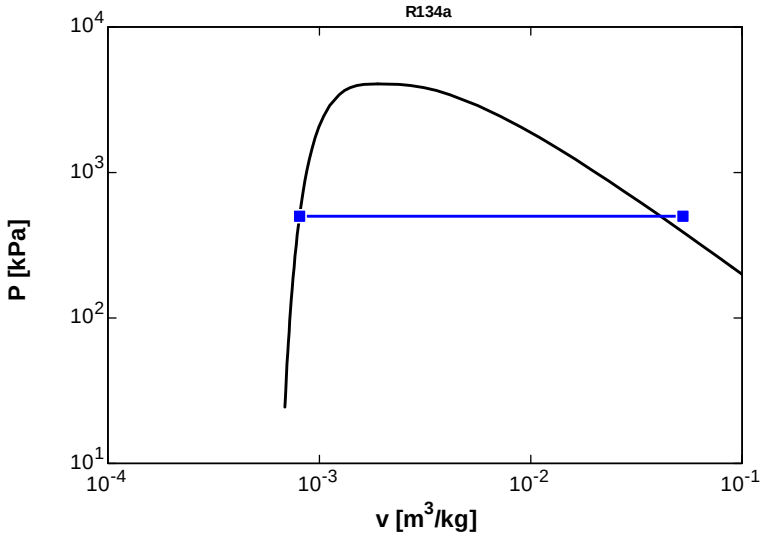
"The mass is:"
Vol_1=m*v_1
v_1=volume(R134a,P=P,x=x_1)
Vol_2=m*v_2
v_2=volume(R134a,P=P,T=T_2)

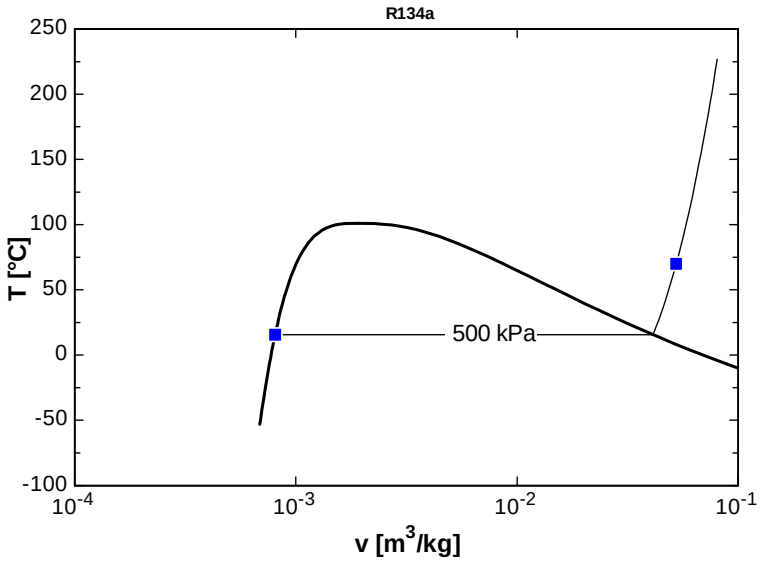
"Plot information:"
v[1]=v_1
v[2]=v_2
P[1]=P
P[2]=P
T[1]=temperature(R134a,P=P,x=x_1)
T[2]=T_2
```

P [kPa]	$W_{boundary}$ [kJ]
200	1801
400	1661
500	1601
600	1546
700	1493
800	1442
900	1393
1000	1344
1100	1297
1200	1250



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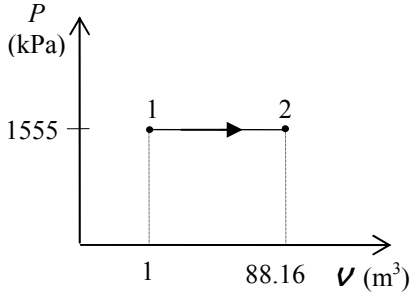
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4-13 Water is expanded isothermally in a closed system. The work produced is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis From water table

$$P_1 = P_2 = P_{\text{sat @ 200°C}} = 1554.9\ \text{kPa}$$
$$v_1 = v_{\text{f @ 200°C}} = 0.001157\ \text{m}^3/\text{kg}$$
$$v_2 = v_f + x v_{fg}$$
$$= 0.001157 + 0.80(0.12721 - 0.001157)$$
$$= 0.10200\ \text{m}^3/\text{kg}$$



The definition of specific volume gives

$$V_2 = V_1 \frac{v_2}{v_1} = (1\ \text{m}^3)\frac{0.10200\ \text{m}^3/\text{kg}}{0.001157\ \text{m}^3/\text{kg}} = 88.16\ \text{m}^3$$

The work done during the process is determined from

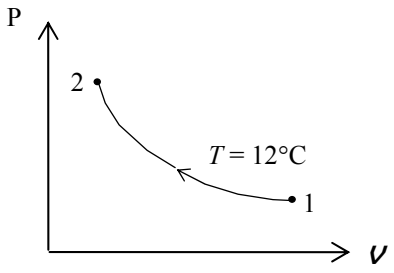
$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = (1554.9\ \text{kPa})(88.16 - 1)\text{m}^3\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = \mathbf{1.355 \times 10^5\ kJ}$$

4-14 Air in a cylinder is compressed at constant temperature until its pressure rises to a specified value. The boundary work done during this process is to be determined.

Assumptions **1** The process is quasi-equilibrium. **2** Air is an ideal gas.

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1).

Analysis The boundary work is determined from its definition to be



$$W_{b,\text{out}} = \int_1^2 P\,dV = P_1 V_1 \ln\frac{V_2}{V_1} = mRT\ln\frac{P_1}{P_2}$$
$$= (2.4\ \text{kg})(0.287\ \text{kJ/kg}\cdot\text{K})(285\ \text{K})\ln\frac{150\ \text{kPa}}{600\ \text{kPa}}$$
$$= \mathbf{-272\ kJ}$$

Discussion The negative sign indicates that work is done on the system (work input).

4-15 Several sets of pressure and volume data are taken as a gas expands. The boundary work done during this process is to be determined using the experimental data.

Assumptions The process is quasi-equilibrium.

Analysis Plotting the given data on a P-V diagram on a graph paper and evaluating the area under the process curve, the work done is determined to be **0.25 kJ**.

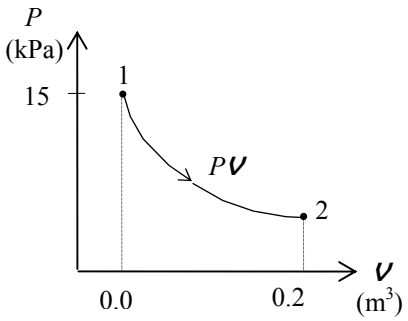
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4-16 A gas in a cylinder expands polytropically to a specified volume. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The boundary work for this polytropic process can be determined directly from

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^n = (350\ \text{kPa})\left(\frac{0.03\ \text{m}^3}{0.2\ \text{m}^3}\right)^{1.5} = 20.33\ \text{kPa}$$



and

$$W_{b,\text{out}} = \int_1^2 P\,dV = \frac{P_2V_2 - P_1V_1}{1-n}$$

$$= \frac{(20.33\times 0.2 - 350\times 0.03)\ \text{kPa}\cdot\text{m}^3}{1-1.5}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{12.9\ kJ}$$

Discussion The positive sign indicates that work is done by the system (work output).

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4-17 Problem 4-16 is reconsidered. The process described in the problem is to be plotted on a *P*-*V* diagram, and the effect of the polytropic exponent n on the boundary work as the polytropic exponent varies from 1.1 to 1.6 is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

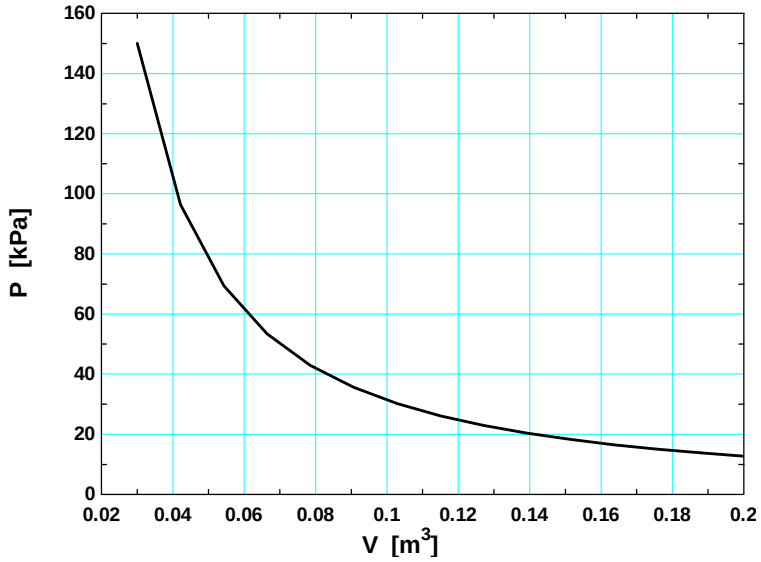
```
Function BoundWork(P[1],V[1],P[2],V[2],n)
"This function returns the Boundary Work for the polytropic process.  This function is required
since the expression for boundary work depens on whether n=1 or n<>1"
  If n<>1 then
    BoundWork:=(P[2]*V[2]-P[1]*V[1])/(1-n)"Use Equation 3-22 when n=1"
  else
   BoundWork:= P[1]*V[1]*ln(V[2]/V[1]) "Use Equation 3-20 when n=1"
  endif
end

"Inputs from the diagram window"
{n=1.5
P[1] = 350 [kPa]
V[1] = 0.03 [m^3]
V[2] = 0.2 [m^3]
Gas$='AIR'}
"System:  The gas enclosed in the piston-cylinder device."
"Process: Polytropic expansion or compression, P*V^n = C"
P[2]*V[2]^n=P[1]*V[1]^n
"n = 1.3" "Polytropic exponent"

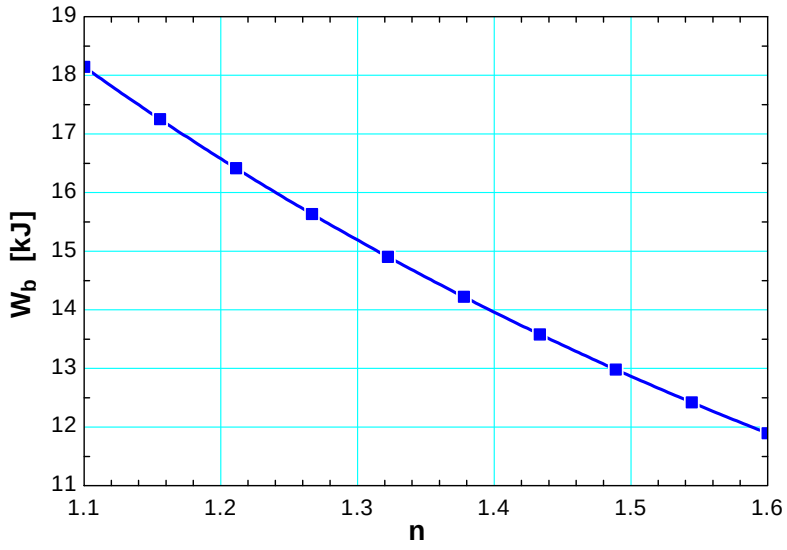
"Input Data"
W_b = BoundWork(P[1],V[1],P[2],V[2],n)"[kJ]"
"If we modify this problem and specify the mass, then we can calculate the final temperature of
the fluid for compression or expansion"
m[1] = m[2] "Conservation of mass for the closed system"
"Let's solve the problem for m[1] = 0.05 kg"
m[1] = 0.05 [kg]

"Find the temperatures from the pressure and specific volume."
T[1]=temperature(gas$,P=P[1],v=V[1]/m[1])
T[2]=temperature(gas$,P=P[2],v=V[2]/m[2])
```

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n	W_b [kJ]
1.1	18.14
1.156	17.25
1.211	16.41
1.267	15.63
1.322	14.9
1.378	14.22
1.433	13.58
1.489	12.98
1.544	12.42
1.6	11.89



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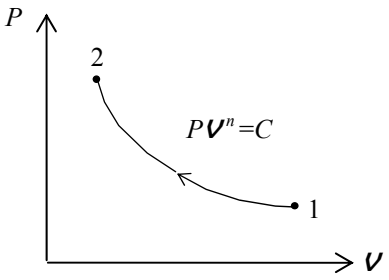
4-18 Nitrogen gas in a cylinder is compressed polytropically until the temperature rises to a specified value. The boundary work done during this process is to be determined.

Assumptions **1** The process is quasi-equilibrium. **2** Nitrogen is an ideal gas.

Properties The gas constant for nitrogen is R = 0.2968 kJ/kg.K (Table A-2a)

Analysis The boundary work for this polytropic process can be determined from

$$W_{b,\text{out}} = \int_1^2 P\,dV = \frac{P_2V_2 - P_1V_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n}$$
$$= \frac{(2\text{ kg})(0.2968\text{ kJ/kg}\cdot\text{K})(360-300)\text{K}}{1-1.4}$$
$$= \mathbf{-89.0\text{ kJ}}$$



Discussion The negative sign indicates that work is done on the system (work input).

4-19 A gas whose equation of state is $\bar{v}(P + 10/\bar{v}^2) = R_uT$ expands in a cylinder isothermally to a specified volume. The unit of the quantity 10 and the boundary work done during this process are to be determined.

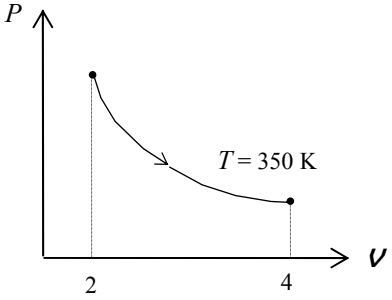
Assumptions The process is quasi-equilibrium.

Analysis (*a*) The term $10/\bar{v}^2$ must have pressure units since it is added to P.

Thus the quantity 10 must have the unit kPa·m^6/kmol^2.

(*b*) The boundary work for this process can be determined from

$$P = \frac{R_uT}{\bar{v}} - \frac{10}{\bar{v}^2} = \frac{R_uT}{V/N} - \frac{10}{(V/N)^2} = \frac{NR_uT}{V} - \frac{10N^2}{V^2}$$



and

$$W_{b,\text{out}} = \int_1^2 P\,dV = \int_1^2 \left(\frac{NR_uT}{V} - \frac{10N^2}{V^2}\right)dV = NR_uT\ln\frac{V_2}{V_1} + 10N^2\left(\frac{1}{V_2} - \frac{1}{V_1}\right)$$
$$= (0.2\text{ kmol})(8.314\text{ kJ/kmol}\cdot\text{K})(350\text{ K})\ln\frac{4\text{ m}^3}{2\text{ m}^3}$$
$$+ (10\text{ kPa}\cdot\text{m}^6/\text{kmol}^2)(0.5\text{kmol})^2\left(\frac{1}{4\text{ m}^3} - \frac{1}{2\text{ m}^3}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{403\text{ kJ}}$$

Discussion The positive sign indicates that work is done by the system (work output).

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4-20 Problem 4-19 is reconsidered. Using the integration feature, the work done is to be calculated and compared, and the process is to be plotted on a *P*-***V*** diagram.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
N=0.2 [kmol]
v1_bar=2/N "[m^3/kmol]"
v2_bar=4/N "[m^3/kmol]"
T=350 [K]
R_u=8.314 [kJ/kmol-K]

"The quation of state is:"
v_bar*(P+10/v_bar^2)=R_u*T "P is in kPa"

"using the EES integral function,  the boundary work, W_bEES, is"
W_b_EES=N*integral(P,v_bar, v1_bar, v2_bar,0.01)

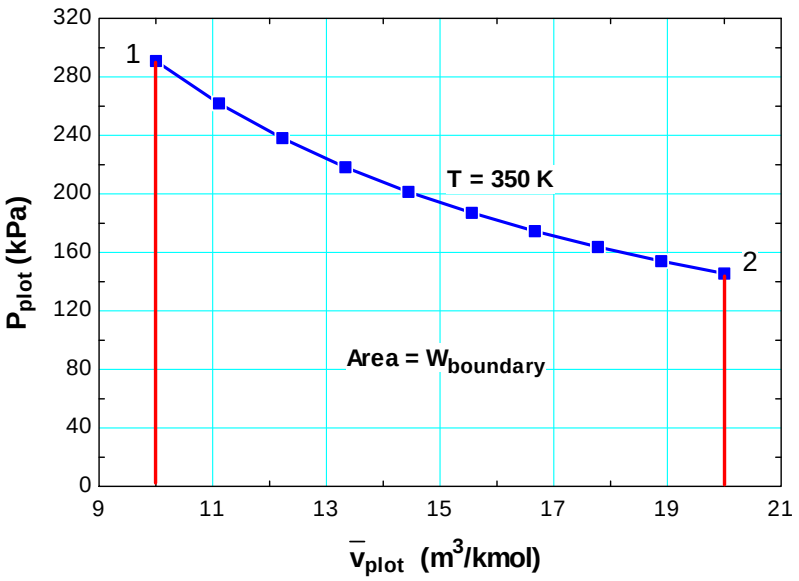
"We can show that W_bhand= integeral of Pdv_bar is
(one should solve for P=F(v_bar) and do the integral 'by hand' for practice)."
W_b_hand = N*(R_u*T*ln(v2_bar/v1_bar) +10*(1/v2_bar-1/v1_bar))

"To plot P vs v_bar, define P_plot =f(v_bar_plot, T) as"

{v_bar_plot*(P_plot+10/v_bar_plot^2)=R_u*T}

" P=P_plot and v_bar=v_bar_plot just to generate the parametric table for plotting purposes.  To plot P vs v_bar
for a new temperature or v_bar_plot range,  remove the '{' and '}' from the above equation, and reset the
v_bar_plot values in the Parametric Table.  Then press F3 or select Solve Table from the Calculate menu.  Next
select New Plot Window under the Plot menu to plot the new data."
```

P_{plot}	v_{plot}
290.9	10
261.8	11.11
238	12.22
218.2	13.33
201.4	14.44
187	15.56
174.6	16.67
163.7	17.78
154	18.89
145.5	20



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4-21 CO_2 gas in a cylinder is compressed until the volume drops to a specified value. The pressure changes during the process with volume as $P = aV^{-2}$. The boundary work done during this process is to be determined.

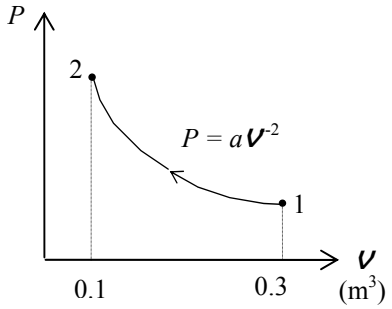
Assumptions The process is quasi-equilibrium.

Analysis The boundary work done during this process is determined from

$$W_{b,\text{out}} = \int_1^2 P\,dV = \int_1^2 \left(\frac{a}{V^2}\right) dV = -a\left(\frac{1}{V_2} - \frac{1}{V_1}\right)$$

$$= -(8\ \text{kPa}\cdot\text{m}^6)\left(\frac{1}{0.1\ \text{m}^3} - \frac{1}{0.3\ \text{m}^3}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{-53.3\ kJ}$$



Discussion The negative sign indicates that work is done on the system (work input).

4-22E A gas in a cylinder is heated and is allowed to expand to a specified pressure in a process during which the pressure changes linearly with volume. The boundary work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis (*a*) The pressure of the gas changes linearly with volume, and thus the process curve on a P-V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

At state 1:

$$P_1 = aV_1 + b$$

$$15\ \text{psia} = (5\ \text{psia/ft}^3)(7\ \text{ft}^3) + b$$

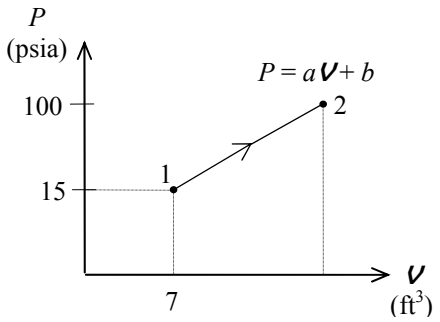
$$b = -20\ \text{psia}$$

At state 2:

$$P_2 = aV_2 + b$$

$$100\ \text{psia} = (5\ \text{psia/ft}^3)V_2 + (-20\ \text{psia})$$

$$V_2 = 24\ \text{ft}^3$$



and,

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100+15)\text{psia}}{2}(24-7)\text{ft}^3\left(\frac{1\ \text{Btu}}{5.4039\ \text{psia}\cdot\text{ft}^3}\right)$$

$$= \mathbf{181\ Btu}$$

Discussion The positive sign indicates that work is done by the system (work output).

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4-23 A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the isothermal expansion of nitrogen.

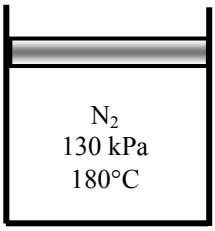
Properties The properties of nitrogen are R = 0.2968 kJ/kg.K , k = 1.4 (Table A-2a).

Analysis We first determine initial and final volumes from ideal gas relation, and find the boundary work using the relation for isothermal expansion of an ideal gas

$$V_1 = \frac{mRT}{P_1} = \frac{(0.25\,\text{kg})(0.2968\,\text{kJ/kg.K})(180+273\,\text{K})}{(130\,\text{kPa})} = 0.2586\,\text{m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.25\,\text{kg})(0.2968\,\text{kJ/kg.K})(180+273\,\text{K})}{80\,\text{kPa}} = 0.4202\,\text{m}^3$$

$$W_b = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (130\,\text{kPa})(0.2586\,\text{m}^3)\ln\left(\frac{0.4202\,\text{m}^3}{0.2586\,\text{m}^3}\right) = \mathbf{16.3\,kJ}$$



4-24 A piston-cylinder device contains air gas at a specified state. The air undergoes a cycle with three processes. The boundary work for each process and the net work of the cycle are to be determined.

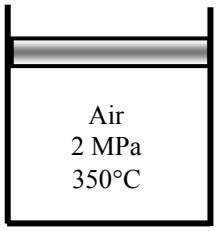
Properties The properties of air are R = 0.287 kJ/kg.K , k = 1.4 (Table A-2a).

Analysis For the isothermal expansion process:

$$V_1 = \frac{mRT}{P_1} = \frac{(0.15\,\text{kg})(0.287\,\text{kJ/kg.K})(350+273\,\text{K})}{(2000\,\text{kPa})} = 0.01341\,\text{m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.15\,\text{kg})(0.287\,\text{kJ/kg.K})(350+273\,\text{K})}{(500\,\text{kPa})} = 0.05364\,\text{m}^3$$

$$W_{b,1-2} = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (2000\,\text{kPa})(0.01341\,\text{m}^3)\ln\left(\frac{0.05364\,\text{m}^3}{0.01341\,\text{m}^3}\right) = \mathbf{37.18\,kJ}$$



For the polytropic compression process:

$$P_2 V_2^n = P_3 V_3^n \longrightarrow (500\,\text{kPa})(0.05364\,\text{m}^3)^{1.2} = (2000\,\text{kPa})V_3^{1.2} \longrightarrow V_3 = 0.01690\,\text{m}^3$$

$$W_{b,2-3} = \frac{P_3 V_3 - P_2 V_2}{1-n} = \frac{(2000\,\text{kPa})(0.01690\,\text{m}^3) - (500\,\text{kPa})(0.05364\,\text{m}^3)}{1-1.2} = \mathbf{-34.86\,kJ}$$

For the constant pressure compression process:

$$W_{b,3-1} = P_3(V_1 - V_3) = (2000\,\text{kPa})(0.01341 - 0.01690)\text{m}^3 = \mathbf{-6.97\,kJ}$$

The net work for the cycle is the sum of the works for each process

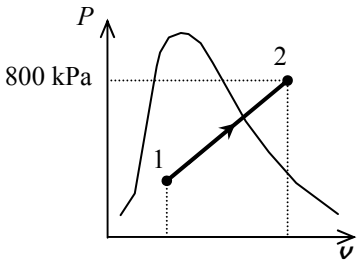
$$W_{\text{net}} = W_{b,1-2} + W_{b,2-3} + W_{b,3-1} = 37.18 + (-34.86) + (-6.97) = \mathbf{-4.65\,kJ}$$

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4-25 A saturated water mixture contained in a spring-loaded piston-cylinder device is heated until the pressure and temperature rises to specified values. The work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The initial state is saturated mixture at 90°C. The pressure and the specific volume at this state are (Table A-4),



$$P_1 = 70.183\ \text{kPa}$$
$$\nu_1 = \nu_f + x\nu_{fg}$$
$$= 0.001036 + (0.10)(2.3593 - 0.001036)$$
$$= 0.23686\ \text{m}^3/\text{kg}$$

The final specific volume at 800 kPa and 250°C is (Table A-6)

$$\nu_2 = 0.29321\ \text{m}^3/\text{kg}$$

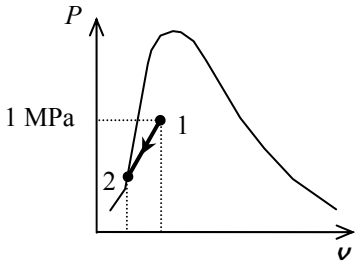
Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2} m(\nu_2 - \nu_1)$$
$$= \frac{(70.183 + 800)\text{kPa}}{2}(1\,\text{kg})(0.29321 - 0.23686)\text{m}^3\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{24.52\ kJ}$$

4-26 A saturated water mixture contained in a spring-loaded piston-cylinder device is cooled until it is saturated liquid at a specified temperature. The work done during this process is to be determined.

Assumptions The process is quasi-equilibrium.

Analysis The initial state is saturated mixture at 1 MPa. The specific volume at this state is (Table A-5),



$$\nu_1 = \nu_f + x\nu_{fg}$$
$$= 0.001127 + (0.30)(0.19436 - 0.001127)$$
$$= 0.059097\ \text{m}^3/\text{kg}$$

The final state is saturated liquid at 100°C (Table A-4)

$$P_2 = 101.42\ \text{kPa}$$
$$\nu_2 = \nu_f = 0.001043\ \text{m}^3/\text{kg}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2} m(\nu_2 - \nu_1)$$
$$= \frac{(1000 + 101.42)\text{kPa}}{2}(1.5\,\text{kg})(0.001043 - 0.059097)\text{m}^3\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{-48.0\ kJ}$$

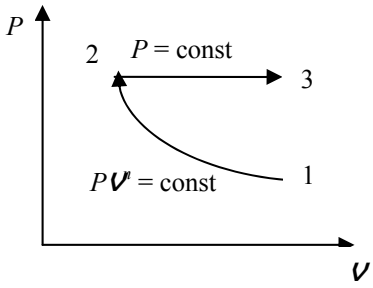
The negative sign shows that the work is done on the system in the amount of 48.0 kJ.

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4-27 An ideal gas undergoes two processes in a piston-cylinder device. The process is to be sketched on a P-V diagram; an expression for the ratio of the compression to expansion work is to be obtained and this ratio is to be calculated for given values of n and r.

Assumptions The process is quasi-equilibrium.

Analysis (a) The processes on a P-V diagram is as follows:



(b) The ratio of the compression-to-expansion work is called the back-work ratio BWR.

Process 1-2: $W_{b,1-2} = \int_1^2 P\,dV$

The process is PV^n = constant, , $P = \dfrac{\text{constant}}{V^n}$, and the integration results in

$$W_{b,1\text{-}2} = \frac{P_2V_2 - P_1V_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n}$$

where the ideal gas law has been used. However, the compression work is

$$W_{\text{comp}} = -W_{\text{b,1-2}} = \frac{mR(T_2 - T_1)}{n-1}$$

Process 2-3: $W_{b,2-3} = \int_2^3 P\,dV$

The process is P = constant and the integration gives

$$W_{b,2-3} = P(V_3 - V_2)$$

where $P = P_2 = P_3$. Using the ideal gas law, the expansion work is

$$W_{\text{exp}} = W_{\text{b,2-3}} = mR(T_3 - T_2)$$

The back-work ratio is defined as

$$BWR = \frac{W_{\text{comp}}}{W_{\text{exp}}} = \frac{\dfrac{mR(T_2 - T_1)}{n-1}}{mR(T_3 - T_2)} = \frac{1}{n-1}\frac{(T_2 - T_1)}{(T_3 - T_2)} = \frac{1}{n-1}\frac{T_2}{T_2}\frac{(1 - T_1/T_2)}{(T_3/T_2 - 1)} = \frac{1}{n-1}\frac{(1 - T_1/T_2)}{(T_3/T_2 - 1)}$$

Since process 1-2 is polytropic, the temperature-volume relation for the ideal gas is

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} = \left(\frac{1}{r}\right)^{n-1} = r^{1-n}$$

where r is the compression ratio V_1/V_2. Since process 2-3 is constant pressure, the combined ideal gas law gives

$$\frac{P_3V_3}{T_3} = \frac{P_2V_2}{T_2} \text{ and } P_3 = P_2, \text{ then } \frac{T_3}{T_2} = \frac{V_3}{V_2} = \frac{V_1}{V_2} = r$$

The back-work ratio becomes

$$BWR = \frac{1}{n-1}\frac{(1 - r^{1-n})}{(r-1)}$$

(c) For n = 1.4 and r = 6, the value of the BWR is

$$BWR = \frac{1}{1.4-1}\frac{(1 - 6^{1-1.4})}{(6-1)} = \mathbf{0.256}$$

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Closed System Energy Analysis

4-28E The table is to be completed using conservation of energy principle for a closed system.

Analysis The energy balance for a closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{out}} = E_2 - E_1 = m(e_2 - e_1)$$

Application of this equation gives the following completed table:

Q_{in} (Btu)	W_{out} (Btu)	E_1 (Btu)	E_2 (Btu)	m (lbm)	$e_2 - e_1$ (Btu/lbm)
350	**510**	1020	860	3	**-53.3**
350	130	550	**770**	5	**44.0**
560	260	600	**900**	2	150
-500	**0**	1400	900	7	**-71.4**
-650	-50	1000	**400**	3	-200

4-29E A piston-cylinder device involves expansion work and work input by a stirring device. The net change of internal energy is to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved **3** The thermal energy stored in the cylinder itself is negligible.

Analysis This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{in}} - W_{\text{out}} = \Delta U \quad \text{(since KE = PE = 0)}$$



where

$$W_{\text{in}} = 10.28 \text{ Btu}$$

and

$$W_{\text{out}} = 15{,}000 \text{ lbf} \cdot \text{ft} = (15{,}000 \text{ lbf} \cdot \text{ft}) \frac{1 \text{ Btu}}{778.17 \text{ lbf} \cdot \text{ft}} = 19.28 \text{ Btu}$$

Substituting,

$$\Delta U = W_{\text{in}} - W_{\text{out}} = 10.28 - 19.28 = \mathbf{-9.00\ Btu}$$

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4-30E The heat transfer during a process that a closed system undergoes without any internal energy change is to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** The compression or expansion process is quasi-equilibrium.

Analysis The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{out}} = \Delta U = 0 \quad \text{(since KE = PE = 0)}$$

$$Q_{\text{in}} = W_{\text{out}}$$

Then,

$$Q_{\text{in}} = 1.1\times 10^6 \text{ lbf}\cdot\text{ft}\left(\frac{1\,\text{Btu}}{778.17\,\text{lbf}\cdot\text{ft}}\right) = \mathbf{1414\ Btu}$$

4-31 Motor oil is contained in a rigid container that is equipped with a stirring device. The rate of specific energy increase is to be determined.

Analysis This is a closed system since no mass enters or leaves. The energy balance for closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{sh,in}} = \Delta \dot{E}$$

Then,

$$\Delta \dot{E} = \dot{Q}_{\text{in}} + \dot{W}_{\text{sh,in}} = 1 + 1.5 = 2.5 = 2.5\text{ W}$$

Dividing this by the mass in the system gives

$$\Delta \dot{e} = \frac{\Delta \dot{E}}{m} = \frac{2.5\text{ J/s}}{1.5\text{ kg}} = \mathbf{1.67\ J/kg\cdot s}$$

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4-32 An insulated rigid tank is initially filled with a saturated liquid-vapor mixture of water. An electric heater in the tank is turned on, and the entire liquid in the tank is vaporized. The length of time the heater was kept on is to be determined, and the process is to be shown on a P-v diagram.

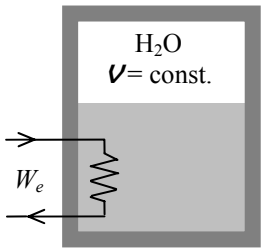
Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The device is well-insulated and thus heat transfer is negligible. **3** The energy stored in the resistance wires, and the heat transferred to the tank itself is negligible.

Analysis We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$\text{V}I\Delta t = m(u_2 - u_1)$$

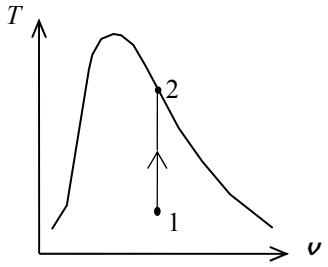


The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 150 \text{ kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} v_f = 0.001053, \quad v_g = 1.1594 \text{ m}^3/\text{kg} \\ u_f = 466.97, \quad u_{fg} = 2052.3 \text{ kJ/kg} \end{array}$$

$$v_1 = v_f + x_1 v_{fg} = 0.001053 + \left[0.25 \times (1.1594 - 0.001053)\right] = 0.29065 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 466.97 + (0.25 \times 2052.3) = 980.03 \text{ kJ/kg}$$



$$\left.\begin{array}{l} v_2 = v_1 = 0.29065 \text{ m}^3/\text{kg} \\ \text{sat.vapor} \end{array}\right\} u_2 = u_{g@0.29065 \text{ m}^3/\text{kg}} = 2569.7 \text{ kJ/kg}$$

Substituting,

$$(110 \text{ V})(8 \text{ A})\Delta t = (2 \text{ kg})(2569.7 - 980.03)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

$$\Delta t = 33613 \text{ s} = \mathbf{60.2 \text{ min}}$$

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4-33  Problem 4-32 is reconsidered. The effect of the initial mass of water on the length of time required to completely vaporize the liquid as the initial mass varies from 1 kg to 10 kg is to be investigated. The vaporization time is to be plotted against the initial mass.

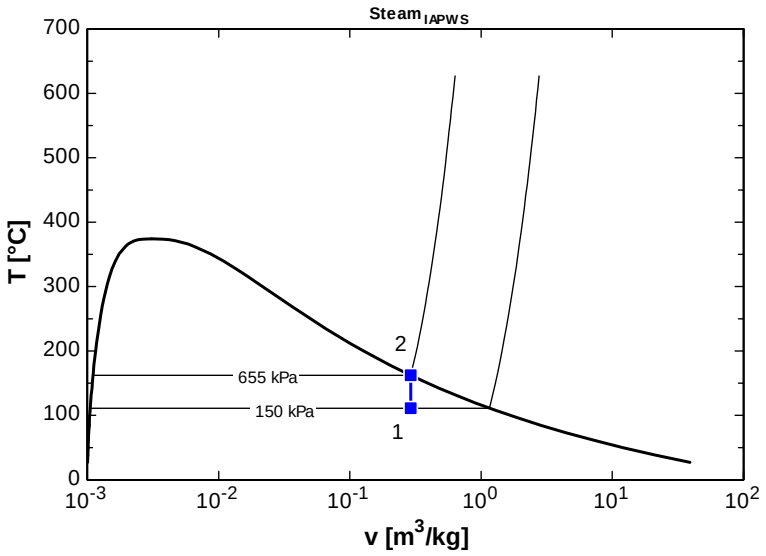
Analysis The problem is solved using EES, and the solution is given below.

```
PROCEDURE P2X2(v[1]:P[2],x[2])
Fluid$='Steam_IAPWS'
If v[1] > V_CRIT(Fluid$) then
P[2]=pressure(Fluid$,v=v[1],x=1)
x[2]=1
else
P[2]=pressure(Fluid$,v=v[1],x=0)
x[2]=0
EndIf
End

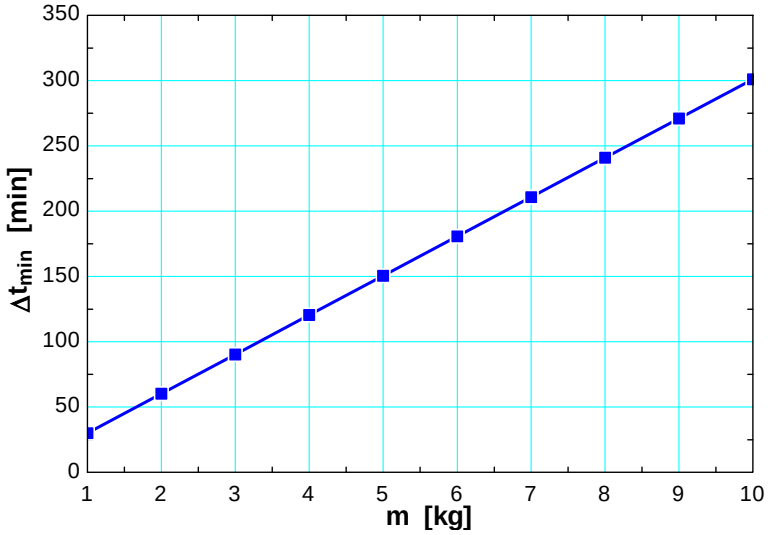
"Knowns"
{m=2 [kg]}
P[1]=150 [kPa]
y=0.75 "moisture"
Volts=110 [V]
I=8 [amp]

"Solution"
"Conservation of Energy for the closed tank:"
E_dot_in-E_dot_out=DELTAE_dot
E_dot_in=W_dot_ele  "[kW]"
W_dot_ele=Volts*I*CONVERT(J/s,kW) "[kW]"
E_dot_out=0 "[kW]"
DELTAE_dot=m*(u[2]-u[1])/DELTAt_s "[kW]"
DELTAt_min=DELTAt_s*convert(s,min) "[min]"
"The quality at state 1 is:"
Fluid$='Steam_IAPWS'
x[1]=1-y
u[1]=INTENERGY(Fluid$,P=P[1], x=x[1]) "[kJ/kg]"
v[1]=volume(Fluid$,P=P[1], x=x[1]) "[m^3/kg]"
T[1]=temperature(Fluid$,P=P[1], x=x[1]) "[C]"
"Check to see if state 2 is on the saturated liquid line or saturated vapor line:"
Call P2X2(v[1]:P[2],x[2])
u[2]=INTENERGY(Fluid$,P=P[2], x=x[2]) "[kJ/kg]"
v[2]=volume(Fluid$,P=P[2], x=x[2]) "[m^3/kg]"
T[2]=temperature(Fluid$,P=P[2], x=x[2]) "[C]"
```

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Δt_{min} [min]	m [kg]
30.11	1
60.21	2
90.32	3
120.4	4
150.5	5
180.6	6
210.7	7
240.9	8
271	9
301.1	10



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4-34 Saturated water vapor is isothermally condensed to a saturated liquid in a piston-cylinder device. The heat transfer and the work done are to be determined.

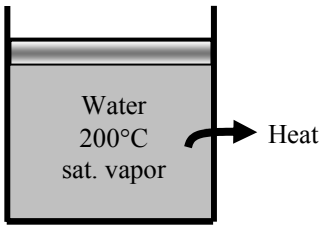
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$

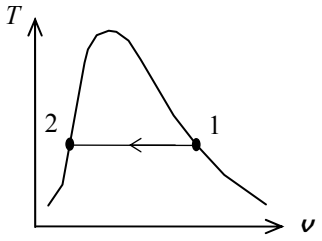


The properties at the initial and final states are (Table A-4)

$$\left.\begin{aligned} T_1 &= 200°\text{C} \\ x_1 &= 1 \end{aligned}\right\} \begin{aligned} v_1 &= v_g = 0.12721\ \text{m}^3/\text{kg} \\ u_1 &= u_g = 2594.2\ \text{kJ/kg} \end{aligned}$$

$$P_1 = P_2 = 1554.9\ \text{kPa}$$

$$\left.\begin{aligned} T_2 &= 200°\text{C} \\ x_2 &= 0 \end{aligned}\right\} \begin{aligned} v_2 &= v_f = 0.001157\ \text{m}^3/\text{kg} \\ u_2 &= u_f = 850.46\ \text{kJ/kg} \end{aligned}$$



The work done during this process is

$$w_{b,\text{out}} = \int_1^2 P\,dv = P(v_2 - v_1) = (1554.9\ \text{kPa})(0.001157 - 0.12721)\ \text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = -196.0\ \text{kJ/kg}$$

That is,

$$w_{b,\text{in}} = \mathbf{196.0\ kJ/kg}$$

Substituting the energy balance equation, we get

$$q_{\text{out}} = w_{b,\text{in}} - (u_2 - u_1) = w_{b,\text{in}} + u_{fg} = 196.0 + 1743.7 = \mathbf{1940\ kJ/kg}$$

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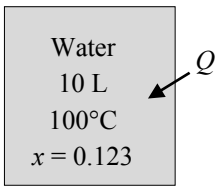
4-35 Water contained in a rigid vessel is heated. The heat transfer is to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved **3** The thermal energy stored in the vessel itself is negligible.

Analysis We take water as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

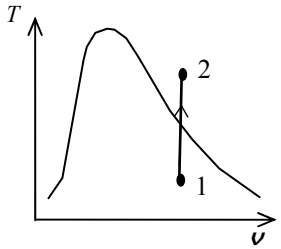
$$Q_{\text{in}} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$



The properties at the initial and final states are (Table A-4)

$$\left.\begin{array}{l} T_1 = 100°\text{C} \\ x_1 = 0.123 \end{array}\right\} \begin{array}{l} v_1 = v_f + x v_{fg} = 0.001043 + (0.123)(1.6720 - 0.001043) = 0.2066\ \text{m}^3/\text{kg} \\ u_1 = u_f + x u_{fg} = 419.06 + (0.123)(2087.0) = 675.76\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_2 = 150°\text{C} \\ v_2 = v_1 = 0.2066\ \text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{v_2 - v_f}{v_{fg}} = \dfrac{0.2066\text{-}0.001091}{0.39248\text{-}0.001091} = 0.5250 \\ u_2 = u_f + x_2 u_{fg} \\ \quad = 631.66 + (0.5250)(1927.4) = 1643.5\ \text{kJ/kg} \end{array}$$



The mass in the system is

$$m = \frac{V_1}{v_1} = \frac{0.100\ \text{m}^3}{0.2066\ \text{m}^3/\text{kg}} = 0.04841\ \text{kg}$$

Substituting,

$$Q_{\text{in}} = m(u_2 - u_1) = (0.04841\,\text{kg})(1643.5 - 675.76)\ \text{kJ/kg} = \mathbf{46.9\ kJ}$$

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4-36 Saturated vapor water is cooled at constant temperature (and pressure) to a saturated liquid. The heat rejected is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

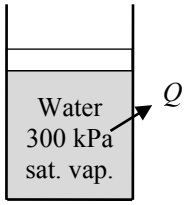
$$-Q_{\text{out}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

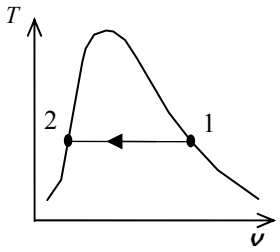
$$-Q_{\text{out}} = W_{b,\text{out}} + m(u_2 - u_1)$$

$$-Q_{\text{out}} = m(h_2 - h_1)$$

$$Q_{\text{out}} = m(h_1 - h_2)$$

$$q_{\text{out}} = h_1 - h_2$$





since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Since water changes from saturated liquid to saturated vapor, we have

$$q_{\text{out}} = h_g - h_f = h_{fg\,@\,300\text{ kPa}} = \mathbf{2163.5\ kJ/kg} \quad \text{(Table A-5)}$$

Note that the temperature also remains constant during the process and it is the saturation temperature at 300 kPa, which is 133.5°C.

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4-37 Saturated vapor water is cooled at constant pressure to a saturated liquid. The heat transferred and the work done are to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

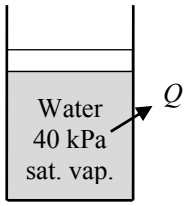
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-q_{\text{out}} - w_{b,\text{out}} = \Delta u = u_2 - u_1 \quad (\text{since KE} = \text{PE} = 0)$$

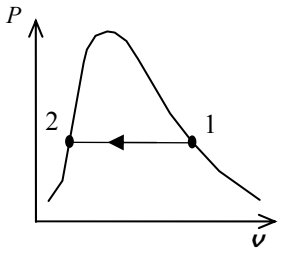
$$-q_{\text{out}} = w_{b,\text{out}} + (u_2 - u_1)$$

$$-q_{\text{out}} = h_2 - h_1$$

$$q_{\text{out}} = h_1 - h_2$$



since $\Delta u + w_b = \Delta h$ during a constant pressure quasi-equilibrium process. Since water changes from saturated liquid to saturated vapor, we have



$$q_{\text{out}} = h_g - h_f = h_{fg\,@\,40\,\text{kPa}} = \mathbf{2318.4\ kJ/kg} \quad \text{(Table A-5)}$$

The specific volumes at the initial and final states are

$$v_1 = v_{g\,@\,40\,\text{kPa}} = 3.993\ \text{m}^3/\text{kg}$$

$$v_2 = v_{f\,@\,40\,\text{kPa}} = 0.001026\ \text{m}^3/\text{kg}$$

Then the work done is determined from

$$w_{b,\text{out}} = \int_1^2 P\,dV = P(v_2 - v_1) = (40\,\text{kPa})(0.001026 - 3.9933)\text{m}^3\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) = \mathbf{159.7\ kJ/kg}$$

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4-38 A cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically as it is stirred by a paddle-wheel at constant pressure. The voltage of the current source is to be determined, and the process is to be shown on a P-v diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

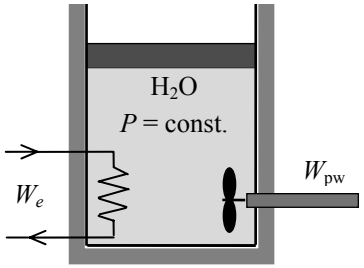
Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} + W_{\text{pw,in}} - W_{\text{b,out}} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{\text{e,in}} + W_{\text{pw,in}} = m(h_2 - h_1)$$

$$(\mathbf{V}I\Delta t) + W_{\text{pw,in}} = m(h_2 - h_1)$$

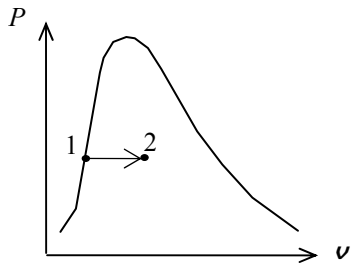


since $\Delta U + W_{\text{b}} = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 175 \text{ kPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_1 = h_{f@175\text{ kPa}} = 487.01 \text{ kJ/kg} \\ v_1 = v_{f@175\text{ kPa}} = 0.001057 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 175 \text{ kPa} \\ x_2 = 0.5 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 487.01 + (0.5 \times 2213.1) = 1593.6 \text{ kJ/kg}$$

$$m = \frac{V_1}{v_1} = \frac{0.005 \text{ m}^3}{0.001057 \text{ m}^3\text{/kg}} = 4.731 \text{ kg}$$



Substituting,

$$\mathbf{V}I\Delta t + (400\text{kJ}) = (4.731 \text{ kg})(1593.6 - 487.01)\text{kJ/kg}$$

$$\mathbf{V}I\Delta t = 4835 \text{ kJ}$$

$$\mathbf{V} = \frac{4835 \text{ kJ}}{(8 \text{ A})(45 \times 60 \text{ s})}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right) = \mathbf{223.9\ V}$$

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4-39 A cylinder equipped with an external spring is initially filled with steam at a specified state. Heat is transferred to the steam, and both the temperature and pressure rise. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a P-v diagram.

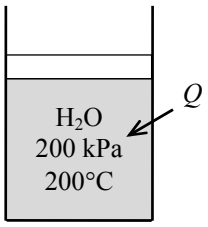
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium. **4** The spring is a linear spring.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Noting that the spring is not part of the system (it is external), the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$Q_{\text{in}} = m(u_2 - u_1) + W_{b,\text{out}}$$



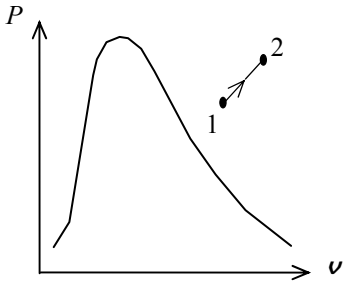
The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.4 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.3702 \text{ kg}$$

$$v_2 = \frac{V_2}{m} = \frac{0.6 \text{ m}^3}{0.3702 \text{ kg}} = 1.6207 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 250 \text{ kPa} \\ v_2 = 1.6207 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{606°C} \\ u_2 = 3312.0 \text{ kJ/kg} \end{array}$$



(*b*) The pressure of the gas changes linearly with volume, and thus the process curve on a P-V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 250)\text{kPa}}{2}(0.6 - 0.4)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{45 \text{ kJ}}$$

(*c*) From the energy balance we have

$$Q_{\text{in}} = (0.3702 \text{ kg})(3312.0 - 2654.6)\text{kJ/kg} + 45 \text{ kJ} = \mathbf{288 \text{ kJ}}$$

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4-40 Problem 4-39 is reconsidered. The effect of the initial temperature of steam on the final temperature, the work done, and the total heat transfer as the initial temperature varies from 150°C to 250°C is to be investigated. The final results are to be plotted against the initial temperature.

Analysis The problem is solved using EES, and the solution is given below.

```
"The process is given by:"
"P[2]=P[1]+k*x*A/A, and as the spring moves 'x' amount, the volume changes by V[2]-V[1]."
P[2]=P[1]+(Spring_const)*(V[2] - V[1]) "P[2] is a linear function of V[2]"
"where Spring_const = k/A, the actual spring constant divided by the piston face area"

"Conservation of mass for the closed system is:"
m[2]=m[1]

"The conservation of energy for the closed system is"
"E_in - E_out = DeltaE,  neglect DeltaKE and DeltaPE for the system"
Q_in - W_out = m[1]*(u[2]-u[1])
DELTAU=m[1]*(u[2]-u[1])

"Input Data"
P[1]=200 [kPa]
V[1]=0.4 [m^3]
T[1]=200 [C]
P[2]=250 [kPa]
V[2]=0.6 [m^3]

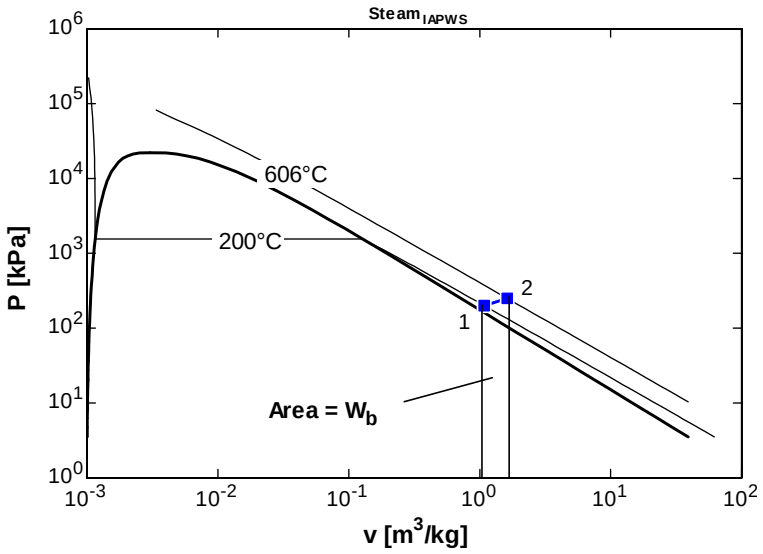
Fluid$='Steam_IAPWS'

m[1]=V[1]/spvol[1]
spvol[1]=volume(Fluid$,T=T[1], P=P[1])
u[1]=intenergy(Fluid$, T=T[1], P=P[1])
spvol[2]=V[2]/m[2]

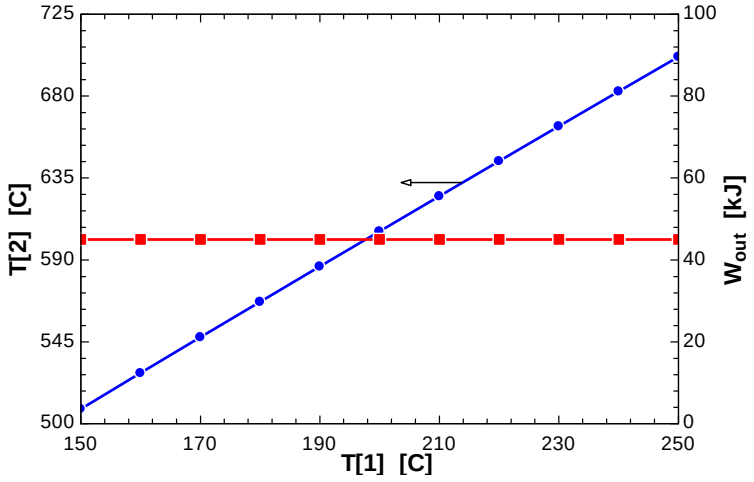
"The final temperature is:"
T[2]=temperature(Fluid$,P=P[2],v=spvol[2])

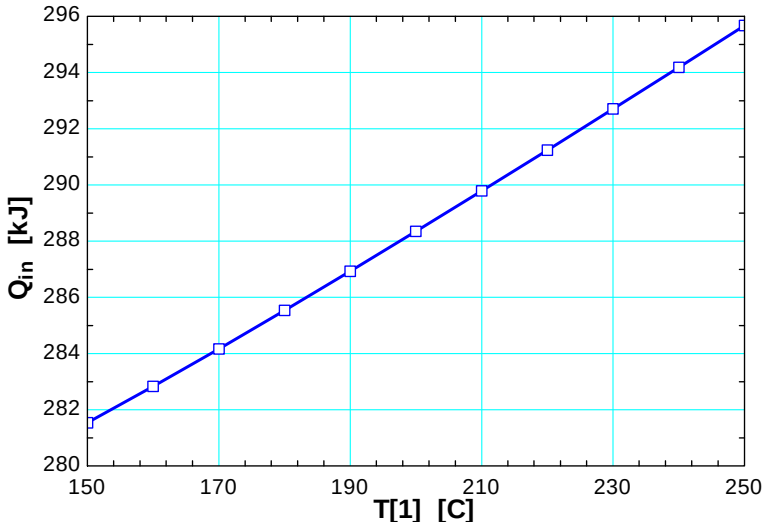
u[2]=intenergy(Fluid$, P=P[2], T=T[2])
Wnet_other = 0
W_out=Wnet_other + W_b
"W_b = integral of P[2]*dV[2] for 0.5<V[2]<0.6 and is given by:"
 W_b=P[1]*(V[2]-V[1])+Spring_const/2*(V[2]-V[1])^2
```

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T_1 [C]	Q_{in} [kJ]	T_2 [C]	W_{out} [kJ]
150	281.5	508.2	45
160	282.8	527.9	45
170	284.2	547.5	45
180	285.5	567	45
190	286.9	586.4	45
200	288.3	605.8	45
210	289.8	625	45
220	291.2	644.3	45
230	292.7	663.4	45
240	294.2	682.6	45
250	295.7	701.7	45





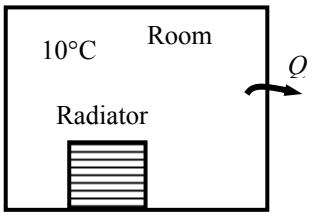
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4-41 A room is heated by an electrical radiator containing heating oil. Heat is lost from the room. The time period during which the heater is on is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa. **5** The room is air-tight so that no air leaks in and out during the process.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). Also, $c_v = 0.718$ kJ/kg.K for air at room temperature (Table A-2). Oil properties are given to be $\rho = 950$ kg/m^3 and $c_p = 2.2$ kJ/kg.°C.

Analysis We take the air in the room and the oil in the radiator to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$(\dot{W}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = \Delta U_{\text{air}} + \Delta U_{\text{oil}}$$
$$\cong [mc_v(T_2 - T_1)]_{\text{air}} + [mc_p(T_2 - T_1)]_{\text{oil}} \quad (\text{since KE} = \text{PE} = 0)$$

The mass of air and oil are

$$m_{\text{air}} = \frac{P\boldsymbol{V}_{\text{air}}}{RT_1} = \frac{(100\ \text{kPa})(50\ \text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(10+273\ \text{K})} = 62.32\ \text{kg}$$

$$m_{\text{oil}} = \rho_{\text{oil}}\boldsymbol{V}_{\text{oil}} = (950\ \text{kg/m}^3)(0.030\ \text{m}^3) = 28.50\ \text{kg}$$

Substituting,

$$(1.8 - 0.35\ \text{kJ/s})\Delta t = (62.32\ \text{kg})(0.718\ \text{kJ/kg}\cdot°\text{C})(20-10)°\text{C} + (28.50\ \text{kg})(2.2\ \text{kJ/kg}\cdot°\text{C})(50-10)°\text{C}$$

$$\longrightarrow \Delta t = \mathbf{2038\ s} = \mathbf{34.0\ min}$$

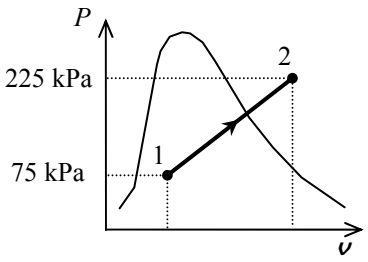
Discussion In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to using ΔH instead of use ΔU in heating and air-conditioning applications.

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4-42 A saturated water mixture contained in a spring-loaded piston-cylinder device is heated until the pressure and volume rise to specified values. The heat transfer and the work done are to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{\text{in}} = W_{b,out} + m(u_2 - u_1)$$

The initial state is saturated mixture at 75 kPa. The specific volume and internal energy at this state are (Table A-5),

$$v_1 = v_f + x v_{fg} = 0.001037 + (0.08)(2.2172 - 0.001037) = 0.1783 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x u_{fg} = 384.36 + (0.08)(2111.8) = 553.30 \text{ kJ/kg}$$

The mass of water is

$$m = \frac{V_1}{v_1} = \frac{2 \text{ m}^3}{0.1783 \text{ m}^3/\text{kg}} = 11.22 \text{ kg}$$

The final specific volume is

$$v_2 = \frac{V_2}{m} = \frac{5 \text{ m}^3}{11.22 \text{ kg}} = 0.4458 \text{ m}^3/\text{kg}$$

The final state is now fixed. The internal energy at this specific volume and 225 kPa pressure is (Table A-6)

$$u_2 = 1650.4 \text{ kJ/kg}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(75 + 225)\text{kPa}}{2}(5 - 2)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{450 \text{ kJ}}$$

Substituting into energy balance equation gives

$$Q_{\text{in}} = W_{b,\text{out}} + m(u_2 - u_1) = 450 \text{ kJ} + (11.22 \text{ kg})(1650.4 - 553.30) \text{ kJ/kg} = \mathbf{12{,}750 \text{ kJ}}$$

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4-43 R-134a contained in a spring-loaded piston-cylinder device is cooled until the temperature and volume drop to specified values. The heat transfer and the work done are to be determined.

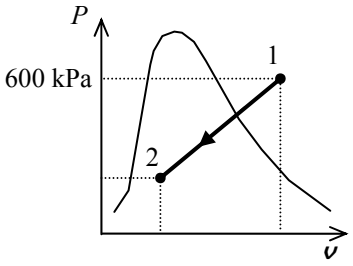
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$



The initial state properties are (Table A-13)

$$\left.\begin{aligned} P_1 &= 600 \text{ kPa} \\ T_1 &= 15°\text{C} \end{aligned}\right\} \begin{aligned} v_1 &= 0.055522 \text{ m}^3/\text{kg} \\ u_1 &= 357.96 \text{ kJ/kg} \end{aligned}$$

The mass of refrigerant is

$$m = \frac{V_1}{v_1} = \frac{0.3 \text{ m}^3}{0.055522 \text{ m}^3/\text{kg}} = 5.4033 \text{ kg}$$

The final specific volume is

$$v_2 = \frac{V_2}{m} = \frac{0.1 \text{ m}^3}{5.4033 \text{ kg}} = 0.018507 \text{ m}^3/\text{kg}$$

The final state at this specific volume and at -30°C is a saturated mixture. The properties at this state are (Table A-11)

$$x_2 = \frac{v_2 - v_f}{v_g - v_f} = \frac{0.018507 - 0.0007203}{0.22580 - 0.0007203} = 0.079024$$

$$u_2 = u_f + x_2 u_{fg} = 12.59 + (0.079024)(200.52) = 28.44 \text{ kJ/kg}$$

$$P_2 = 84.43 \text{ kPa}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$W_{b,\text{in}} = \text{Area} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(600 + 84.43)\text{kPa}}{2}(0.3 - 0.1)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{68.44\ kJ}$$

Substituting into energy balance equation gives

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1) = 68.44 \text{ kJ} - (5.4033 \text{ kg})(28.44 - 357.96) \text{ kJ/kg} = \mathbf{1849\ kJ}$$

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4-44E Saturated R-134a vapor is condensed at constant pressure to a saturated liquid in a piston-cylinder device. The heat transfer and the work done are to be determined.

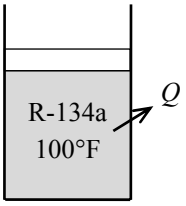
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

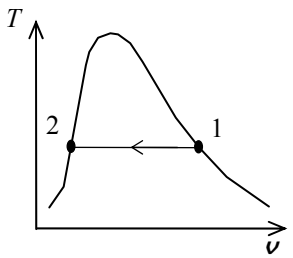
$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$



The properties at the initial and final states are (Table A-11E)

$$\left.\begin{aligned} T_1 &= 100°\text{F} \\ x_1 &= 1 \end{aligned}\right\} \begin{aligned} v_1 &= v_g = 0.34045\ \text{ft}^3/\text{lbm} \\ u_1 &= u_g = 107.45\ \text{Btu/lbm} \end{aligned}$$

$$\left.\begin{aligned} T_2 &= 100°\text{F} \\ x_2 &= 0 \end{aligned}\right\} \begin{aligned} v_2 &= v_f = 0.01386\ \text{ft}^3/\text{lbm} \\ u_2 &= u_f = 44.768\ \text{Btu/lbm} \end{aligned}$$



Also from Table A-11E,

$$P_1 = P_2 = 138.93\ \text{psia}$$
$$u_{fg} = 62.683\ \text{Btu/lbm}$$
$$h_{fg} = 71.080\ \text{Btu/lbm}$$

The work done during this process is

$$w_{b,\text{out}} = \int_1^2 P\,dv = P(v_2 - v_1) = (138.93\text{psia})(0.01386 - 0.34045)\text{ft}^3/\text{lbm}\left(\frac{1\,\text{Btu}}{5.404\,\text{psia}\cdot\text{ft}^3}\right) = -8.396\,\text{Btu/lbm}$$

That is,

$$w_{b,\text{in}} = \mathbf{8.396\ Btu/lbm}$$

Substituting into energy balance equation gives

$$q_{\text{out}} = w_{b,\text{in}} - (u_2 - u_1) = w_{b,\text{in}} + u_{fg} = 8.396 + 62.683 = \mathbf{71.080\ Btu/lbm}$$

Discussion The heat transfer may also be determined from

$$-q_{\text{out}} = h_2 - h_1$$
$$q_{\text{out}} = h_{fg} = 71.080\,\text{Btu/lbm}$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.

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4-45 Saturated R-134a liquid is contained in an insulated piston-cylinder device. Electrical work is supplied to R-134a. The time required for the refrigerant to turn into saturated vapor and the final temperature are to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

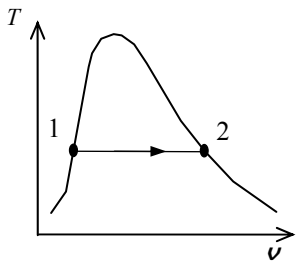
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$W_{e,\text{in}} = W_{b,\text{out}} + m(u_2 - u_1)$$

$$W_{e,\text{in}} = H_2 - H_1 = m(h_2 - h_1) = mh_{fg}$$

$$\dot{W}_{e,\text{in}}\Delta t = mh_{fg}$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The electrical power and the enthalpy of vaporization of R-134a are

$$\dot{W}_{e,\text{in}} = \mathbf{V}I = (10\text{ V})(2\text{ A}) = 20\text{ W}$$

$$h_{fg\,@-5°\text{C}} = 202.34\text{ kJ/kg} \quad (\text{Table A-11})$$

Substituting,

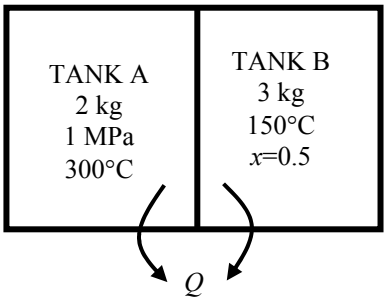
$$(0.020\text{ kJ/s})\Delta t = (2\text{ kg})(202.34\text{kJ/kg}) \longrightarrow \Delta t = 8093\text{ s} = \mathbf{2.25\,h}$$

The temperature remains constant during this phase change process:

$$T_2 = T_1 = \mathbf{-5°C}$$

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4-46 Two tanks initially separated by a partition contain steam at different states. Now the partition is removed and they are allowed to mix until equilibrium is established. The temperature and quality of the steam at the final state and the amount of heat lost from the tanks are to be determined.



Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

Analysis (*a*) We take the contents of both tanks as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_A + \Delta U_B = \left[m(u_2 - u_1)\right]_A + \left[m(u_2 - u_1)\right]_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

The properties of steam in both tanks at the initial state are (Tables A-4 through A-6)

$$\left.\begin{aligned} P_{1,A} &= 1000 \text{ kPa} \\ T_{1,A} &= 300°\text{C} \end{aligned}\right\} \begin{aligned} v_{1,A} &= 0.25799 \text{ m}^3/\text{kg} \\ u_{1,A} &= 2793.7 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} T_{1,B} &= 150°\text{C} \\ x_1 &= 0.50 \end{aligned}\right\} \begin{aligned} v_f &= 0.001091, \quad v_g = 0.39248 \text{ m}^3/\text{kg} \\ u_f &= 631.66, \quad u_{fg} = 1927.4 \text{ kJ/kg} \end{aligned}$$

$$v_{1,B} = v_f + x_1 v_{fg} = 0.001091 + \left[0.50 \times (0.39248 - 0.001091)\right] = 0.19679 \text{ m}^3/\text{kg}$$

$$u_{1,B} = u_f + x_1 u_{fg} = 631.66 + (0.50 \times 1927.4) = 1595.4 \text{ kJ/kg}$$

The total volume and total mass of the system are

$$V = V_A + V_B = m_A v_{1,A} + m_B v_{1,B} = (2 \text{ kg})(0.25799 \text{ m}^3/\text{kg}) + (3 \text{ kg})(0.19679 \text{ m}^3/\text{kg}) = 1.106 \text{ m}^3$$

$$m = m_A + m_B = 3 + 2 = 5 \text{ kg}$$

Now, the specific volume at the final state may be determined

$$v_2 = \frac{V}{m} = \frac{1.106 \text{ m}^3}{5 \text{ kg}} = 0.22127 \text{ m}^3/\text{kg}$$

which fixes the final state and we can determine other properties

$$\left.\begin{aligned} P_2 &= 300 \text{ kPa} \\ v_2 &= 0.22127 \text{ m}^3/\text{kg} \end{aligned}\right\} \begin{aligned} T_2 &= T_{\text{sat @ 300 kPa}} = \mathbf{133.5°C} \\ x_2 &= \frac{v_2 - v_f}{v_g - v_f} = \frac{0.22127 - 0.001073}{0.60582 - 0.001073} = \mathbf{0.3641} \\ u_2 &= u_f + x_2 u_{fg} = 561.11 + (0.3641 \times 1982.1) = 1282.8 \text{ kJ/kg} \end{aligned}$$

(*b*) Substituting,

$$\begin{aligned} -Q_{\text{out}} &= \Delta U_A + \Delta U_B = \left[m(u_2 - u_1)\right]_A + \left[m(u_2 - u_1)\right]_B \\ &= (2 \text{ kg})(1282.8 - 2793.7)\text{kJ/kg} + (3 \text{ kg})(1282.8 - 1595.4)\text{kJ/kg} = -3959 \text{ kJ} \end{aligned}$$

or

$$Q_{\text{out}} = \mathbf{3959 \text{ kJ}}$$

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Specific Heats, Δu and Δh of Ideal Gases

4-47C It can be either. The difference in temperature in both the K and °C scales is the same.

4-48C It can be used for any kind of process of an ideal gas.

4-49C It can be used for any kind of process of an ideal gas.

4-50C Very close, but no. Because the heat transfer during this process is $Q = mc_p\Delta T$, and c_p varies with temperature.

4-51C The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the c_p of an ideal gas does not vary with pressure.

4-52C The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the c_p of an ideal gas does not vary with volume.

4-53C Modeling both gases as ideal gases with constant specific heats, we have

$$\Delta u = c_v \Delta T$$
$$\Delta h = c_p \Delta T$$

Since both gases undergo the same temperature change, the gas with the greatest c_v will experience the largest change in internal energy. Similarly, the gas with the largest c_p will have the greatest enthalpy change. Inspection of Table A-2*a* indicates that air will experience the greatest change in both cases.

4-54 The desired result is obtained by multiplying the first relation by the molar mass M,

$$Mc_p = Mc_v + MR$$

or $$\bar{c}_p = \bar{c}_v + R_u$$

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4-55 The enthalpy change of oxygen is to be determined for two cases of specified temperature changes.

Assumptions At specified conditions, oxygen behaves as an ideal gas.

Properties The constant-pressure specific heat of oxygen at room temperature is $c_p = 0.918$ kJ/kg·K (Table A-2a).

Analysis Using the specific heat at constant pressure,

$$\Delta h = c_p \Delta T = (0.918 \text{ kJ/kg} \cdot \text{K})(200 - 150)\text{K} = \mathbf{45.9\ kJ/kg}$$

If we use the same room temperature specific heat value, the enthalpy change will be the same for the second case. However, if we consider the variation of specific heat with temperature and use the specific heat values from Table A-2b, we have $c_p = 0.956$ kJ/kg·K at 175°C (≅ 450 K) and $c_p = 0.918$ kJ/kg·K at 25°C (≅ 300 K). Then,

$$\Delta h_1 = c_p \Delta T_1 = (0.956 \text{ kJ/kg} \cdot \text{K})(200 - 150)\text{K} = \mathbf{47.8\ kJ/kg}$$

$$\Delta h_2 = c_p \Delta T_1 = (0.918 \text{ kJ/kg} \cdot \text{K})(50 - 0)\text{K} = \mathbf{45.9\ kJ/kg}$$

The two results differ from each other by about 4%. The pressure has no influence on the enthalpy of an ideal gas.

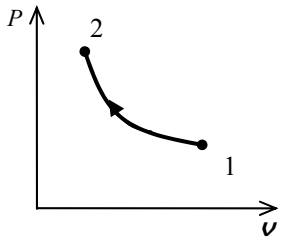
4-56E Air is compressed isothermally in a compressor. The change in the specific volume of air is to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.3704$ psia·ft^3/lbm·R (Table A-1E).

Analysis At the compressor inlet, the specific volume is

$$v_1 = \frac{RT}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(70 + 460 \text{ R})}{20 \text{ psia}} = 9.816 \text{ ft}^3/\text{lbm}$$



Similarly, at the compressor exit,

$$v_2 = \frac{RT}{P_2} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(70 + 460 \text{ R})}{150 \text{ psia}} = 1.309 \text{ ft}^3/\text{lbm}$$

The change in the specific volume caused by the compressor is

$$\Delta v = v_2 - v_1 = 1.309 - 9.816 = \mathbf{-8.51\ ft^3/lbm}$$

4-57 The total internal energy changes for neon and argon are to be determined for a given temperature change.

Assumptions At specified conditions, neon and argon behave as an ideal gas.

Properties The constant-volume specific heats of neon and argon are 0.6179 kJ/kg·K and 0.3122 kJ/kg·K, respectively (Table A-2*a*).

Analysis The total internal energy changes are

$$\Delta U_{\text{neon}} = mc_v \Delta T = (2 \text{ kg})(0.6179 \text{ kJ/kg} \cdot \text{K})(180 - 20)\text{K} = \mathbf{197.7\ kJ}$$

$$\Delta U_{\text{argon}} = mc_v \Delta T = (2 \text{ kg})(0.3122 \text{ kJ/kg} \cdot \text{K})(180 - 20)\text{K} = \mathbf{99.9\ kJ}$$

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4-58 The enthalpy changes for neon and argon are to be determined for a given temperature change.

Assumptions At specified conditions, neon and argon behave as an ideal gas.

Properties The constant-pressure specific heats of argon and neon are 0.5203 kJ/kg·K and 1.0299 kJ/kg·K, respectively (Table A-2*a*).

Analysis The enthalpy changes are

$$\Delta h_{\text{argon}} = c_p \Delta T = (0.5203 \text{ kJ/kg} \cdot \text{K})(25-100)\text{K} = \mathbf{-39.0 \text{ kJ/kg}}$$
$$\Delta h_{\text{neon}} = c_p \Delta T = (1.0299 \text{ kJ/kg} \cdot \text{K})(25-100)\text{K} = \mathbf{-77.2 \text{ kJ/kg}}$$

4-59E The enthalpy change of oxygen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

Analysis (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2Ec,

$$\bar{c}_p = a + bT + cT^2 + dT^3$$

where a = 6.085, b = 0.2017×10^{-2}, c = -0.05275×10^{-5}, and d = 0.05372×10^{-9}. Then,

$$\begin{aligned}\Delta\bar{h} &= \int_1^2 \bar{c}_p(T)dT = \int_1^2 \left[a + bT + cT^2 + dT^3\right]dT \\ &= a(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 + T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\ &= 6.085(1500-800) + \tfrac{1}{2}(0.2017\times10^{-2})(1500^2 - 800^2) \\ &\quad - \tfrac{1}{3}(0.05275\times10^{-5})(1500^3 - 800^3) + \tfrac{1}{4}(0.05372\times10^{-9})(1500^4 - 800^4) \\ &= 5442.3 \text{ Btu/lbmol}\end{aligned}$$

$$\Delta h = \frac{\Delta\bar{h}}{M} = \frac{5442.3 \text{ Btu/lbmol}}{31.999 \text{ lbm/lbmol}} = \mathbf{170 \text{ Btu/lbm}}$$

(*b*) Using the constant c_p value from Table A-2Eb at the average temperature of 1150 R,

$$c_{p,\text{avg}} = c_{p@1150\text{ R}} = 0.242 \text{ Btu/lbm} \cdot \text{R}$$
$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (0.242 \text{ Btu/lbm} \cdot \text{R})(1500-800)\text{ R} = \mathbf{169 \text{ Btu/lbm}}$$

(*c*) Using the constant c_p value from Table A-2Ea at room temperature,

$$c_{p,\text{avg}} = c_{p@537\text{ R}} = 0.219 \text{ Btu/lbm} \cdot \text{R}$$
$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (0.219 \text{ Btu/lbm} \cdot \text{R})(1500-800)\text{R} = \mathbf{153 \text{ Btu/lbm}}$$

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4-60 The internal energy change of hydrogen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

Analysis (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2c and relating it to $\bar{c}_v(T)$,

$$\bar{c}_v(T) = \bar{c}_p - R_u = (a - R_u) + bT + cT^2 + dT^3$$

where a = 29.11, b = -0.1916×10^{-2}, c = 0.4003×10^{-5}, and d = -0.8704×10^{-9}. Then,

$$\begin{aligned}
\Delta\bar{u} &= \int_1^2 \bar{c}_v(T)dT = \int_1^2 \left[(a - R_u) + bT + cT^2 + dT^3\right]dT \\
&= (a - R_u)(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 + T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\
&= (29.11 - 8.314)(800 - 200) - \tfrac{1}{2}(0.1961\times10^{-2})(800^2 - 200^2) \\
&\quad + \tfrac{1}{3}(0.4003\times10^{-5})(800^3 - 200^3) - \tfrac{1}{4}(0.8704\times10^{-9})(800^4 - 200^4) \\
&= 12{,}487 \text{ kJ/kmol} \\
\Delta u &= \frac{\Delta\bar{u}}{M} = \frac{12{,}487 \text{ kJ/kmol}}{2.016 \text{ kg/kmol}} = \mathbf{6194 \text{ kJ/kg}}
\end{aligned}$$

(*b*) Using a constant c_p value from Table A-2b at the average temperature of 500 K,

$$\begin{aligned}
c_{v,\text{avg}} &= c_{v@500\text{ K}} = 10.389 \text{ kJ/kg}\cdot\text{K} \\
\Delta u &= c_{v,\text{avg}}(T_2 - T_1) = (10.389 \text{ kJ/kg}\cdot\text{K})(800 - 200)\text{K} = \mathbf{6233 \text{ kJ/kg}}
\end{aligned}$$

(*c*) Using a constant c_p value from Table A-2a at room temperature,

$$\begin{aligned}
c_{v,\text{avg}} &= c_{v@300\text{ K}} = 10.183 \text{ kJ/kg}\cdot\text{K} \\
\Delta u &= c_{v,\text{avg}}(T_2 - T_1) = (10.183 \text{ kJ/kg}\cdot\text{K})(800 - 200)\text{K} = \mathbf{6110 \text{ kJ/kg}}
\end{aligned}$$

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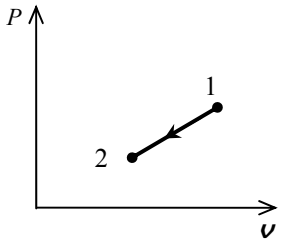
4-61E A spring-loaded piston-cylinder device is filled with air. The air is now cooled until its volume decreases by 50%. The changes in the internal energy and enthalpy of the air are to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.3704$ psia·ft^3/lbm·R (Table A-1E). The specific heats of air at room temperature are $c_v = 0.171$ Btu/lbm·R and $c_p = 0.240$ Btu/lbm·R (Table A-2Ea).

Analysis The mass of the air in this system is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(250 \text{ psia})(1 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(460 + 460 \text{ R})} = 0.7336 \text{ lbm}$$



The final specific volume is then

$$v_2 = \frac{V_2}{m} = \frac{0.5(1 \text{ ft}^3)}{0.7336 \text{ lbm}} = 0.6816 \text{ ft}^3/\text{lbm}$$

As the volume of the air decreased, the length of the spring will increase by

$$\Delta x = \frac{\Delta V}{A_p} = \frac{\Delta V}{\pi D^2 / 4} = \frac{4(0.5 \text{ ft}^3)}{\pi (10/12 \text{ ft})^2} = 0.9167 \text{ ft} = 11.00 \text{ in}$$

The final pressure is then

$$\begin{aligned} P_2 &= P_1 - \Delta P = P_1 - \frac{\Delta F}{A_p} = P_1 - \frac{k\Delta x}{A_p} = P_1 - \frac{k\Delta x}{\pi D^2 / 4} \\ &= 250 \text{ psia} - \frac{4(5 \text{ lbf/in})(11.00 \text{ in})}{\pi (10 \text{ in})^2} = 249.3 \text{ lbf/in}^2 \\ &= 249.3 \text{ psia} \end{aligned}$$

Employing the ideal gas equation of state, the final temperature will be

$$T_2 = \frac{P_2 V_2}{mR} = \frac{(249.3 \text{ psia})(0.5 \text{ ft}^3)}{(0.7336 \text{ lbm})(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})} = 458.7 \text{ R}$$

Using the specific heats,

$$\Delta u = c_v \Delta T = (0.171 \text{ Btu/lbm} \cdot \text{R})(458.7 - 920)\text{R} = \mathbf{-78.9 \text{ Btu/lbm}}$$

$$\Delta h = c_p \Delta T = (0.240 \text{ Btu/lbm} \cdot \text{R})(458.7 - 920)\text{R} = \mathbf{-110.7 \text{ Btu/lbm}}$$

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Closed System Energy Analysis: Ideal Gases

4-62C No, it isn't. This is because the first law relation $Q - W = \Delta U$ reduces to $W = 0$ in this case since the system is adiabatic ($Q = 0$) and $\Delta U = 0$ for the isothermal processes of ideal gases. Therefore, this adiabatic system cannot receive any net work at constant temperature.

4-63 Oxygen is heated to experience a specified temperature change. The heat transfer is to be determined for two cases.

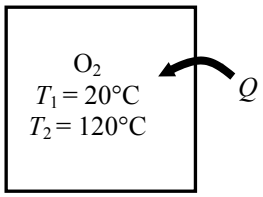
Assumptions **1** Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 154.8 K and 5.08 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for oxygen.

Properties The specific heats of oxygen at the average temperature of (20+120)/2=70°C=343 K are $c_p = 0.927$ kJ/kg·K and $c_v = 0.667$ kJ/kg·K (Table A-2*b*).

Analysis We take the oxygen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for a constant-volume process can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = mc_v(T_2 - T_1)$$



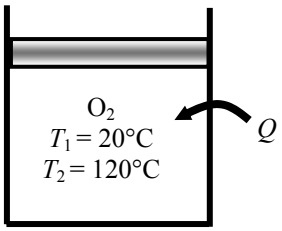
The energy balance during a constant-pressure process (such as in a piston-cylinder device) can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U$$

$$Q_{\text{in}} = W_{b,\text{out}} + \Delta U$$

$$Q_{\text{in}} = \Delta H = mc_p(T_2 - T_1)$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting for both cases,

$$Q_{\text{in, }v=\text{const}} = mc_v(T_2 - T_1) = (1\,\text{kg})(0.667\,\text{kJ/kg}\cdot\text{K})(120 - 20)\text{K} = \mathbf{66.7\ kJ}$$

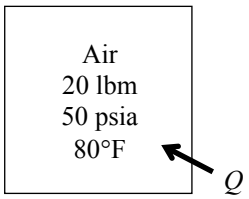
$$Q_{\text{in, }P=\text{const}} = mc_p(T_2 - T_1) = (1\,\text{kg})(0.927\,\text{kJ/kg}\cdot\text{K})(120 - 20)\text{K} = \mathbf{92.7\ kJ}$$

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4-64E The air in a rigid tank is heated until its pressure doubles. The volume of the tank and the amount of heat transfer are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -221°F and 547 psia. **2** The kinetic and potential energy changes are negligible, $\Delta pe \cong \Delta ke \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

Properties The gas constant of air is $R = 0.3704$ psia.ft^3/lbm.R (Table A-1E).



Analysis (*a*) The volume of the tank can be determined from the ideal gas relation,

$$V = \frac{mRT_1}{P_1} = \frac{(20\ \text{lbm})(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540\ \text{R})}{50\ \text{psia}} = \mathbf{80.0\ ft^3}$$

(b) We take the air in the tank as our system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U$$

$$Q_{\text{in}} = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$

The final temperature of air is

$$\frac{P_1V}{T_1} = \frac{P_2V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = 2 \times (540\ \text{R}) = 1080\ \text{R}$$

The internal energies are (Table A-17E)

$$u_1 = u_{@\,540\ \text{R}} = 92.04\ \text{Btu / lbm}$$

$$u_2 = u_{@\,1080\ \text{R}} = 186.93\ \text{Btu / lbm}$$

Substituting,

$$Q_{\text{in}} = (20\ \text{lbm})(186.93 - 92.04)\text{Btu/lbm} = \mathbf{1898\ Btu}$$

Alternative solutions The specific heat of air at the average temperature of $T_{\text{avg}} = (540+1080)/2 = 810$ R = 350°F is, from Table A-2Eb, $c_{v,\text{avg}} = 0.175$ Btu/lbm.R. Substituting,

$$Q_{\text{in}} = (20\ \text{lbm})(0.175\ \text{Btu/lbm.R})(1080 - 540)\ \text{R} = \mathbf{1890\ Btu}$$

Discussion Both approaches resulted in almost the same solution in this case.

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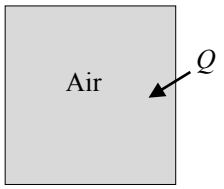
4-65E Heat is transferred to air contained in a rifid container. The internal energy change is to be determined.

Assumptions **1** Air is an ideal gas since it is probably at a high temperature and low pressure relative to its critical point values of 238.5 R and 547 psia. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air.

Analysis We take air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$q_{in} = \Delta u \quad (\text{since KE} = \text{PE} = 0)$$



Substituting,

$$\Delta u = q_{in} = \mathbf{50\ Btu/lbm}$$

4-66E Paddle Wheel work is applied to nitrogen in a rigid container. The final temperature is to be determined.

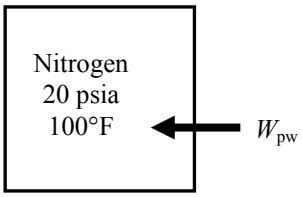
Assumptions **1** Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 126.2 K (227.1 R) and 3.39 MPa (492 psia). **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for nitrogen.

Properties For nitrogen, $c_v = 0.177$ Btu/lbm·R at room temperature and $R = 0.3830\ \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$ (Table A-1E and A-2E*a*).

Analysis We take the nitrogen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{pw,in} = \Delta U = mc_v(T_2 - T_1) \quad (\text{since KE} = \text{PE} = 0)$$



The mass of nitrogen is

$$m = \frac{P_1 V}{RT_1} = \frac{(20\ \text{psia})(1\ \text{ft}^3)}{(0.3830\ \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(560\ \text{R})} = 0.09325\ \text{lbm}$$

Substituting,

$$W_{pw,in} = mc_v(T_2 - T_1)$$

$$(5000\ \text{lbf} \cdot \text{ft})\frac{1\ \text{Btu}}{778.17\ \text{lbf} \cdot \text{ft}} = (0.09325\ \text{lbm})(0.177\ \text{Btu/lbm} \cdot \text{R})(T_2 - 560)\text{R}$$

$$T_2 = \mathbf{949\ R} = \mathbf{489°F}$$

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4-67 Nitrogen in a piston-cylinder device undergoes an isobaric process. The final pressure and the heat transfer are to be determined.

Assumptions **1** Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 126.2 K (227.1 R) and 3.39 MPa (492 psia). **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for nitrogen.

Properties For nitrogen, c_p = 1.039 kJ/kg·K at room temperature (Table A-2*a*).

Analysis Since this is an isobaric pressure, the pressure remains constant during the process and thus,

$$P_2 = P_1 = \mathbf{1MPa}$$

We take the nitrogen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

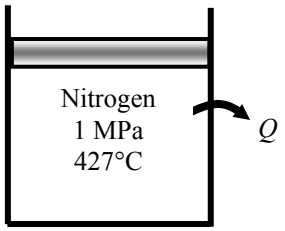
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-q_{\text{out}} - w_{b,out} = \Delta u = (u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

$$-q_{\text{out}} = w_{b,\text{out}} + (u_2 - u_1)$$

$$-q_{\text{out}} = h_2 - h_1$$

$$q_{\text{out}} = mc_p(T_1 - T_2)$$



since $\Delta u + w_b = \Delta h$ during a constant pressure (isobaric) quasi-equilibrium process. Substituting,

$$q_{\text{out}} = c_p(T_1 - T_2) = (1.039 \text{ kJ/kg} \cdot \text{°F})(427 - 27)\text{°F} = \mathbf{416\ kJ/kg}$$

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4-68 A resistance heater is to raise the air temperature in the room from 5 to 25°C within 11 min. The required power rating of the resistance heater is to be determined.

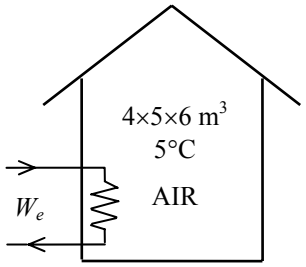
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** Heat losses from the room are negligible. **5** The room is air-tight so that no air leaks in and out during the process.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). Also, $c_v = 0.718$ kJ/kg.K for air at room temperature (Table A-2).

Analysis We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U \cong mc_{v,\text{avg}}(T_2 - T_1) \ \ (\text{since } Q = \text{KE} = \text{PE} = 0)$$



or,

$$\dot{W}_{e,\text{in}}\Delta t = mc_{v,\text{avg}}(T_2 - T_1)$$

The mass of air is

$$V = 4 \times 5 \times 6 = 120 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(278 \text{ K})} = 150.6 \text{ kg}$$

Substituting, the power rating of the heater becomes

$$\dot{W}_{e,\text{in}} = \frac{(150.6 \text{ kg})(0.718 \text{ kJ/kg} \cdot {}^\circ\text{C})(25-5)^\circ\text{C}}{11 \times 60 \text{ s}} = \mathbf{3.28 \text{ kW}}$$

Discussion In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to use ΔH instead of using ΔU in heating and air-conditioning applications.

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4-69 A student living in a room turns her 150-W fan on in the morning. The temperature in the room when she comes back 10 h later is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** All the doors and windows are tightly closed, and heat transfer through the walls and the windows is disregarded.

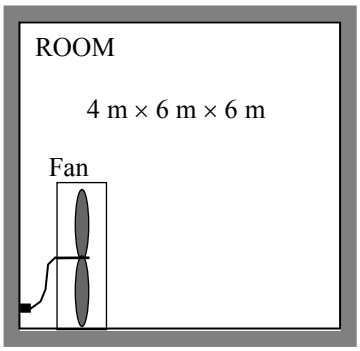
Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). Also, $c_v = 0.718$ kJ/kg.K for air at room temperature (Table A-2).

Analysis We take the room as the system. This is a *closed system* since the doors and the windows are said to be tightly closed, and thus no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} = \Delta U$$

$$W_{e,in} = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$



The mass of air is

$$V = 4 \times 6 \times 6 = 144 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(144 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 174.2 \text{ kg}$$

The electrical work done by the fan is

$$W_e = \dot{W}_e \Delta t = (0.15 \text{ kJ/s})(10 \times 3600 \text{ s}) = 5400 \text{ kJ}$$

Substituting and using the c_v value at room temperature,

$$5400 \text{ kJ} = (174.2 \text{ kg})(0.718 \text{ kJ/kg·°C})(T_2 - 15)°\text{C}$$

$$T_2 = \mathbf{58.2°C}$$

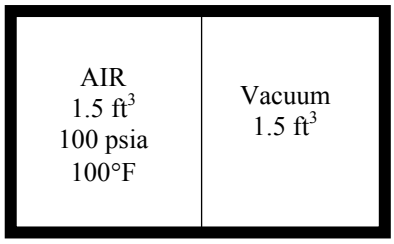
Discussion Note that a fan actually causes the internal temperature of a confined space to rise. In fact, a 100-W fan supplies a room with as much energy as a 100-W resistance heater.

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4-70E One part of an insulated rigid tank contains air while the other side is evacuated. The internal energy change of the air and the final air pressure are to be determined when the partition is removed.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of –221.5°F and 547 psia. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **3** The tank is insulated and thus heat transfer is negligible.

Analysis We take the entire tank as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as



$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$\mathbf{0} = \Delta U = mc_v(T_2 - T_1)$$

Since the internal energy does not change, the temperature of the air will also not change. Applying the ideal gas equation gives

$$P_1 V_1 = P_2 V_2 \longrightarrow P_2 = P_1 \frac{V_1}{V_2} = P_1 \frac{V_2/2}{V_2} = \frac{P_1}{2} = \frac{100 \text{ psia}}{2} = \mathbf{50 \text{ psia}}$$

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4-71 Air in a closed system undergoes an isothermal process. The initial volume, the work done, and the heat transfer are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 132.5 K and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for air.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-1).

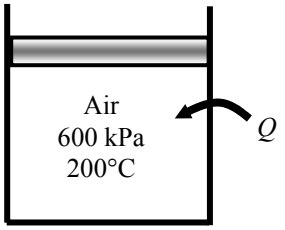
Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_{\nu}(T_2 - T_1)$$

$$Q_{\text{in}} - W_{b,\text{out}} = 0 \quad (\text{since } T_1 = T_2)$$

$$Q_{\text{in}} = W_{b,\text{out}}$$



The initial volume is

$$V_1 = \frac{mRT_1}{P_1} = \frac{(2\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\text{ K})}{600\text{ kPa}} = \mathbf{0.4525\ m^3}$$

Using the boundary work relation for the isothermal process of an ideal gas gives

$$W_{b,\text{out}} = m\int_1^2 P\,dv = mRT\int_1^2 \frac{dv}{v} = mRT\ln\frac{v_2}{v_1} = mRT\ln\frac{P_1}{P_2}$$

$$= (2\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\text{ K})\ln\frac{600\text{ kPa}}{80\text{ kPa}} = \mathbf{547.1\,kJ}$$

From energy balance equation,

$$Q_{\text{in}} = W_{b,\text{out}} = \mathbf{547.1\,kJ}$$

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4-72 Argon in a piston-cylinder device undergoes an isothermal process. The mass of argon and the work done are to be determined.

Assumptions **1** Argon is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 151 K and 4.86 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The gas constant of argon is $R = 0.2081$kJ/kg·K (Table A-1).

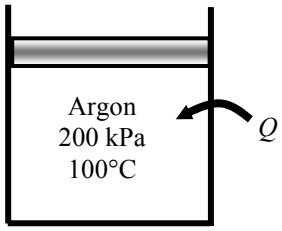
Analysis We take argon as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_{\upsilon}(T_2 - T_1)$$

$$Q_{\text{in}} - W_{b,\text{out}} = 0 \quad (\text{since } T_1 = T_2)$$

$$Q_{\text{in}} = W_{b,\text{out}}$$



Thus,

$$W_{b,\text{out}} = Q_{\text{in}} = \mathbf{1500\ kJ}$$

Using the boundary work relation for the isothermal process of an ideal gas gives

$$W_{b,\text{out}} = m\int_1^2 P d\upsilon = mRT\int_1^2 \frac{d\upsilon}{\upsilon} = mRT\ln\frac{\upsilon_2}{\upsilon_1} = mRT\ln\frac{P_1}{P_2}$$

Solving for the mass of the system,

$$m = \frac{W_{b,\text{out}}}{RT\ln\dfrac{P_1}{P_2}} = \frac{1500\,\text{kJ}}{(0.2081\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(373\,\text{K})\ln\dfrac{200\,\text{kPa}}{50\,\text{kPa}}} = \mathbf{13.94\ kg}$$

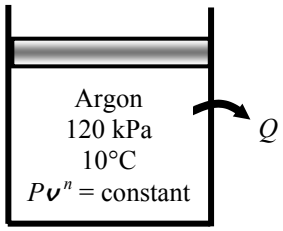
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4-73 Argon is compressed in a polytropic process. The work done and the heat transfer are to be determined.

Assumptions **1** Argon is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 151 K and 4.86 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The properties of argon are $R = 0.2081$kJ/kg·K and $c_v = 0.3122$ kJ/kg·K (Table A-2*a*).

Analysis We take argon as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

Using the boundary work relation for the polytropic process of an ideal gas gives

$$w_{b,\text{out}} = \frac{RT_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] = \frac{(0.2081\,\text{kJ/kg}\cdot\text{K})(283\,\text{K})}{1\text{-}1.2}\left[\left(\frac{800}{120}\right)^{0.2/1.2} - 1\right] = -109.5\,\text{kJ/kg}$$

Thus,

$$w_{b,\text{in}} = \mathbf{109.5\ kJ/kg}$$

The temperature at the final state is

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n\text{-}1)/n} = (283\,\text{K})\left(\frac{800\,\text{kPa}}{120\,\text{kPa}}\right)^{0.2/1.2} = 388.2\,\text{K}$$

From the energy balance equation,

$$q_{\text{in}} = w_{b,\text{out}} + c_v(T_2 - T_1) = -109.5\,\text{kJ/kg} + (0.3122\,\text{kJ/kg}\cdot\text{K})(388.2 - 283)\text{K} = -76.6\,\text{kJ/kg}$$

Thus,

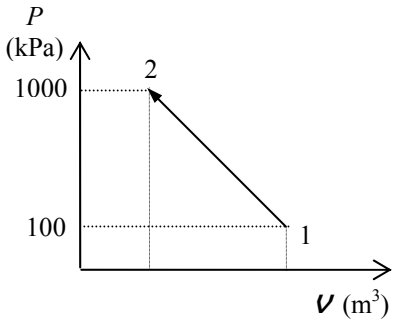
$$q_{\text{out}} = \mathbf{76.6\ kJ/kg}$$

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4-74 Carbon dioxide contained in a spring-loaded piston-cylinder device is heated. The work done and the heat transfer are to be determined.

Assumptions **1** CO_2 is an ideal gas since it is at a high temperature relative to its critical temperature of 304.2 K. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The properties of CO_2 are $R = 0.1889$ kJ/kg·K and $c_v = 0.657$ kJ/kg·K (Table A-2*a*).



Analysis We take CO_2 as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

The initial and final specific volumes are

$$v_1 = \frac{mRT_1}{P_1} = \frac{(1\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\,\text{K})}{100\,\text{kPa}} = 0.5629\ \text{m}^3$$

$$v_2 = \frac{mRT_2}{P_2} = \frac{(1\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(573\,\text{K})}{1000\,\text{kPa}} = 0.1082\ \text{m}^3$$

Pressure changes linearly with volume and the work done is equal to the area under the process line 1-2:

$$\begin{aligned} W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2}(v_2 - v_1) \\ &= \frac{(100 + 1000)\text{kPa}}{2}(0.1082 - 0.5629)\text{m}^3\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) \\ &= -250.1\,\text{kJ} \end{aligned}$$

Thus,

$$W_{b,\text{in}} = \mathbf{250.1\,kJ}$$

Using the energy balance equation,

$$Q_{\text{in}} = W_{b,\text{out}} + mc_v(T_2 - T_1) = -250.1\,\text{kJ} + (1\,\text{kg})(0.657\,\text{kJ/kg}\cdot\text{K})(300 - 25)\text{K} = -69.4\,\text{kJ}$$

Thus,

$$Q_{\text{out}} = \mathbf{69.4\,kJ}$$

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4-75 A piston-cylinder device contains air. A paddle wheel supplies a given amount of work to the air. The heat transfer is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 132.5 K and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for air.

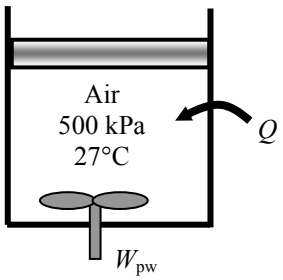
Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} - W_{b,\text{out}} + Q_{\text{in}} = \Delta U = mc_{\upsilon}(T_2 - T_1)$$

$$W_{\text{pw,in}} - W_{b,\text{out}} + Q_{\text{in}} = 0 \quad (\text{since } T_1 = T_2)$$

$$Q_{\text{in}} = W_{b,\text{out}} - W_{\text{pw,in}}$$



Using the boundary work relation on a unit mass basis for the isothermal process of an ideal gas gives

$$w_{b,\text{out}} = RT \ln \frac{\upsilon_2}{\upsilon_1} = RT \ln 3 = (0.287 \text{ kJ/kg} \cdot \text{K})(300 \text{ K}) \ln 3 = 94.6 \text{ kJ/kg}$$

Substituting into the energy balance equation (expressed on a unit mass basis) gives

$$q_{\text{in}} = w_{b,\text{out}} - w_{\text{pw,in}} = 94.6 - 50 = \mathbf{44.6 \text{ kJ/kg}}$$

Discussion Note that the energy content of the system remains constant in this case, and thus the total energy transfer output via boundary work must equal the total energy input via shaft work and heat transfer.

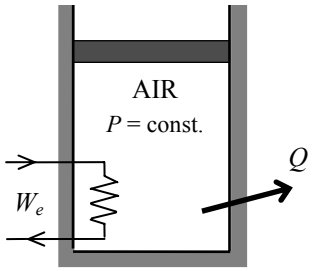
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4-76 A cylinder is initially filled with air at a specified state. Air is heated electrically at constant pressure, and some heat is lost in the process. The amount of electrical energy supplied is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** Air is an ideal gas with variable specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The initial and final enthalpies of air are (Table A-17)

$$h_1 = h_{@\,298\text{ K}} = 298.18\text{ kJ/kg}$$
$$h_2 = h_{@\,350\text{ K}} = 350.49\text{ kJ/kg}$$



Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} - W_{\text{b,out}} = \Delta U \longrightarrow W_{\text{e,in}} = m(h_2 - h_1) + Q_{\text{out}}$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting,

$$W_{\text{e,in}} = (15\text{ kg})(350.49 - 298.18)\text{kJ/kg} + (60\text{ kJ}) = 845\text{ kJ}$$

or,

$$W_{e,in} = (845\text{kJ})\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

Alternative solution The specific heat of air at the average temperature of $T_{\text{avg}} = (25+ 77)/2 = 51°\text{C} = 324\text{ K}$ is, from Table A-2b, $c_{p,\text{avg}} = 1.0065\text{ kJ/kg.°C}$. Substituting,

$$W_{\text{e,in}} = mc_p(T_2 - T_1) + Q_{\text{out}} = (15\text{ kg})(1.0065\text{ kJ/kg.°C})(77 - 25)°\text{C} + 60\text{ kJ} = 845\text{ kJ}$$

or,

$$W_{\text{e,in}} = (845\text{ kJ})\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

Discussion Note that for small temperature differences, both approaches give the same result.

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4-77 A cylinder initially contains nitrogen gas at a specified state. The gas is compressed polytropically until the volume is reduced by one-half. The work done and the heat transfer are to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The N_2 is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

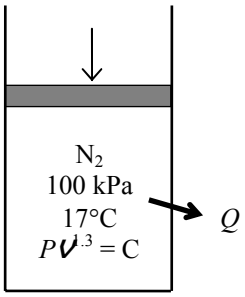
Properties The gas constant of N_2 is $R = 0.2968$ kPa.m^3/kg.K (Table A-1). The c_v value of N_2 at the anticipated average temperature of 350 K is 0.744 kJ/kg.K (Table A-2b).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$W_{\text{b,in}} - Q_{\text{out}} = mc_v(T_2 - T_1)$$



The final pressure and temperature of nitrogen are

$$P_2V_2^{1.3} = P_1V_1^{1.3} \longrightarrow P_2 = \left(\frac{V_1}{V_2}\right)^{1.3} P_1 = 2^{1.3}(100 \text{ kPa}) = 246.2 \text{ kPa}$$

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1}T_1 = \frac{246.2 \text{ kPa}}{100 \text{ kPa}} \times 0.5 \times (290 \text{ K}) = 357.0 \text{ K}$$

Then the boundary work for this polytropic process can be determined from

$$W_{\text{b,in}} = -\int_1^2 P dV = -\frac{P_2V_2 - P_1V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(1.5 \text{ kg})(0.2968 \text{ kJ/kg} \cdot \text{K})(357.0 - 290)\text{K}}{1 - 1.3} = \mathbf{99.5\ kJ}$$

Substituting into the energy balance gives

$$Q_{\text{out}} = W_{\text{b,in}} - mc_v(T_2 - T_1)$$

$$= 99.5 \text{ kJ} - (1.5 \text{ kg})(0.744 \text{ kJ/kg.K})(357.0 - 290)\text{K}$$

$$= \mathbf{24.7\ kJ}$$

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4-78 Problem 4-77 is reconsidered. The process is to be plotted on a *P-V* diagram, and the effect of the polytropic exponent n on the boundary work and heat transfer as the polytropic exponent varies from 1.1 to 1.6 is to be investigated. The boundary work and the heat transfer are to be plotted versus the polytropic exponent.

Analysis The problem is solved using EES, and the solution is given below.

```
Procedure      Work(P[2],V[2],P[1],V[1],n:W_in)
If n=1 then
W_in=-P[1]*V[1]*ln(V[2]/V[1])
Else
W_in=-(P[2]*V[2]-P[1]*V[1])/(1-n)
endif
End

"Input Data"

Vratio=0.5 "V[2]/V[1] = Vratio"
"n=1.3" "Polytropic exponent"
P[1] = 100 [kPa]
T[1] = (17+273) [K]
m=1.5 [kg]
MM=molarmass(nitrogen)
R_u=8.314 [kJ/kmol-K]
R=R_u/MM

V[1]=m*R*T[1]/P[1]

"Process equations"
V[2]=Vratio*V[1]
P[2]*V[2]/T[2]=P[1]*V[1]/T[1]"The combined ideal gas law for
states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P[2]*V[2]^n=P[1]*V[1]^n

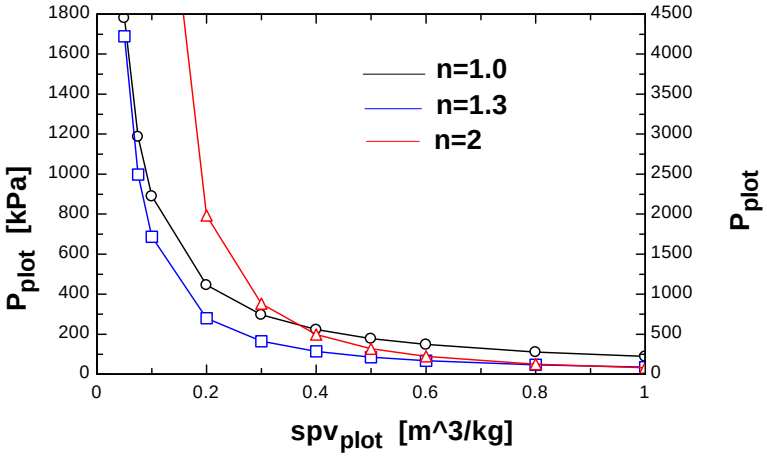
"Conservation of Energy for the closed system:"
"E_in - E_out = DeltaE,  we neglect Delta KE and Delta PE for the system, the nitrogen."

Q_out= W_in-m*(u[2]-u[1])
u[1]=intenergy(N2, T=T[1])  "internal energy for nitrogen as an ideal gas, kJ/kg"
u[2]=intenergy(N2, T=T[2])
Call Work(P[2],V[2],P[1],V[1],n:W_in)

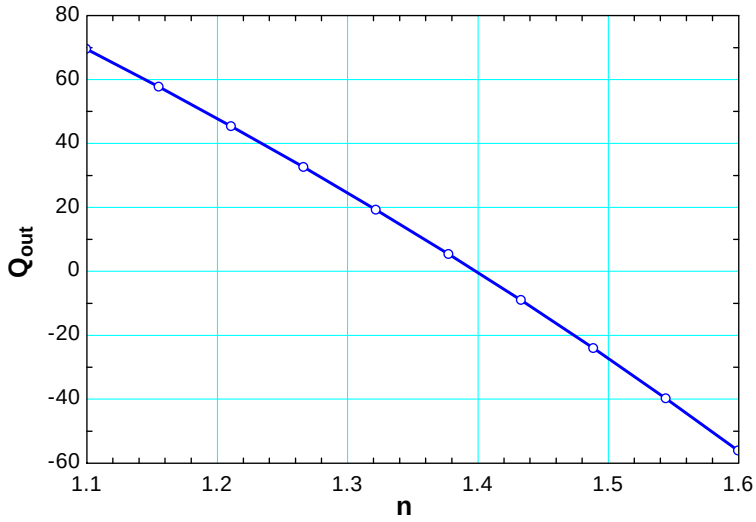
"The following is required for the P-v plots"
{P_plot*spv_plot/T_plot=P[1]*V[1]/m/T[1]
"The combined ideal gas law for states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P_plot*spv_plot^n=P[1]*(V[1]/m)^n}
{spV_plot=R*T_plot/P_plot"[m^3]"}
```

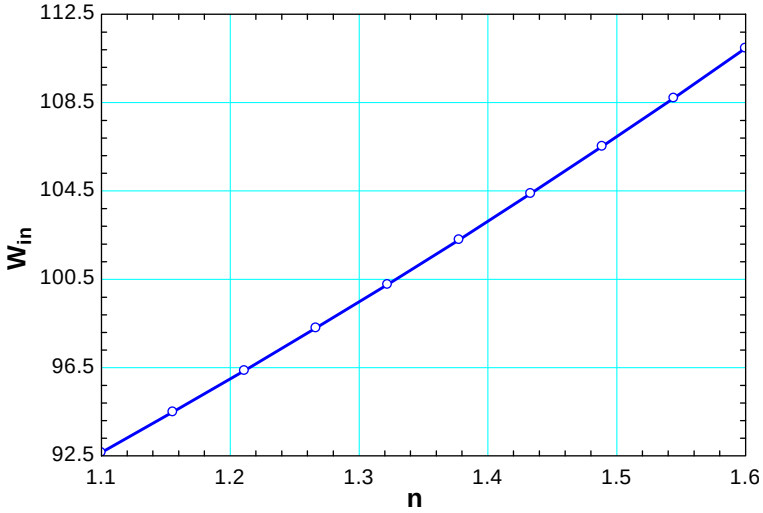
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Pressure vs. specific volume as function of polytropic exponent



n	Q_{out} kJ]	W_{in} [kJ]
1.1	69.41	92.66
1.156	57.59	94.49
1.211	45.29	96.37
1.267	32.49	98.29
1.322	19.17	100.3
1.378	5.299	102.3
1.433	-9.141	104.4
1.489	-24.18	106.5
1.544	-39.87	108.7
1.6	-56.23	111





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4-79 It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 6500 kJ/h. The power rating of the heater is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The temperature of the room is said to remain constant during this process.

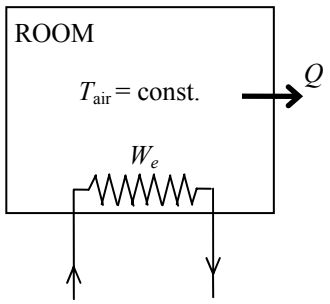
Analysis We take the room as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this system reduces to

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = 0 \longrightarrow W_{\text{e,in}} = Q_{\text{out}}$$

since $\Delta U = mc_v \Delta T = 0$ for isothermal processes of ideal gases. Thus,

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} = (6500 \text{ kJ/h})\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{1.81\ kW}$$

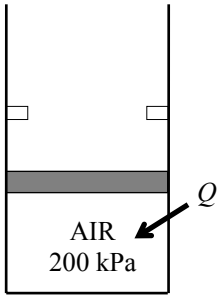


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4-80 A cylinder equipped with a set of stops on the top is initially filled with air at a specified state. Heat is transferred to the air until the piston hits the stops, and then the pressure doubles. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a *P-v* diagram.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The thermal energy stored in the cylinder itself is negligible.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1).



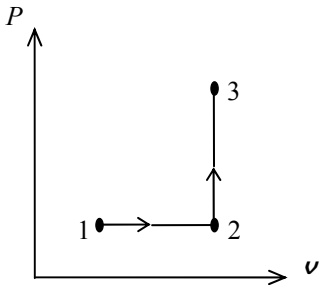
Analysis We take the air in the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_3 - u_1)$$

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}}$$

The initial and the final volumes and the final temperature of air are determined from



$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2\times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}}\times 2\times(300\text{ K}) = 1200\text{ K}$$

No work is done during process 2-3 since $V_2 = V_3$. The pressure remains constant during process 1-2 and the work done during this process is

$$W_b = \int_1^2 P\,dV = P_2(V_2 - V_1)$$

$$= (200\text{ kPa})(2.58 - 1.29)\text{ m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{258\text{ kJ}}$$

The initial and final internal energies of air are (Table A-17)

$$u_1 = u_{@300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_3 = u_{@1200\text{ K}} = 933.33\text{ kJ/kg}$$

Substituting,

$$Q_{\text{in}} = (3\text{ kg})(933.33 - 214.07)\text{kJ/kg} + 258\text{ kJ} = \mathbf{2416\text{ kJ}}$$

Alternative solution The specific heat of air at the average temperature of $T_{\text{avg}} = (300 + 1200)/2 = 750$ K is, from Table A-2b, $c_{v\text{avg}} = 0.800$ kJ/kg.K. Substituting

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}} \cong mc_v(T_3 - T_1) + W_{\text{b,out}}$$

$$= (3\text{ kg})(0.800\text{ kJ/kg.K})(1200 - 300)\text{ K} + 258\text{ kJ} = \mathbf{2418\text{ kJ}}$$

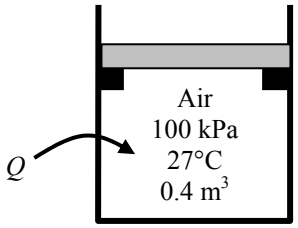
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4-81 Air at a specified state contained in a piston-cylinder device with a set of stops is heated until a final temperature. The amount of heat transfer is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature relative to its critical temperature of 304.2 K. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The properties of air are $R = 0.287$ kJ/kg·K and $c_v = 0.718$ kJ/kg·K (Table A-2*a*).

Analysis We take air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

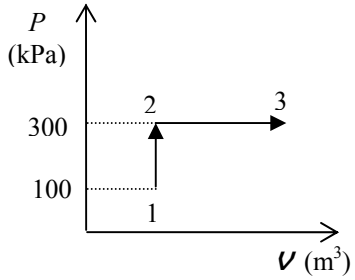


$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_{\boldsymbol{v}}(T_2 - T_1)$$

The volume will be constant until the pressure is 300 kPa:

$$T_2 = T_1 \frac{P_2}{P_1} = (300\text{ K})\frac{300\text{ kPa}}{100\text{ kPa}} = 900\text{ K}$$



The mass of the air is

$$m = \frac{P_1\boldsymbol{V}_1}{RT_1} = \frac{(100\text{ kPa})(0.4\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 0.4646\text{ kg}$$

The boundary work done during process 2-3 is

$$\begin{aligned} W_{b,\text{out}} &= P_2(\boldsymbol{V}_3 - \boldsymbol{V}_2) = mR(T_3 - T_2) \\ &= (0.4646\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(1200 - 900)\text{K} \\ &= 40\text{ kJ} \end{aligned}$$

Substituting these values into energy balance equation,

$$Q_{\text{in}} = W_{b,\text{out}} + mc_{\boldsymbol{v}}(T_3 - T_1) = 40\text{ kJ} + (0.4646\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(1200 - 300)\text{K} = \mathbf{340\text{ kJ}}$$

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4-82 Air at a specified state contained in a piston-cylinder device undergoes an isothermal and constant volume process until a final temperature. The process is to be sketched on the *P*-$\mathcal{V}$ diagram and the amount of heat transfer is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature relative to its critical temperature of 304.2 K. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

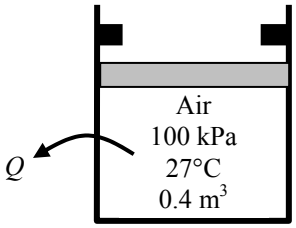
Properties The properties of air are $R = 0.287$ kJ/kg·K and $c_v = 0.718$ kJ/kg·K (Table A-2*a*).

Analysis (a) The processes 1-2 (isothermal) and 2-3 (constant-volume) are sketched on the P-$\mathcal{V}$ diagram as shown.

(b) We take air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system fort he process 1-3 can be expressed as

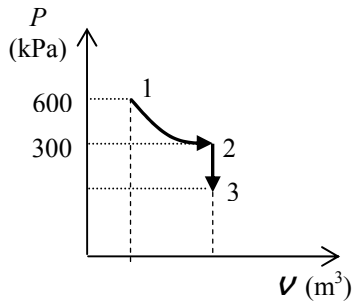
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-W_{b,\text{out},1-2} + Q_{\text{in}} = \Delta U = mc_{\mathcal{V}}(T_3 - T_1)$$



The mass of the air is

$$m = \frac{P_1\mathcal{V}_1}{RT_1} = \frac{(600\text{ kPa})(0.8\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(1200\text{ K})} = 1.394\text{ kg}$$



The work during process 1-2 is determined from boundary work relation for an isothermal process to be

$$W_{b,out,1-2} = mRT_1 \ln\frac{\mathcal{V}_2}{\mathcal{V}_1} = mRT_1 \ln\frac{P_1}{P_2}$$

$$= (1.394\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(1200\text{ K})\ln\frac{600\text{ kPa}}{300\text{ kPa}}$$

$$= 332.8\text{ kJ}$$

since $\dfrac{\mathcal{V}_2}{\mathcal{V}_1} = \dfrac{P_1}{P_2}$ for an isothermal process.

Substituting these values into energy balance equation,

$$Q_{\text{in}} = W_{b,\text{out},1\text{-}2} + mc_{\mathcal{V}}(T_3 - T_1)$$

$$= 332.8\text{ kJ} + (1.394\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(300 - 1200)\text{K}$$

$$= \mathbf{-568\text{ kJ}}$$

Thus,

$$Q_{\text{out}} = \mathbf{568\text{ kJ}}$$

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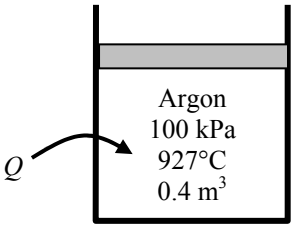
4-83 Argon at a specified state contained in a piston-cylinder device with a set of stops is compressed at constant temperature until a final volume. The process is to be sketched on the *P*-$\boldsymbol{V}$ diagram and the amount of heat transfer is to be determined.

Assumptions **1** Argon is an ideal gas **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The properties of argon are $R = 0.2081$ kJ/kg·K and $c_v = 0.3122$ kJ/kg·K (Table A-2*a*).

Analysis (a) The processes 1-2 (isothermal) and 2-3 (isobaric) are sketched on the P-$\boldsymbol{V}$ diagram as shown.

(b) We take argon as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system fort he process 1-3 can be expressed as

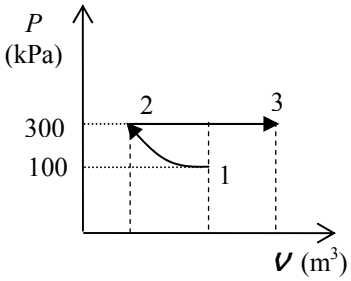


$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,\text{in},1-2} + Q_{\text{in}} - W_{b,\text{out},2\text{-}3} = \Delta U = mc_{\boldsymbol{v}}(T_3 - T_1)$$

The mass of the air is

$$m = \frac{P_1\boldsymbol{V}_1}{RT_1} = \frac{(100\text{ kPa})(0.4\text{ m}^3)}{(0.2081\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 0.6407\text{ kg}$$



The pressure at state 2 is (1-2, isothermal process)

$$P_2 = P_1\frac{\boldsymbol{V}_1}{\boldsymbol{V}_2} = (100\text{ kPa})\frac{0.4\text{ m}^3}{0.2\text{ m}^3} = 200\text{ kPa}$$

The temperature at state 3 is (2-3, isobaric process)

$$T_3 = T_2\frac{\boldsymbol{V}_3}{\boldsymbol{V}_2} = (300\text{ K})\frac{0.6\text{ m}^3}{0.2\text{ m}^3} = 900\text{ K}$$

The compression work during process 1-2 is determined from boundary work relation for an isothermal process to be

$$W_{b,out,1-2} = mRT_1\ln\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} = (0.6407\text{ kg})(0.2081\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})\ln\frac{0.2\text{ m}^3}{0.4\text{ m}^3} = -27.7\text{ kJ}$$

Thus, $W_{b,in,1-2} = 27.7$ kJ

The expansion work during process 2-3 is determined from boundary work relation for a constant-pressure process to be

$$W_{b,\text{out},2\text{-}3} = P_2(\boldsymbol{V}_3 - \boldsymbol{V}_2) = mR(T_3 - T_2) = (0.6407\text{ kg})(0.2081\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(900 - 300)\text{K} = 80.0\text{ kJ}$$

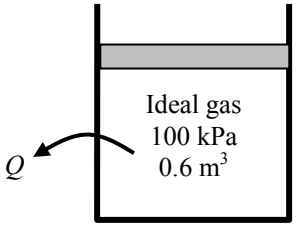
Substituting these values into energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= W_{b,\text{out},2\text{-}3} - W_{b,\text{in},1-2} + mc_{\boldsymbol{v}}(T_3 - T_1) \\ &= 80\text{ kJ} - 27.7\text{ kJ} + (0.6407\text{ kg})(0.3122\text{ kJ/kg}\cdot\text{K})(900 - 300)\text{K} \\ &= \mathbf{172.2\text{ kJ}} \end{aligned}$$

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4-84 An ideal gas at a specified state contained in a piston-cylinder device undergoes an isothermal process during which heat is lost from the gas. The final pressure and volume are to be determined.

***Assumptions* 1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.



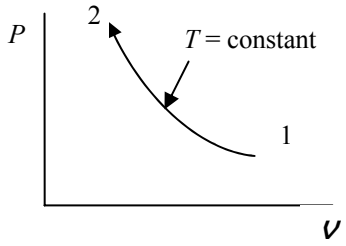
Properties The properties of air are $R = 0.287$ kJ/kg·K and $c_v = 0.718$ kJ/kg·K (Table A-2*a*).

Analysis We take the ideal gas as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system fort he process 1-3 can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} - W_{b,\text{out}} = \Delta U = mc_v(T_3 - T_1) = 0 \text{ since } T_2 = T_1$$

$$-Q_{\text{out}} = W_{b,\text{out}}$$



The boundary work for an isothermal process is determined from

$$W_{b,\text{out}} = \int_1^2 P\,dV = \int_1^2 \frac{mRT}{V}\,dV = mRT\ln\frac{V_2}{V_1} = P_1V_1\ln\frac{V_2}{V_1}$$

Thus,

$$\frac{V_2}{V_1} = \exp\left(\frac{W_{b,out}}{P_1V_1}\right) = \exp\left(\frac{-Q_{\text{out}}}{P_1V_1}\right) = \exp\left(\frac{-60\text{ kJ}}{(100\text{ kPa})(0.6\text{ m}^3)}\right) = \exp(-1) = 0.3679$$

and

$$V_2 = 0.3679V_1 = 0.3679(0.6\text{ m}^3) = \mathbf{0.221\,m^3}$$

The final pressure is found from the combined ideal gas law:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \longrightarrow P_2 = P_1\frac{V_1}{V_2} = (100\text{ kPa})\frac{1}{0.3679} = \mathbf{272\,kPa}$$

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Closed System Energy Analysis: Solids and Liquids

4-85 An iron block is heated. The internal energy and enthalpy changes are to be determined for a given temperature change.

Assumptions Iron is an incompressible substance with a constant specific heat.

Properties The specific heat of iron is 0.45 kJ/kg·K (Table A-3*b*).

Analysis The internal energy and enthalpy changes are equal for a solid. Then,

$$\Delta H = \Delta U = mc\Delta T = (1\,\text{kg})(0.45\,\text{kJ/kg}\cdot\text{K})(75-25)\text{K} = \mathbf{22.5\ kJ}$$

4-86E Liquid water experiences a process from one state to another. The internal energy and enthalpy changes are to be determined under different assumptions.

Analysis (a) Using the properties from compressed liquid tables

$$u_1 \cong u_{f@50°\text{F}} = 18.07\ \text{Btu/lbm} \quad \text{(Table A - 4E)}$$

$$h_1 = h_{f@50°\text{F}} + v_f (P - P_{\text{sat}@T})$$
$$= 18.07\ \text{Btu/lbm} + (0.01602\ \text{ft}^3/\text{lbm})(50-0.17812)\ \text{psia}\left(\frac{1\,\text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) = 18.21\ \text{Btu/lbm}$$

$$\left.\begin{matrix} P_2 = 2000\ \text{psia} \\ T_2 = 100°\text{F} \end{matrix}\right\} \begin{matrix} u_2 = 67.36\ \text{Btu/lbm} \\ h_2 = 73.30\ \text{Btu/lbm} \end{matrix} \quad \text{(Table A - 7E)}$$

$$\Delta u = u_2 - u_1 = 67.36 - 18.07 = \mathbf{49.29\ Btu/lbm}$$
$$\Delta h = h_2 - h_1 = 73.30 - 18.21 = \mathbf{55.08\ Btu/lbm}$$

(b) Using incompressible substance approximation and property tables (Table A-4E),

$$u_1 \cong u_{f@50°\text{F}} = 18.07\ \text{Btu/lbm}$$
$$h_1 \cong h_{f@50°\text{F}} = 18.07\ \text{Btu/lbm}$$
$$u_2 \cong u_{f@100°\text{F}} = 68.03\ \text{Btu/lbm}$$
$$h_2 \cong h_{f@100°\text{F}} = 68.03\ \text{Btu/lbm}$$

$$\Delta u = u_2 - u_1 = 68.03 - 18.07 = \mathbf{49.96\ Btu/lbm}$$
$$\Delta h = h_2 - h_1 = 68.03 - 18.07 = \mathbf{49.96\ Btu/lbm}$$

(c) Using specific heats and taking the specific heat of water to be 1.00 Btu/lbm·R (Table A-3E*a*),

$$\Delta h = \Delta u = c\Delta T = (1.00\,\text{Btu/lbm}\cdot\text{R})(100-50)\text{R} = \mathbf{50\ Btu/lbm}$$

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4-87E A person shakes a canned of drink in a iced water to cool it. The mass of the ice that will melt by the time the canned drink is cooled to a specified temperature is to be determined.

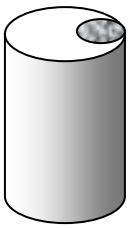
Assumptions **1** The thermal properties of the drink are constant, and are taken to be the same as those of water. **2** The effect of agitation on the amount of ice melting is negligible. **3** The thermal energy capacity of the can itself is negligible, and thus it does not need to be considered in the analysis.

Properties The density and specific heat of water at the average temperature of (75+45)/2 = 60°F are ρ = 62.3 lbm/ft^3, and c_p = 1.0 Btu/lbm.°F (Table A-3E). The heat of fusion of water is 143.5 Btu/lbm.

Analysis We take a canned drink as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{canned drink}} = m(u_2 - u_1) \longrightarrow Q_{\text{out}} = mc(T_1 - T_2)$$



Noting that 1 gal = 128 oz and 1 ft^3 = 7.48 gal = 957.5 oz, the total amount of heat transfer from a ball is

$$m = \rho V = (62.3\ \text{lbm/ft}^3)(12\ \text{oz/can})\left(\frac{1\ \text{ft}^3}{7.48\ \text{gal}}\right)\left(\frac{1\ \text{gal}}{128\ \text{fluid oz}}\right) = 0.781\ \text{lbm/can}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.781\ \text{lbm/can})(1.0\ \text{Btu/lbm.°F})(75 - 45)\text{°F} = 23.4\ \text{Btu/can}$$

Noting that the heat of fusion of water is 143.5 Btu/lbm, the amount of ice that will melt to cool the drink is

$$m_{\text{ice}} = \frac{Q_{\text{out}}}{h_{if}} = \frac{23.4\ \text{Btu/can}}{143.5\ \text{Btu/lbm}} = \mathbf{0.163\ lbm}\ \ \text{(per can of drink)}$$

since heat transfer to the ice must be equal to heat transfer from the can.

Discussion The actual amount of ice melted will be greater since agitation will also cause some ice to melt.

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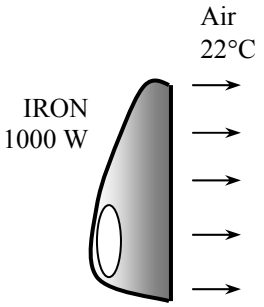
4-88 An iron whose base plate is made of an aluminum alloy is turned on. The minimum time for the plate to reach a specified temperature is to be determined.

Assumptions **1** It is given that 85 percent of the heat generated in the resistance wires is transferred to the plate. **2** The thermal properties of the plate are constant. **3** Heat loss from the plate during heating is disregarded since the minimum heating time is to be determined. **4** There are no changes in kinetic and potential energies. **5** The plate is at a uniform temperature at the end of the process.

Properties The density and specific heat of the aluminum alloy plate are given to be $\rho = 2770$ kg/m^3 and $c_p = 875$ kJ/kg.°C.

Analysis The mass of the iron's base plate is

$$m = \rho \boldsymbol{V} = \rho LA = (2770 \text{ kg/m}^3)(0.005 \text{ m})(0.03 \text{ m}^2) = 0.4155 \text{ kg}$$



Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{\text{in}} = 0.90 \times 1000 \text{ W} = 900 \text{ W}$$

We take plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) \longrightarrow \dot{Q}_{\text{in}} \Delta t = mc(T_2 - T_1)$$

Solving for Δt and substituting,

$$\Delta t = \frac{mc\Delta T_{\text{plate}}}{\dot{Q}_{\text{in}}} = \frac{(0.4155 \text{ kg})(875 \text{ J/kg.°C})(200 - 22)°\text{C}}{900 \text{ J/s}} = \mathbf{71.9 \text{ s}}$$

which is the time required for the plate temperature to reach the specified temperature.

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4-89 Stainless steel ball bearings leaving the oven at a specified uniform temperature at a specified rate are exposed to air and are cooled before they are dropped into the water for quenching. The rate of heat transfer from the ball bearing to the air is to be determined.

Assumptions **1** The thermal properties of the bearing balls are constant. **2** The kinetic and potential energy changes of the balls are negligible. **3** The balls are at a uniform temperature at the end of the process

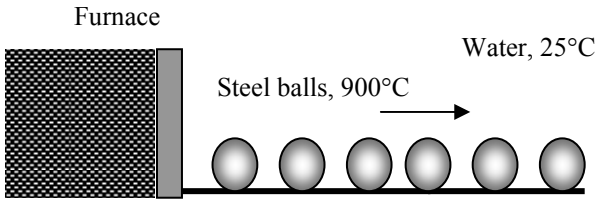
Properties The density and specific heat of the ball bearings are given to be $\rho = 8085$ kg/m^3 and $c_p = 0.480$ kJ/kg.°C.

Analysis We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8085\ \text{kg/m}^3)\frac{\pi(0.012\ \text{m})^3}{6} = 0.007315\ \text{kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.007315\ \text{kg})(0.480\ \text{kJ/kg.°C})(900 - 850)°\text{C} = 0.1756\ \text{kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{out (per ball)}} = (800\ \text{balls/min}) \times (0.1756\ \text{kJ/ball}) = \mathbf{140.5\ kJ/min} = \mathbf{2.34\ kW}$$

Therefore, heat is lost to the air at a rate of 2.34 kW.

4-90 Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air is to be determined.

Assumptions **1** The thermal properties of the balls are constant. **2** There are no changes in kinetic and potential energies. **3** The balls are at a uniform temperature at the end of the process

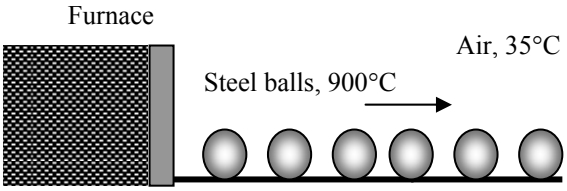
Properties The density and specific heat of the balls are given to be $\rho = 7833$ kg/m^3 and $c_p = 0.465$ kJ/kg.°C.

Analysis We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$



(*b*) The amount of heat transfer from a single ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (7833\ \text{kg/m}^3)\frac{\pi(0.008\ \text{m})^3}{6} = 0.00210\ \text{kg}$$

$$Q_{\text{out}} = mc_p(T_1 - T_2) = (0.0021\ \text{kg})(0.465\ \text{kJ/kg.°C})(900 - 100)°\text{C} = 0.781\ \text{kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

$$\dot{Q}_{\text{out}} = \dot{n}_{\text{ball}} Q_{\text{out}} = (2500\ \text{balls/h}) \times (0.781\ \text{kJ/ball}) = 1{,}953\ \text{kJ/h} = \mathbf{542\ W}$$

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4-91 An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked is to be determined.

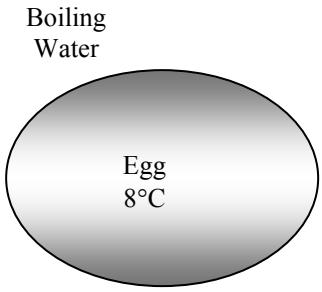
Assumptions **1** The egg is spherical in shape with a radius of $r_0 = 2.75$ cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies.

Properties The density and specific heat of the egg are given to be $\rho = 1020$ kg/m^3 and $c_p = 3.32$ kJ/kg.°C.

Analysis We take the egg as the system. This is a closes system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{egg}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020 \text{ kg/m}^3)\frac{\pi (0.055 \text{ m})^3}{6} = 0.0889 \text{ kg}$$

$$Q_{\text{in}} = mc_p(T_2 - T_1) = (0.0889 \text{ kg})(3.32 \text{ kJ/kg.°C})(80 - 8)°\text{C} = \mathbf{21.2\ kJ}$$

4-92E Large brass plates are heated in an oven at a rate of 300/min. The rate of heat transfer to the plates in the oven is to be determined.

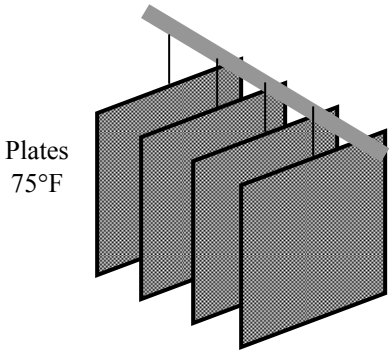
Assumptions **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the brass are given to be $\rho = 532.5$ lbm/ft^3 and $c_p = 0.091$ Btu/lbm.°F.

Analysis We take the plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



The mass of each plate and the amount of heat transfer to each plate is

$$m = \rho V = \rho LA = (532.5 \text{ lbm/ft}^3)[(1.2/12 \text{ ft})(2 \text{ ft})(2 \text{ ft})] = 213 \text{ lbm}$$

$$Q_{\text{in}} = mc(T_2 - T_1) = (213 \text{ lbm/plate})(0.091 \text{ Btu/lbm.°F})(1000 - 75)°\text{F} = 17{,}930 \text{ Btu/plate}$$

Then the total rate of heat transfer to the plates becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{plate}} Q_{\text{in, per plate}} = (300 \text{ plates/min}) \times (17{,}930 \text{ Btu/plate}) = \mathbf{5{,}379{,}000\ Btu/min = 89{,}650\ Btu/s}$$

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4-93 Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven is to be determined.

Assumptions **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the steel rods are given to be $\rho = 7833$ kg/m^3 and $c_p = 0.465$ kJ/kg.°C.

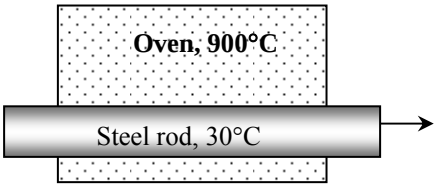
Analysis Noting that the rods enter the oven at a velocity of 2 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho \boldsymbol{V} = \rho LA = \rho L(\pi D^2 / 4) = (7833 \text{ kg/m}^3)(2 \text{ m})[\pi(0.08 \text{ m})^2 / 4] = 78.75 \text{ kg}$$

We take the 2-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (78.75 \text{ kg})(0.465 \text{ kJ/kg.°C})(700 - 30)°\text{C} = 24{,}530 \text{ kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}} / \Delta t = (24{,}530 \text{ kJ})/(1 \text{ min}) = 24{,}530 \text{ kJ/min} = \mathbf{409 \text{ kW}}$$

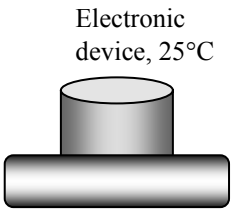
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4-94 An electronic device is on for 5 minutes, and off for several hours. The temperature of the device at the end of the 5-min operating period is to be determined for the cases of operation with and without a heat sink.

Assumptions **1** The device and the heat sink are isothermal. **2** The thermal properties of the device and of the sink are constant. **3** Heat loss from the device during on time is disregarded since the highest possible temperature is to be determined.

Properties The specific heat of the device is given to be c_p = 850 J/kg.°C. The specific heat of aluminum at room temperature of 300 K is 902 J/kg.°C (Table A-3).

Analysis We take the device to be the system. Noting that electrical energy is supplied, the energy balance for this closed system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U_{\text{device}} = m(u_2 - u_1)$$

$$\dot{W}_{\text{e,in}}\Delta t = mc(T_2 - T_1)$$

Substituting, the temperature of the device at the end of the process is determined to be

$$(25\text{ J/s})(5\times 60\text{ s}) = (0.020\text{ kg})(850\text{ J/kg.°C})(T_2 - 25)\text{°C} \;\rightarrow\; T_2 = \mathbf{466°C} \text{ (without the heat sink)}$$

Case 2 When a heat sink is attached, the energy balance can be expressed as

$$W_{\text{e,in}} = \Delta U_{\text{device}} + \Delta U_{\text{heat sink}}$$

$$\dot{W}_{\text{e,in}}\Delta t = mc(T_2 - T_1)_{\text{device}} + mc(T_2 - T_1)_{\text{heat sink}}$$

Substituting, the temperature of the device-heat sink combination is determined to be

$$(25\text{ J/s})(5\times 60\text{ s}) = (0.020\text{ kg})(850\text{ J/kg.°C})(T_2 - 25)\text{°C} + (0.500\text{ kg})(902\text{ J/kg.°C})(T_2 - 25)\text{°C}$$

$$T_2 = \mathbf{41.0°C} \text{ (with heat sink)}$$

Discussion These are the maximum temperatures. In reality, the temperatures will be lower because of the heat losses to the surroundings.

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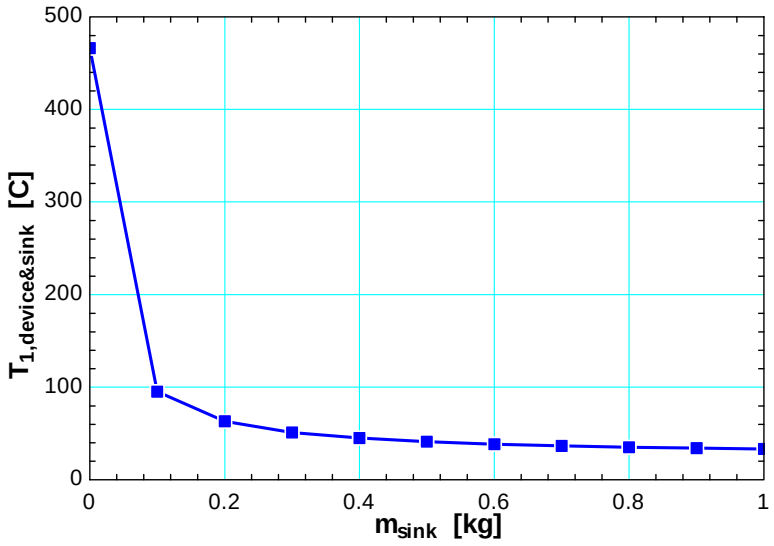
4-95 Problem 4-94 is reconsidered. The effect of the mass of the heat sink on the maximum device temperature as the mass of heat sink varies from 0 kg to 1 kg is to be investigated. The maximum temperature is to be plotted against the mass of heat sink.

Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns:"
"T_1 is the maximum temperature of the device"
Q_dot_out = 25 [W]
m_device=20 [g]
Cp_device=850 [J/kg-C]
A=5 [cm^2]
DELTAt=5 [min]
T_amb=25 [C]
{m_sink=0.5 [kg]}
"Cp_al taken from Table A-3(b) at 300K"
Cp_al=0.902 [kJ/kg-C]
T_2=T_amb

"Solution:"
"The device without the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the system, the device."
E_dot_in - E_dot_out = DELTAE_dot
E_dot_in =0
E_dot_out = Q_dot_out
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot= m_device*convert(g,kg)*Cp_device*(T_2-T_1_device)/(DELTAt*convert(min,s))
"The device with the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the device with the heat sink."
E_dot_in - E_dot_out = DELTAE_dot_combined
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot_combined= (m_device*convert(g,kg)*Cp_device*(T_2-T_1_device&sink)+m_sink*Cp_al*(T_2-
T_1_device&sink)*convert(kJ,J))/(DELTAt*convert(min,s))
```

m_{sink} [kg]	$T_{1,device\&sink}$ [C]
0	466.2
0.1	94.96
0.2	62.99
0.3	51.08
0.4	44.85
0.5	41.03
0.6	38.44
0.7	36.57
0.8	35.15
0.9	34.05
1	33.16



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4-96 The face of a person is slapped. For the specified temperature rise of the affected part, the impact velocity of the hand is to be determined.

Assumptions **1** The hand is brought to a complete stop after the impact. **2** The face takes the blow without significant movement. **3** No heat is transferred from the affected area to the surroundings, and thus the process is adiabatic. **4** No work is done on or by the system. **5** The potential energy change is zero, $\Delta PE = 0$ and $\Delta E = \Delta U + \Delta KE$.

Analysis We analyze this incident in a professional manner without involving any emotions. First, we identify the system, draw a sketch of it, and state our observations about the specifics of the problem. We take the hand and the affected portion of the face as the *system*. This is a *closed system* since it involves a fixed amount of mass (no mass transfer). We observe that the kinetic energy of the hand decreases during the process, as evidenced by a decrease in velocity from initial value to zero, while the internal energy of the affected area increases, as evidenced by an increase in the temperature. There seems to be no significant energy transfer between the system and its surroundings during this process. Under the stated assumptions and observations, the energy balance on the system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U_{\text{affected tissue}} + \Delta \text{KE}_{\text{hand}}$$

$$0 = (mc\Delta T)_{\text{affected tissue}} + [m(0 - V^2)/2]_{\text{hand}}$$

That is, the decrease in the kinetic energy of the hand must be equal to the increase in the internal energy of the affected area. Solving for the velocity and substituting the given quantities, the impact velocity of the hand is determined to be

$$V_{\text{hand}} = \sqrt{\frac{2(mc\Delta T)_{\text{affected tissue}}}{m_{\text{hand}}}}$$

$$V_{\text{hand}} = \sqrt{\frac{2(0.15\,\text{kg})(3.8\,\text{kJ/kg}\cdot{}^\circ\text{C})(1.8{}^\circ\text{C})}{1.2\,\text{kg}}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)}$$

$$= \mathbf{41.4\ m/s}\ (\text{or } 149\ \text{km/h})$$

Discussion Reconstruction of events such as this by making appropriate assumptions are commonly used in forensic engineering.

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Special Topic: Biological Systems

4-97C Metabolism refers to the chemical activity in the cells associated with the burning of foods. The basal metabolic rate is the metabolism rate of a resting person, which is 84 W for an average man.

4-98C The food we eat is not entirely metabolized in the human body. The fraction of metabolizable energy contents are 95.5% for carbohydrates, 77.5% for proteins, and 97.7% for fats. Therefore, the metabolizable energy content of a food is not the same as the energy released when it is burned in a bomb calorimeter.

4-99C Yes. Each body rejects the heat generated during metabolism, and thus serves as a heat source. For an average adult male it ranges from 84 W at rest to over 1000 W during heavy physical activity. Classrooms are designed for a large number of occupants, and thus the total heat dissipated by the occupants must be considered in the design of heating and cooling systems of classrooms.

4-100C 1 kg of natural fat contains almost 8 times the metabolizable energy of 1 kg of natural carbohydrates. Therefore, a person who fills his stomach with carbohydrates will satisfy his hunger without consuming too many calories.

4-101 Six people are fast dancing in a room, and there is a resistance heater in another identical room. The room that will heat up faster is to be determined.

Assumptions **1** The rooms are identical in every other aspect. **2** Half of the heat dissipated by people is in sensible form. **3** The people are of average size.

Properties An average fast dancing person dissipates 600 Cal/h of energy (sensible and latent) (Table 4-2).

Analysis Three couples will dissipate

$$E = (6 \text{ persons})(600 \text{ Cal/h.person})(4.1868 \text{ kJ/Cal}) = 15{,}072 \text{ kJ/h} = 4190 \text{ W}$$

of energy. (About half of this is sensible heat). Therefore, the room with the **people dancing** will warm up much faster than the room with a 2-kW resistance heater.

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4-102 Two men are identical except one jogs for 30 min while the other watches TV. The weight difference between these two people in one month is to be determined.

Assumptions The two people have identical metabolism rates, and are identical in every other aspect.

Properties An average 68-kg person consumes 540 Cal/h while jogging, and 72 Cal/h while watching TV (Table 4-2).

Analysis An 80-kg person who jogs 0.5 h a day will have jogged a total of 15 h a month, and will consume

$$\Delta E_{\text{consumed}} = [(540 - 72)\ \text{Cal/h}](15\ \text{h})\left(\frac{4.1868\ \text{kJ}}{1\ \text{Cal}}\right)\left(\frac{80\ \text{kg}}{68\ \text{kg}}\right) = 34{,}578\ \text{kJ}$$

more calories than the person watching TV. The metabolizable energy content of 1 kg of fat is 33,100 kJ. Therefore, the weight difference between these two people in 1-month will be

$$\Delta m_{\text{fat}} = \frac{\Delta E_{\text{consumed}}}{\text{Energy content of fat}} = \frac{34{,}578\ \text{kJ}}{33{,}100\ \text{kJ/kg}} = \mathbf{1.045\ kg}$$

4-103 A bicycling woman is to meet her entire energy needs by eating 30-g candy bars. The number of candy bars she needs to eat to bicycle for 1-h is to be determined.

Assumptions The woman meets her entire calorie needs from candy bars while bicycling.

Properties An average 68-kg person consumes 639 Cal/h while bicycling, and the energy content of a 20-g candy bar is 105 Cal (Tables 4-1 and 4-2).

Analysis Noting that a 20-g candy bar contains 105 Calories of metabolizable energy, a 30-g candy bar will contain

$$E_{\text{candy}} = (105\ \text{Cal})\left(\frac{30\ \text{g}}{20\ \text{g}}\right) = 157.5\ \text{Cal}$$

of energy. If this woman is to meet her entire energy needs by eating 30-g candy bars, she will need to eat

$$N_{\text{candy}} = \frac{639\text{Cal/h}}{157.5\text{Cal}} \cong \mathbf{4 candy bars/h}$$

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4-104 A 75-kg man eats 1-L of ice cream. The length of time this man needs to jog to burn off these calories is to be determined.

Assumptions The man meets his entire calorie needs from the ice cream while jogging.

Properties An average 68-kg person consumes 540 Cal/h while jogging, and the energy content of a 100-ml of ice cream is 110 Cal (Tables 4-1 and 4-2).

Analysis The rate of energy consumption of a 55-kg person while jogging is

$$\dot{E}_{\text{consumed}} = (540 \text{ Cal/h})\left(\frac{75 \text{ kg}}{68 \text{ kg}}\right) = 596 \text{ Cal/h}$$

Noting that a 100-ml serving of ice cream has 110 Cal of metabolizable energy, a 1-liter box of ice cream will have 1100 Calories. Therefore, it will take

$$\Delta t = \frac{1100 \text{ Cal}}{596 \text{ Cal/h}} = \mathbf{1.85\ h}$$

of jogging to burn off the calories from the ice cream.

4-105 A man with 20-kg of body fat goes on a hunger strike. The number of days this man can survive on the body fat alone is to be determined.

Assumptions **1** The person is an average male who remains in resting position at all times. **2** The man meets his entire calorie needs from the body fat alone.

Properties The metabolizable energy content of fat is 33,100 Cal/kg. An average resting person burns calories at a rate of 72 Cal/h (Table 4-2).

Analysis The metabolizable energy content of 20 kg of body fat is

$$E_{\text{fat}} = (33{,}100 \text{ kJ/kg})(20 \text{ kg}) = 662{,}000 \text{ kJ}$$

The person will consume

$$E_{\text{consumed}} = (72 \text{ Cal/h})(24 \text{ h})\left(\frac{4.1868 \text{ kJ}}{1 \text{ Cal}}\right) = 7235 \text{ kJ/day}$$

Therefore, this person can survive

$$\Delta t = \frac{662{,}000 \text{ kJ}}{7235 \text{ kJ / day}} = \mathbf{91.5\ days}$$

on his body fat alone. This result is not surprising since people are known to survive over 100 days without any food intake.

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4-106 Two 50-kg women are identical except one eats her baked potato with 4 teaspoons of butter while the other eats hers plain every evening. The weight difference between these two woman in one year is to be determined.

Assumptions **1** These two people have identical metabolism rates, and are identical in every other aspect. **2** All the calories from the butter are converted to body fat.

Properties The metabolizable energy content of 1 kg of body fat is 33,100 kJ. The metabolizable energy content of 1 teaspoon of butter is 35 Calories (Table 4-1).

Analysis A person who eats 4 teaspoons of butter a day will consume

$$E_{\text{consumed}} = (35\ \text{Cal/teaspoon})(4\ \text{teaspoons/day})\left(\frac{365\ \text{days}}{1\ \text{year}}\right) = 51{,}100\ \text{Cal/year}$$

Therefore, the woman who eats her potato with butter will gain

$$m_{\text{fat}} = \frac{51{,}100\ \text{Cal}}{33{,}100\ \text{kJ/kg}}\left(\frac{4.1868\ \text{kJ}}{1\ \text{Cal}}\right) = \mathbf{6.5\ kg}$$

of additional body fat that year.

4-107 A woman switches from 1-L of regular cola a day to diet cola and 2 slices of apple pie. It is to be determined if she is now consuming more or less calories.

Properties The metabolizable energy contents are 300 Cal for a slice of apple pie, 87 Cal for a 200-ml regular cola, and 0 for the diet drink (Table 4-3).

Analysis The energy contents of 2 slices of apple pie and 1-L of cola are

$$E_{\text{pie}} = 2 \times (300\ \text{Cal}) = 600\ \text{Cal}$$
$$E_{\text{cola}} = 5 \times (87\ \text{Cal}) = 435\ \text{Cal}$$

Therefore, the woman is now consuming **more calories**.

4-108 A person drinks a 12-oz beer, and then exercises on a treadmill. The time it will take to burn the calories from a 12-oz can of regular and light beer are to be determined.

Assumptions The drinks are completely metabolized by the body.

Properties The metabolizable energy contents of regular and light beer are 150 and 100 Cal, respectively. Exercising on a treadmill burns calories at an average rate of 700 Cal/h (given).

Analysis The exercising time it will take to burn off beer calories is determined directly from

(*a*) Regular beer: $$\Delta t_{\text{regular beer}} = \frac{150\ \text{Cal}}{700\ \text{Cal/h}} = 0.214\ \text{h} = \mathbf{12.9\ min}$$

(*b*) Light beer: $$\Delta t_{\text{light beer}} = \frac{100\ \text{Cal}}{700\ \text{Cal/h}} = 0.143\ \text{h} = \mathbf{8.6\ min}$$

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4-109 A man and a woman have lunch at Burger King, and then shovel snow. The shoveling time it will take to burn off the lunch calories is to be determined for both.

Assumptions The food intake during lunch is completely metabolized by the body.

Properties The metabolizable energy contents of different foods are as given in the problem statement. Shoveling snow burns calories at a rate of 360 Cal/h for the woman and 480 Cal/h for the man (given).

Analysis The total calories consumed during lunch and the time it will take to burn them are determined for both the man and woman as follows:

*Man***:** Lunch calories = 720+400+225 = 1345 Cal.

$$\text{Shoveling time: } \Delta t_{\text{shoveling, man}} = \frac{1345 \text{ Cal}}{480 \text{ Cal/h}} = \mathbf{2.80\ h}$$

*Woman***:** Lunch calories = 330+400+0 = 730 Cal.

$$\text{Shoveling time: } \Delta t_{\text{shoveling, woman}} = \frac{730 \text{ Cal}}{360 \text{ Cal/h}} = \mathbf{2.03\ h}$$

4-110 Two friends have identical metabolic rates and lead identical lives, except they have different lunches. The weight difference between these two friends in a year is to be determined.

***Assumptions* 1** The diet and exercise habits of the people remain the same other than the lunch menus. **2** All the excess calories from the lunch are converted to body fat.

Properties The metabolizable energy content of body fat is 33,100 Cal/kg (text). The metabolizable energy contents of different foods are given in problem statement.

Analysis The person who has the double whopper sandwich consumes 1600 – 800 = 800 Cal more every day. The difference in calories consumed per year becomes

Calorie consumption difference = (800 Cal/day)(365 days/year) = 292,000 Cal/year

Therefore, assuming all the excess calories to be converted to body fat, the weight difference between the two persons after 1 year will be

$$\text{Weight difference} = \frac{\text{Calorie intake difference}}{\text{Enegy content of fat}} = \frac{\Delta E_{\text{intake}}}{e_{\text{fat}}} = \frac{292{,}000 \text{ Cal/yr}}{33{,}100 \text{ kJ/kg}}\left(\frac{4.1868 \text{ kJ}}{1 \text{ Cal}}\right) = \mathbf{36.9\ kg/yr}$$

or about 80 pounds of body fat per year.

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4-111E A person eats dinner at a fast-food restaurant. The time it will take for this person to burn off the dinner calories by climbing stairs is to be determined.

Assumptions The food intake from dinner is completely metabolized by the body.

Properties The metabolizable energy contents are 270 Cal for regular roast beef, 410 Cal for big roast beef, and 150 Cal for the drink. Climbing stairs burns calories at a rate of 400 Cal/h (given).

Analysis The total calories consumed during dinner and the time it will take to burn them by climbing stairs are determined to be

Dinner calories = 270+410+150 = 830 Cal.

Stair climbing time: $\Delta t = \dfrac{830 \text{ Cal}}{400 \text{ Cal/h}} = \mathbf{2.08\ h}$

4-112 Three people have different lunches. The person who consumed the most calories from lunch is to be determined.

Properties The metabolizable energy contents of different foods are 530 Cal for the Big Mac, 640 Cal for the whopper, 350 Cal for french fries, and 5 Cal for each olive (given).

Analysis The total calories consumed by each person during lunch are:

Person 1**:** Lunch calories = 530 Cal

Person 2**:** Lunch calories = **640 Cal**

Person 3**:** Lunch calories = 350+5×50 = 600 Cal

Therefore, the person with the Whopper will consume the most calories.

4-113 A 100-kg man decides to lose 10 kg by exercising without reducing his calorie intake. The number of days it will take for this man to lose 10 kg is to be determined.

Assumptions **1** The diet and exercise habits of the person remain the same other than the new daily exercise program. **2** The entire calorie deficiency is met by burning body fat.

Properties The metabolizable energy content of body fat is 33,100 Cal/kg (text).

Analysis The energy consumed by an average 68-kg adult during fast-swimming, fast dancing, jogging, biking, and relaxing are 860, 600, 540, 639, and 72 Cal/h, respectively (Table 4-2). The daily energy consumption of this 100-kg man is

$$\left[(860+600+540+639 \text{ Cal/h})(1\text{h})+(72 \text{ Cal/h})(23 \text{ h})\right]\left(\frac{100 \text{ kg}}{68 \text{ kg}}\right) = 6316 \text{ Cal}$$

Therefore, this person burns 6316-4000 = 2316 more Calories than he takes in, which corresponds to

$$m_{\text{fat}} = \frac{2316 \text{ Cal}}{33{,}100 \text{ kJ/kg}}\left(\frac{4.1868 \text{ kJ}}{1 \text{ Cal}}\right) = 0.293 \text{ kg}$$

of body fat per day. Thus it will take only

$$\Delta t = \frac{10 \text{ kg}}{0.293 \text{ kg}} = \mathbf{34.1\ days}$$

for this man to lose 10 kg.

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4-114 The effect of supersizing in fast food restaurants on monthly weight gain is to be determined for a given case.

Properties The metabolizable energy content of 1 kg of body fat is 33,100 kJ.

Analysis The increase in the body fat mass due to extra 1000 calories is

$$m_{\text{fat}} = \frac{1000\,\text{Cal/day}}{33{,}100\,\text{kJ/kg}}\left(\frac{4.1868\,\text{kJ}}{1\,\text{Cal}}\right)(30\,\text{days/month}) = \mathbf{3.79\,kg/month}$$

4-115E The range of healthy weight for adults is usually expressed in terms of the *body mass index* (BMI) in SI units as $\text{BMI} = \frac{W(\text{kg})}{H^2(\text{m}^2)}$. This formula is to be converted to English units such that the weight is in pounds and the height in inches.

Analysis Noting that 1 kg = 2.2 lbm and 1 m =39.37 in, the weight in lbm must be divided by 2.2 to convert it to kg, and the height in inches must be divided by 39.37 to convert it to m before inserting them into the formula. Therefore,

$$\text{BMI} = \frac{W(\text{kg})}{H^2(\text{m}^2)} = \frac{W(\text{lbm})/2.2}{H^2(\text{in}^2)/(39.37)^2} = 705\frac{W(\text{lbm})}{H^2(\text{in}^2)}$$

Every person can calculate their own BMI using either SI or English units, and determine if it is in the healthy range.

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4-116 A person changes his/her diet to lose weight. The time it will take for the body mass index (BMI) of the person to drop from 30 to 20 is to be determined.

Assumptions The deficit in the calori intake is made up by burning body fat.

Properties The metabolizable energy contents are 350 Cal for a slice of pizza and 87 Cal for a 200-ml regular cola. The metabolizable energy content of 1 kg of body fat is 33,100 kJ.

Analysis The lunch calories before the diet is

$$E_{\text{old}} = 3 \times e_{\text{pizza}} + 2 \times e_{\text{coke}} = 3 \times (350\ \text{Cal}) + 2 \times (87\ \text{Cal}) = 1224\ \text{Cal}$$

The lunch calories after the diet is

$$E_{\text{old}} = 2 \times e_{\text{pizza}} + 1 \times e_{\text{coke}} = 2 \times (350\ \text{Cal}) + 1 \times (87\ \text{Cal}) = 787\ \text{Cal}$$

The calorie reduction is

$$E_{\text{reduction}} = 1224 - 787 = 437\ \text{Cal}$$

The corresponding reduction in the body fat mass is

$$m_{\text{fat}} = \frac{437\ \text{Cal}}{33{,}100\ \text{kJ/kg}} \left(\frac{4.1868\ \text{kJ}}{1\ \text{Cal}} \right) = 0.05528\ \text{kg}$$

The weight of the person before and after the diet is

$$W_1 = \text{BMI}_1 \times h^2_{\text{pizza}} = 30 \times (1.6\ \text{m})^2 = 76.8\ \text{kg}$$
$$W_2 = \text{BMI}_2 \times h^2_{\text{pizza}} = 20 \times (1.6\ \text{m})^2 = 51.2\ \text{kg}$$

Then it will take

$$\text{Time} = \frac{W_1 - W_2}{m_{\text{fat}}} = \frac{(76.8 - 51.2)\ \text{kg}}{0.05528\ \text{kg/day}} = \mathbf{463\ days}$$

for the BMI of this person to drop to 20.

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Review Problems

4-117 During a phase change, a constant pressure process becomes a constant temperature process. Hence,

$$c_p = \left.\frac{\partial h}{\partial T}\right|_T = \frac{\text{finite}}{0} = \text{infinite}$$

and the specific heat at constant pressure has no meaning. The specific heat at constant volume does have a meaning since

$$c_v = \left.\frac{\partial u}{\partial T}\right|_v = \frac{\text{finite}}{\text{finite}} = \text{finite}$$

4-118 Nitrogen is heated to experience a specified temperature change. The heat transfer is to be determined for two cases.

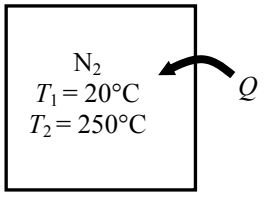
Assumptions **1** Nitrogen is an ideal gas since it is at a high temperature and probably low pressure relative to its critical point values of 126.2 K and 3.39 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for nitrogen.

Properties The specific heats of nitrogen at the average temperature of (20+250)/2=135°C=408 K are c_p = 1.045 kJ/kg·K and c_v = 0.748 kJ/kg·K (Table A-2*b*).

Analysis We take the nitrogen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for a constant-volume process can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = mc_v(T_2 - T_1)$$



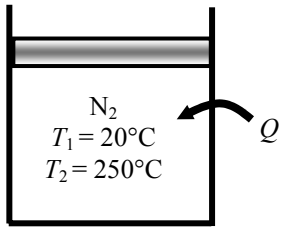
The energy balance during a constant-pressure process (such as in a piston-cylinder device) can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U$$

$$Q_{\text{in}} = W_{b,\text{out}} + \Delta U$$

$$Q_{\text{in}} = \Delta H = mc_p(T_2 - T_1)$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting for both cases,

$$Q_{\text{in, }v=\text{const}} = mc_v(T_2 - T_1) = (10\text{ kg})(0.748\text{ kJ/kg}\cdot\text{K})(250 - 20)\text{K} = \mathbf{1720 kJ}$$

$$Q_{\text{in, }P=\text{const}} = mc_p(T_2 - T_1) = (10\text{ kg})(1.045\text{ kJ/kg}\cdot\text{K})(250 - 20)\text{K} = \mathbf{2404\text{ kJ}}$$

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4-119 A classroom has 40 students, each dissipating 84 W of sensible heat. It is to be determined if it is necessary to turn the heater on in the room to avoid cooling of the room.

Properties Each person is said to be losing sensible heat to the room air at a rate of 84 W.

Analysis We take the room is losing heat to the outdoors at a rate of

$$\dot{Q}_{\text{loss}} = (12{,}000 \text{ kJ/h})\left(\frac{1 \text{ h}}{3600 \text{ s}}\right) = 3.33 \text{ kW}$$

The rate of sensible heat gain from the students is

$$\dot{Q}_{\text{gain}} = (84 \text{ W/student})(40 \text{ students}) = 3360 \text{ W} = 3.36 \text{ kW}$$

which is greater than the rate of heat loss from the room. Therefore, it is **not necessary** to turn the heater.

4-120E An insulated rigid vessel contains air. A paddle wheel supplies work to the air. The work supplied and final temperature are to be determined.

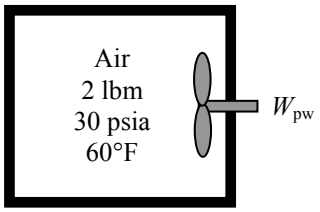
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 238.5 R and 547 psia. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for air.

Properties The specific heats of air at room temperature are $c_p = 0.240$ Btu/lbm·R and $c_v = 0.171$ Btu/lbm·R (Table A-2E*a*).

Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} = \Delta U = mc_v(T_2 - T_1)$$



As the specific volume remains constant during this process, the ideal gas equation gives

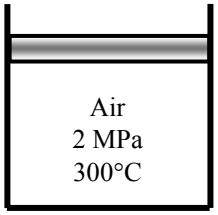
$$T_2 = T_1 \frac{P_2}{P_1} = (520 \text{ R})\frac{40 \text{ psia}}{30 \text{ psia}} = 693.3 \text{ R} = \mathbf{233.3°F}$$

Substituting,

$$W_{\text{pw,in}} = mc_v(T_2 - T_1) = (2 \text{ lbm})(0.171 \text{ Btu/lbm} \cdot \text{R})(693.3 - 520)\text{R} = \mathbf{59.3 \text{ Btu}}$$

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4-121 Air at a given state is expanded polytropically in a piston-cylinder device to a specified pressure. The final temperature is to be determined.



Analysis The polytropic relations for an ideal gas give

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{n-1/n} = (300+273\,\text{K})\left(\frac{80\,\text{kPa}}{2000\,\text{kPa}}\right)^{0.5/1.5} = \mathbf{196\,K = -77°C}$$

4-122 Nitrogen in a rigid vessel is heated. The work done and heat transfer are to be determined.

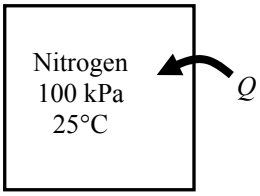
Assumptions **1** Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 126.2 K and 3.39 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for nitrogen.

Properties For nitrogen, $c_v = 0.743$ kJ/kg·K at room temperature (Table A-2*a*).

Analysis We take the nitrogen as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = mc_v(T_2 - T_1)$$



There is no work done since the vessel is rigid:

$$w = \mathbf{0\,kJ/kg}$$

Since the specific volume is constant during the process, the final temperature is determined from ideal gas equation to be

$$T_2 = T_1\frac{P_2}{P_1} = (298\,\text{K})\frac{300\,\text{kPa}}{100\,\text{psia}} = 894\,\text{K}$$

Substituting,

$$q_{\text{in}} = c_v(T_1 - T_2) = (0.743\,\text{kJ/kg}\cdot\text{K})(894-298)\text{K} = \mathbf{442.8\,kJ/kg}$$

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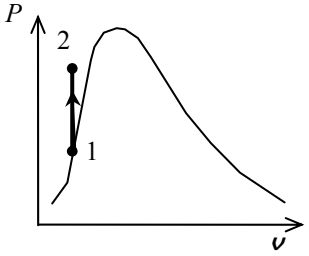
4-123 A well-insulated rigid vessel contains saturated liquid water. Electrical work is done on water. The final temperature is to be determined.

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the tank itself is negligible.

Analysis We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1)$$



The amount of electrical work added during 30 minutes period is

$$W_{\text{e,in}} = \mathbf{V}I\Delta t = (50\text{ V})(10\text{ A})(30\times 60\text{ s})\left(\frac{1\text{ W}}{1\text{ VA}}\right) = 900{,}000\text{ J} = 900\text{ kJ}$$

The properties at the initial state are (Table A-4)

$$u_1 = u_{f\,@40°\text{C}} = 167.53\text{ kJ/kg}$$

$$\boldsymbol{\nu}_1 = \boldsymbol{\nu}_{f\,@40°\text{C}} = 0.001008\text{ m}^3\text{/kg}..$$

Substituting,

$$W_{e,\text{in}} = m(u_2 - u_1) \longrightarrow u_2 = u_1 + \frac{W_{e,\text{in}}}{m} = 167.53\text{ kJ/kg} + \frac{900\text{ kJ}}{3\text{ kg}} = 467.53\text{ kJ/kg}$$

The final state is compressed liquid. Noting that the specific volume is constant during the process, the final temperature is determined using EES to be

$$\left.\begin{aligned} u_2 &= 467.53\text{ kJ/kg}\\ \boldsymbol{\nu}_2 &= \boldsymbol{\nu}_1 = 0.001008\text{ m}^3\text{/kg}\end{aligned}\right\} T_2 = \mathbf{118.9°C} \quad \text{(from EES)}$$

Discussion We cannot find this temperature directly from steam tables at the end of the book. Approximating the final compressed liquid state as saturated liquid at the given internal energy, the final temperature is determined from Table A-4 to be

$$T_2 \cong T_{\text{sat@}\,u=467.53\text{ kJ/kg}} = 111.5°\text{C}$$

Note that this approximation resulted in an answer, which is not very close to the exact answer determined using EES.

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4-124 The boundary work expression during a polytropic process of an ideal gas is to be derived.

Assumptions The process is quasi-equilibrium.

Analysis For a polytropic process,

$$P_1 v_1^n = P_2 v_2^n = \text{Constant}$$

Substituting this into the boundary work expression gives

$$\begin{aligned} w_{b,\text{out}} &= \int_1^2 P dv = P_1 v_1^n \int_1^2 v^{-n} dv = \frac{P_1 v_1}{1-n}(v_2^{1-n} - v_1^{1-n}) \\ &= \frac{P_1}{1-n}\left(\frac{v_2}{v_2^n} v_1^n - \frac{v_1}{v_1^n} v_1^n\right) \\ &= \frac{P_1 v_1}{1-n}(v_2^{1-n} v_1^{n-1} - 1) \\ &= \frac{RT_1}{1-n}\left[\left(\frac{v_2}{v_1}\right)^{1-n} - 1\right] \end{aligned}$$

When the specific volume ratio is eliminated with the expression for the constant,

$$w_{b,\text{out}} = \frac{RT_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]$$

where $n \neq 1$

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4-125 A cylinder equipped with an external spring is initially filled with air at a specified state. Heat is transferred to the air, and both the temperature and pressure rise. The total boundary work done by the air, and the amount of work done against the spring are to be determined, and the process is to be shown on a *P*-$\boldsymbol{v}$ diagram.

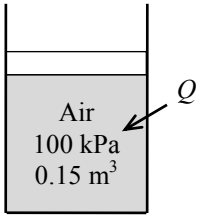
Assumptions **1** The process is quasi-equilibrium. **2** The spring is a linear spring.

Analysis (*a*) The pressure of the gas changes linearly with volume during this process, and thus the process curve on a *P*-$\boldsymbol{v}$ diagram will be a straight line. Then the boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$W_{\text{b,out}} = \text{Area} = \frac{P_1 + P_2}{2}(\boldsymbol{v}_2 - \boldsymbol{v}_1)$$

$$= \frac{(100 + 800)\text{kPa}}{2}(0.45 - 0.15)\text{m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{135\ kJ}$$

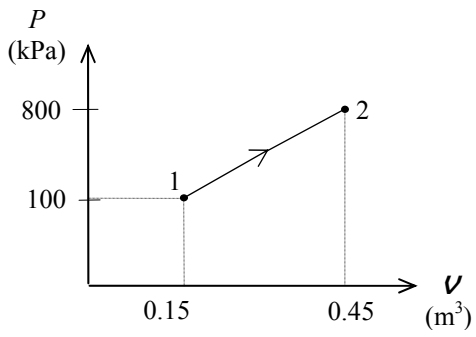


(*b*) If there were no spring, we would have a constant pressure process at P = 100 kPa. The work done during this process is

$$W_{\text{b,out,no spring}} = \int_1^2 P\,d\boldsymbol{v} = P(\boldsymbol{v}_2 - \boldsymbol{v}_1)$$

$$= (100\text{ kPa})(0.45 - 0.15)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 30\text{ kJ}$$



Thus,

$$W_{\text{spring}} = W_b - W_{\text{b,no spring}} = 135 - 30 = \mathbf{105\ kJ}$$

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4-126 A cylinder equipped with a set of stops for the piston is initially filled with saturated liquid-vapor mixture of water a specified pressure. Heat is transferred to the water until the volume increases by 20%. The initial and final temperature, the mass of the liquid when the piston starts moving, and the work done during the process are to be determined, and the process is to be shown on a *P-v* diagram.

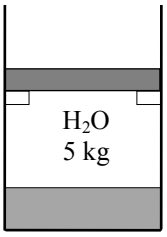
Assumptions The process is quasi-equilibrium.

Analysis (*a*) Initially the system is a saturated mixture at 125 kPa pressure, and thus the initial temperature is

$$T_1 = T_{\text{sat@125 kPa}} = \mathbf{106.0°C}$$

The total initial volume is

$$V_1 = m_f v_f + m_g v_g = 2 \times 0.001048 + 3 \times 1.3750 = 4.127 \text{ m}^3$$



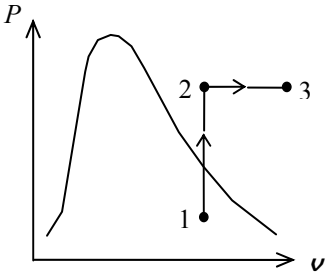
Then the total and specific volumes at the final state are

$$V_3 = 1.2V_1 = 1.2 \times 4.127 = 4.953 \text{ m}^3$$

$$v_3 = \frac{V_3}{m} = \frac{4.953 \text{ m}^3}{5 \text{ kg}} = 0.9905 \text{ m}^3/\text{kg}$$

Thus,

$$\left.\begin{array}{l} P_3 = 300 \text{ kPa} \\ v_3 = 0.9905 \text{ m}^3/\text{kg} \end{array}\right\} T_3 = \mathbf{373.6°C}$$



(*b*) When the piston first starts moving, $P_2 = 300$ kPa and $V_2 = V_1 = 4.127$ m^3. The specific volume at this state is

$$v_2 = \frac{V_2}{m} = \frac{4.127 \text{ m}^3}{5 \text{ kg}} = 0.8254 \text{ m}^3/\text{kg}$$

which is greater than $v_g = 0.60582$ m^3/kg at 300 kPa. Thus **no liquid** is left in the cylinder when the piston starts moving.

(*c*) No work is done during process 1-2 since $V_1 = V_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_b = \int_2^3 P dV = P_2(V_3 - V_2) = (300 \text{ kPa})(4.953 - 4.127)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{247.6 \text{ kJ}}$$

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4-127E A spherical balloon is initially filled with air at a specified state. The pressure inside is proportional to the square of the diameter. Heat is transferred to the air until the volume doubles. The work done is to be determined.

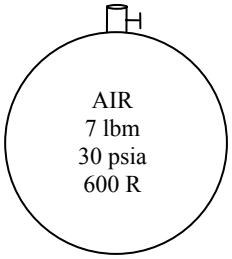
Assumptions **1** Air is an ideal gas. **2** The process is quasi-equilibrium.

Properties The gas constant of air is R = 0.06855 Btu/lbm.R (Table A-1E).

Analysis The dependence of pressure on volume can be expressed as

$$\boldsymbol{V} = \frac{1}{6}\pi D^3 \longrightarrow D = \left(\frac{6\boldsymbol{V}}{\pi}\right)^{1/3}$$

$$P \propto D^2 \longrightarrow P = kD^2 = k\left(\frac{6\boldsymbol{V}}{\pi}\right)^{2/3}$$



or,

$$k\left(\frac{6}{\pi}\right)^{2/3} = P_1\boldsymbol{V}_1^{-2/3} = P_2\boldsymbol{V}_2^{-2/3}$$

Also,

$$\frac{P_2}{P_1} = \left(\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1}\right)^{2/3} = 2^{2/3} = 1.587$$

and

$$\frac{P_1\boldsymbol{V}_1}{T_1} = \frac{P_2\boldsymbol{V}_2}{T_2} \longrightarrow T_2 = \frac{P_2\boldsymbol{V}_2}{P_1\boldsymbol{V}_1}T_1 = 1.587 \times 2 \times (600\ \text{R}) = 1905\ \text{R}$$

Thus,

$$W_b = \int_1^2 P d\boldsymbol{V} = \int_1^2 k\left(\frac{6\boldsymbol{V}}{\pi}\right)^{2/3} d\boldsymbol{V} = \frac{3k}{5}\left(\frac{6}{\pi}\right)^{2/3}\left(\boldsymbol{V}_2^{5/3} - \boldsymbol{V}_1^{5/3}\right) = \frac{3}{5}(P_2\boldsymbol{V}_2 - P_1\boldsymbol{V}_1)$$

$$= \frac{3}{5}mR(T_2 - T_1) = \frac{3}{5}(7\ \text{lbm})(0.06855\ \text{Btu/lbm} \cdot \text{R})(1905 - 600)\text{R}$$

$$= \mathbf{376\ Btu}$$

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4-128E Problem 4-127E is reconsidered. Using the integration feature, the work done is to be determined and compared to the 'hand calculated' result.

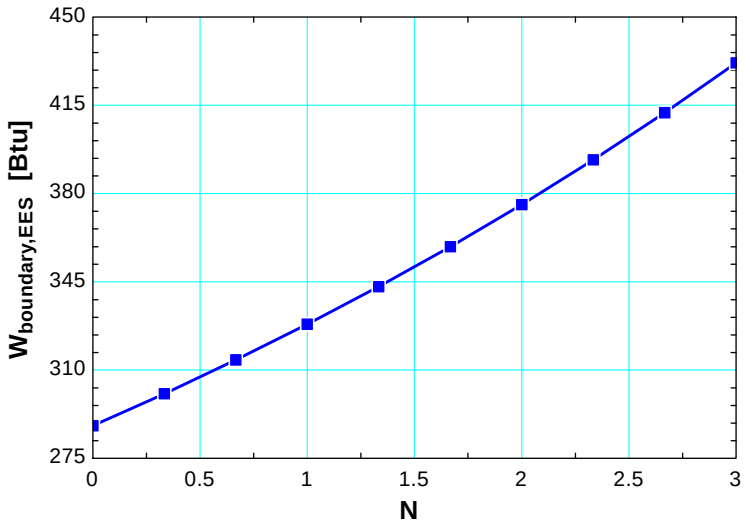
Analysis The problem is solved using EES, and the solution is given below.

```
N=2
m=7 [lbm]
P_1=30 [psia]
T_1=600 [R]
V_2=2*V_1
R=R_u/molarmass(air)
R_u=1545 [ft-lbf/lbmol-R]
P_1*Convert(psia,lbf/ft^2)*V_1=m*R*T_1
V_1=4*pi*(D_1/2)^3/3
C=P_1/D_1^N
(D_1/D_2)^3=V_1/V_2
P_2=C*D_2^N
P_2*Convert(psia,lbf/ft^2)*V_2=m*R*T_2

P=C*D^N*Convert(psia,lbf/ft^2)
V=4*pi*(D/2)^3/3

W_boundary_EES=integral(P,V,V_1,V_2)*convert(ft-lbf,Btu)
W_boundary_HAND=pi*C/(2*(N+3))*(D_2^(N+3)-D_1^(N+3))*convert(ft-lbf,Btu)*convert(ft^2,in^2)
```

N	$W_{boundary}$ [Btu]
0	287.9
0.3333	300.6
0.6667	314
1	328.1
1.333	343.1
1.667	358.9
2	375.6
2.333	393.3
2.667	412
3	431.8



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4-129 A cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant is heated both electrically and by heat transfer at constant pressure for 6 min. The electric current is to be determined, and the process is to be shown on a T-v diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself and the wires is negligible. **3** The compression or expansion process is quasi-equilibrium.

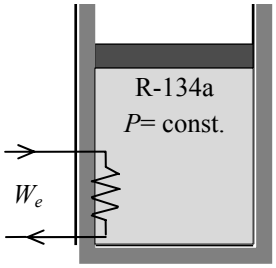
Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{e,in}} - W_{b,out} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{in}} + W_{\text{e,in}} = m(h_2 - h_1)$$

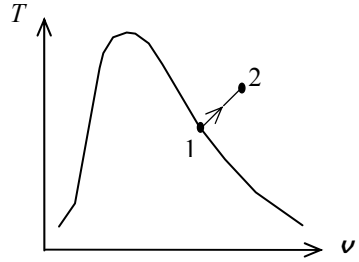
$$Q_{\text{in}} + (\text{V}I\Delta t) = m(h_2 - h_1)$$



since $\Delta U + W_{\text{b}} = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 240 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} h_1 = h_{g@240\text{kPa}} = 247.28 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 240 \text{ kPa} \\ T_1 = 70°\text{C} \end{array}\right\} h_2 = 314.51 \text{ kJ/kg}$$



Substituting,

$$300{,}000 \text{ VAs} + (110 \text{ V})(I)(6 \times 60 \text{ s}) = (12 \text{ kg})(314.51 - 247.28)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

$$I = \mathbf{12.8\ A}$$

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4-130 Saturated water vapor contained in a spring-loaded piston-cylinder device is condensed until it is saturated liquid at a specified temperature. The heat transfer is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

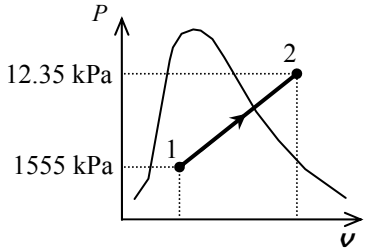
Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$

$$q_{\text{out}} = w_{b,in} - (u_2 - u_1)$$



The properties at the initial and final states are (Table A-4),

$$P_1 = P_{\text{sat@200°C}} = 1555\ \text{kPa}$$

$$\nu_1 = \nu_{g@200°\text{C}} = 0.1272\ \text{m}^3/\text{kg}$$

$$u_1 = u_{g@200°\text{C}} = 2594.2\ \text{kJ/kg}$$

$$P_2 = P_{\text{sat@50°C}} = 12.35\ \text{kPa}$$

$$\nu_2 = \nu_{f@50°\text{C}} = 0.001012\ \text{m}^3/\text{kg}$$

$$u_2 = u_{f@50°\text{C}} = 209.33\ \text{kJ/kg}$$

Since this is a linear process, the work done is equal to the area under the process line 1-2:

$$w_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(\nu_2 - \nu_1)$$

$$= \frac{(1555 + 12.35)\text{kPa}}{2}(0.001012 - 0.1272\ \text{m}^3/\text{kg})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = -98.9\ \text{kJ/kg}$$

Thus,

$$w_{b,\text{in}} = 98.9\ \text{kJ/kg}$$

Substituting into energy balance equation gives

$$q_{\text{out}} = w_{b,in} - (u_2 - u_1) = 98.9\ \text{kJ/kg} - (209.33 - 2594.2)\ \text{kJ/kg} = \mathbf{2484\ kJ/kg}$$

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4-131 A cylinder is initially filled with helium gas at a specified state. Helium is compressed polytropically to a specified temperature and pressure. The heat transfer during the process is to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

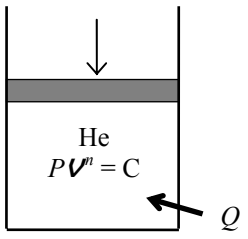
Properties The gas constant of helium is $R = 2.0769$ kPa.m^3/kg.K (Table A-1). Also, $c_v = 3.1156$ kJ/kg.K (Table A-2).

Analysis The mass of helium and the exponent n are determined to be

$$m = \frac{P_1 V_1}{RT_1} = \frac{(150\ \text{kPa})(0.5\ \text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 0.123\ \text{kg}$$

$$\frac{P_1 V_1}{RT_1} = \frac{P_2 V_2}{RT_2} \longrightarrow V_2 = \frac{T_2 P_1}{T_1 P_2} V_1 = \frac{413\ \text{K}}{293\ \text{K}} \times \frac{150\ \text{kPa}}{400\ \text{kPa}} \times 0.5\ \text{m}^3 = 0.264\ \text{m}^3$$

$$P_2 V_2^n = P_1 V_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{V_1}{V_2}\right)^n \longrightarrow \frac{400}{150} = \left(\frac{0.5}{0.264}\right)^n \longrightarrow n = 1.536$$



Then the boundary work for this polytropic process can be determined from

$$W_{b,in} = -\int_1^2 P\,dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.123\ \text{kg})(2.0769\ \text{kJ/kg}\cdot\text{K})(413-293)\text{K}}{1-1.536} = 57.2\ \text{kJ}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Taking the direction of heat transfer to be to the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} + W_{b,in} = \Delta U = m(u_2 - u_1)$$

$$Q_{in} = m(u_2 - u_1) - W_{b,in}$$

$$= mc_v(T_2 - T_1) - W_{b,in}$$

Substituting,

$$Q_{in} = (0.123\ \text{kg})(3.1156\ \text{kJ/kg}\cdot\text{K})(413 - 293)\text{K} - (57.2\ \text{kJ}) = \mathbf{-11.2\ kJ}$$

The negative sign indicates that heat is lost from the system.

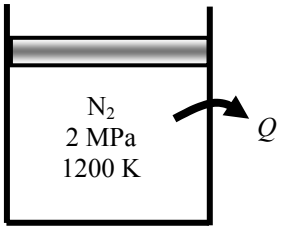
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4-132 Nitrogen gas is expanded in a polytropic process. The work done and the heat transfer are to be determined.

Assumptions **1** Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of 126.2 K and 3.39 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used.

Properties The properties of nitrogen are $R = 0.2968$ kJ/kg·K and $c_v = 0.743$ kJ/kg·K (Table A-2*a*).

Analysis We take nitrogen in the cylinder as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$

Using the boundary work relation for the polytropic process of an ideal gas gives

$$w_{b,\text{out}} = \frac{RT_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] = \frac{(0.2968\,\text{kJ/kg}\cdot\text{K})(1200\,\text{K})}{1-1.45}\left[\left(\frac{120}{2000}\right)^{0.45/1.45} - 1\right] = \mathbf{404.1\,kJ/kg}$$

The temperature at the final state is

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n-1)/n} = (1200\,\text{K})\left(\frac{200\,\text{kPa}}{2000\,\text{kPa}}\right)^{0.45/1.45} = 587.3\,\text{K}$$

Substituting into the energy balance equation,

$$q_{\text{in}} = w_{b,\text{out}} + c_v(T_2 - T_1) = 404.1\,\text{kJ/kg} + (0.743\,\text{kJ/kg}\cdot\text{K})(587.3 - 1200)\text{K} = \mathbf{-51.1\,kJ/kg}$$

The negative sign indicates that heat is lost from the device. That is,

$$q_{\text{out}} = \mathbf{51.1\,kJ/kg}$$

4-133 A cylinder and a rigid tank initially contain the same amount of an ideal gas at the same state. The temperature of both systems is to be raised by the same amount. The amount of extra heat that must be transferred to the cylinder is to be determined.

Analysis In the absence of any work interactions, other than the boundary work, the ΔH and ΔU represent the heat transfer for ideal gases for constant pressure and constant volume processes, respectively. Thus the extra heat that must be supplied to the air maintained at constant pressure is

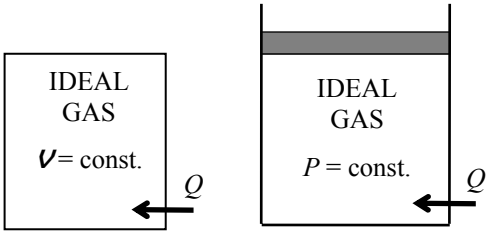
$$Q_{\text{in, extra}} = \Delta H - \Delta U = mc_p\Delta T - mc_v\Delta T = m(c_p - c_v)\Delta T = mR\Delta T$$

where

$$R = \frac{R_u}{M} = \frac{8.314\,\text{kJ/kmol}\cdot\text{K}}{25\,\text{kg/kmol}} = 0.3326\,\text{kJ/kg}\cdot\text{K}$$

Substituting,

$$Q_{\text{in, extra}} = (12\,\text{kg})(0.3326\,\text{kJ/kg}\cdot\text{K})(15\,\text{K}) = \mathbf{59.9\,kJ}$$



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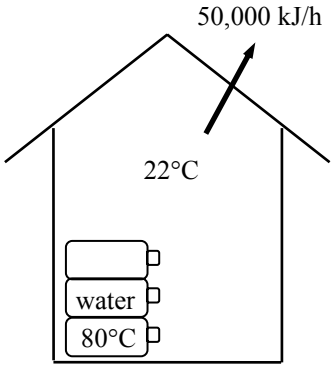
4-134 The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night with or without solar heating are to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times.

Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and c = 4.18 kJ/kg·°C (Table A-3).

Analysis (a) The total mass of water is

$$m_w = \rho V = (1 \text{ kg/L})(50 \times 20 \text{ L}) = 1000 \text{ kg}$$



Taking the contents of the house, including the water as our system, the energy balance relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$
$$= (\Delta U)_{\text{water}} = mc(T_2 - T_1)_{\text{water}}$$

or,

$$\dot{W}_{\text{e,in}}\Delta t - Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$(15 \text{ kJ/s})\Delta t - (50{,}000 \text{ kJ/h})(10 \text{ h}) = (1000 \text{ kg})(4.18 \text{ kJ/kg·°C})(22 - 80)°\text{C}$$

It gives

$$\Delta t = 17{,}170 \text{ s} = \mathbf{4.77\ h}$$

(b) If the house incorporated no solar heating, the energy balance relation above would simplify further to

$$\dot{W}_{\text{e,in}}\Delta t - Q_{\text{out}} = 0$$

Substituting,

$$(15 \text{ kJ/s})\Delta t - (50{,}000 \text{ kJ/h})(10 \text{ h}) = 0$$

It gives

$$\Delta t = 33{,}333 \text{ s} = \mathbf{9.26\ h}$$

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4-135 An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature is to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible.

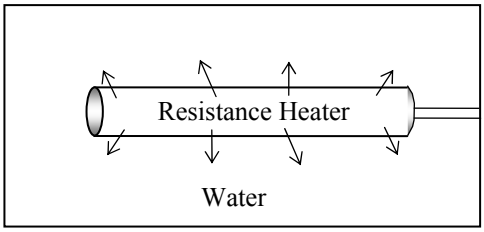
Properties The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-3).

Analysis Taking the water in the container as the system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = (\Delta U)_{\text{water}}$$

$$\dot{W}_{\text{e,in}}\Delta t = mc(T_2 - T_1)_{\text{water}}$$



Substituting,

$$(1800 \text{ J/s})\Delta t = (40 \text{ kg})(4180 \text{ J/kg}\cdot°\text{C})(80 - 20)°\text{C}$$

Solving for Δt gives

$$\Delta t = \mathbf{5573 \text{ s} = 92.9 \text{ min} = 1.55 \text{ h}}$$

4-136 One ton of liquid water at 50°C is brought into a room. The final equilibrium temperature in the room is to be determined.

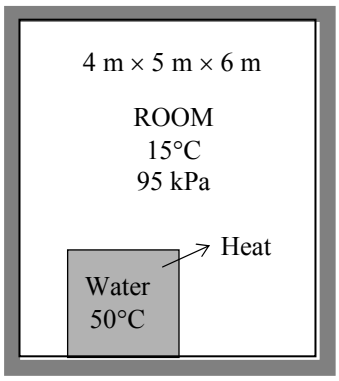
Assumptions **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-3).

Analysis The volume and the mass of the air in the room are

$$V = 4 \times 5 \times 6 = 120 \text{ m}^3$$

$$m_{\text{air}} = \frac{P_1 V_1}{RT_1} = \frac{(95 \text{ kPa})(120 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288 \text{ K})} = 137.9 \text{ kg}$$



Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

or

$$[mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$$

Substituting,

$$(1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot°\text{C})(T_f - 50)°\text{C} + (137.9 \text{ kg})(0.718 \text{ kJ/kg}\cdot°\text{C})(T_f - 15)°\text{C} = 0$$

It gives

$$T_f = \mathbf{49.2°C}$$

where T_f is the final equilibrium temperature in the room.

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4-137 A room is to be heated by 1 ton of hot water contained in a tank placed in the room. The minimum initial temperature of the water is to be determined if it to meet the heating requirements of this room for a 24-h period.

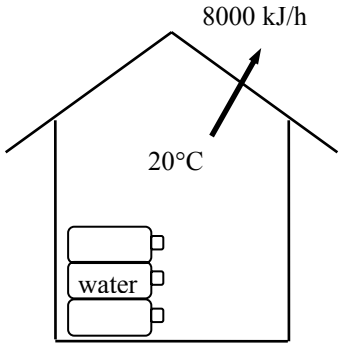
Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 24-h period.

Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis Heat loss from the room during a 24-h period is

$$Q_{\text{loss}} = (8000 \text{ kJ/h})(24 \text{ h}) = 192{,}000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + \cancelto{0}{(\Delta U)_{\text{air}}}$$

or

$$-Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$-192{,}000 \text{ kJ} = (1000 \text{ kg})(4.18 \text{ kJ/kg·°C})(20 - T_1)$$

It gives

$$T_1 = \mathbf{65.9°C}$$

where T_1 is the temperature of the water when it is first brought into the room.

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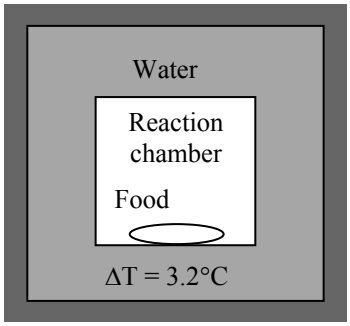
4-138 A sample of a food is burned in a bomb calorimeter, and the water temperature rises by 3.2°C when equilibrium is established. The energy content of the food is to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the reaction chamber is negligible relative to the energy stored in water. **4** The energy supplied by the mixer is negligible.

Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3). The constant volume specific heat of air at room temperature is $c_v = 0.718$ kJ/kg·°C (Table A-2).

Analysis The chemical energy released during the combustion of the sample is transferred to the water as heat. Therefore, disregarding the change in the sensible energy of the reaction chamber, the energy content of the food is simply the heat transferred to the water. Taking the water as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \rightarrow Q_{\text{in}} = \Delta U$$



or

$$Q_{in} = (\Delta U)_{\text{water}} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$Q_{\text{in}} = (3 \text{ kg})(4.18 \text{ kJ/kg·°C})(3.2°\text{C}) = 40.13 \text{ kJ}$$

for a 2-g sample. Then the energy content of the food per unit mass is

$$\frac{40.13 \text{ kJ}}{2 \text{ g}}\left(\frac{1000 \text{ g}}{1 \text{ kg}}\right) = \mathbf{20{,}060 \ kJ/kg}$$

To make a rough estimate of the error involved in neglecting the thermal energy stored in the reaction chamber, we treat the entire mass within the chamber as air and determine the change in sensible internal energy:

$$(\Delta U)_{\text{chamber}} = [mc_v(T_2 - T_1)]_{\text{chamber}} = (0.102 \text{ kg})(0.718 \text{ kJ/kg·}^\circ\text{C})(3.2^\circ\text{C}) = 0.23 \text{ kJ}$$

which is less than 1% of the internal energy change of water. Therefore, it is reasonable to disregard the change in the sensible energy content of the reaction chamber in the analysis.

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4-139 A man drinks one liter of cold water at 3°C in an effort to cool down. The drop in the average body temperature of the person under the influence of this cold water is to be determined.

Assumptions **1** Thermal properties of the body and water are constant. **2** The effect of metabolic heat generation and the heat loss from the body during that time period are negligible.

Properties The density of water is very nearly 1 kg/L, and the specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3). The average specific heat of human body is given to be 3.6 kJ/kg.°C.

Analysis. The mass of the water is

$$m_w = \rho V = (1\text{ kg/L})(1\text{ L}) = 1\text{ kg}$$

We take the man and the water as our system, and disregard any heat and mass transfer and chemical reactions. Of course these assumptions may be acceptable only for very short time periods, such as the time it takes to drink the water. Then the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$



$$0 = \Delta U = \Delta U_{\text{body}} + \Delta U_{\text{water}}$$

or $$[mc(T_2 - T_1)]_{\text{body}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting $$(68\text{ kg})(3.6\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 39)^\circ\text{C} + (1\text{ kg})(4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 3)^\circ\text{C} = 0$$

It gives

$$T_f = 38.4°\text{C}$$

Then

$$\Delta T = 39 - 38.4 = \mathbf{0.6°C}$$

Therefore, the average body temperature of this person should drop about half a degree celsius.

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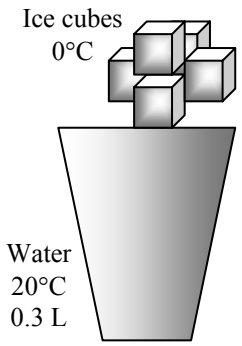
4-140 A 0.3-L glass of water at 20°C is to be cooled with ice to 5°C. The amount of ice or cold water that needs to be added to the water is to be determined.

Assumptions **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the glass is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

Properties The density of water is 1 kg/L, and the specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3). The specific heat of ice at about 0°C is $c = 2.11$ kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg,.

Analysis (*a*) The mass of the water is

$$m_w = \rho V = (1\,\text{kg/L})(0.3\,\text{L}) = 0.3\,\text{kg}$$



We take the ice and the water as our system, and disregard any heat and mass transfer. This is a reasonable assumption since the time period of the process is very short. Then the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mc(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Noting that $T_{1,\text{ice}} = 0$°C and $T_2 = 5$°C and substituting gives

$$m[0 + 333.7\,\text{kJ/kg} + (4.18\,\text{kJ/kg·°C})(5-0)\text{°C}] + (0.3\,\text{kg})(4.18\,\text{kJ/kg·°C})(5-20)\text{°C} = 0$$

$$m = 0.0546\,\text{kg} = \mathbf{54.6\ g}$$

(*b*) When $T_{1,\text{ice}} = -20$°C instead of 0°C, substituting gives

$$m[(2.11\,\text{kJ/kg·°C})[0-(-20)]\text{°C} + 333.7\,\text{kJ/kg} + (4.18\,\text{kJ/kg·°C})(5-0)\text{°C}]$$
$$+ (0.3\,\text{kg})(4.18\,\text{kJ/kg·°C})(5-20)\text{°C} = 0$$

$$m = 0.0487\,\text{kg} = \mathbf{48.7\ g}$$

Cooling with cold water can be handled the same way. All we need to do is replace the terms for ice by a term for cold water at 0°C:

$$(\Delta U)_{\text{cold water}} + (\Delta U)_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{cold water}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$[m_{\text{cold water}}\,(4.18\,\text{kJ/kg·°C})(5-0)\text{°C}] + (0.3\,\text{kg})(4.18\,\text{kJ/kg·°C})(5-20)\text{°C} = 0$$

It gives

$$m = 0.9\,\text{kg} = \mathbf{900\ g}$$

Discussion Note that this is about 16 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks. Also, the temperature of ice does not seem to make a significant difference.

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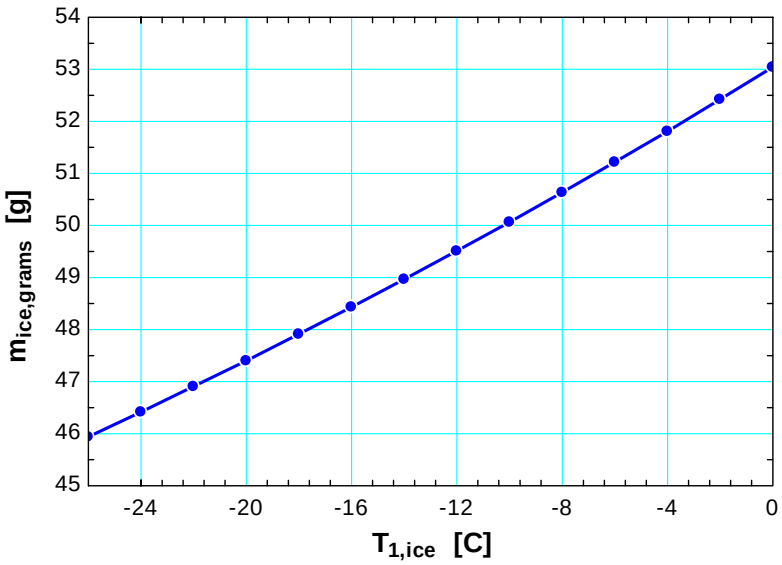
4-141 Problem 4-140 is reconsidered. The effect of the initial temperature of the ice on the final mass of ice required as the ice temperature varies from -26°C to 0°C is to be investigated. The mass of ice is to be plotted against the initial temperature of ice.

Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns"
rho_water = 1 [kg/L]
V = 0.3 [L]
T_1_ice = 0 [C]
T_1 = 20 [C]
T_2 = 5 [C]
C_ice = 2.11 [kJ/kg-C]
C_water = 4.18 [kJ/kg-C]
h_if = 333.7 [kJ/kg]
T_1_ColdWater = 0 [C]

m_water = rho_water*V "[kg]" "The mass of the water"
"The system is the water plus the ice.  Assume a short time period and neglect
any heat  and mass transfer.  The energy balance becomes:"
E_in - E_out = DELTAE_sys "[kJ]"
E_in = 0 "[kJ]"
E_out = 0"[kJ]"
DELTAE_sys = DELTAU_water+DELTAU_ice"[kJ]"
DELTAU_water = m_water*C_water*(T_2 - T_1)"[kJ]"
DELTAU_ice = DELTAU_solid_ice+DELTAU_melted_ice"[kJ]"
DELTAU_solid_ice =m_ice*C_ice*(0-T_1_ice) + m_ice*h_if"[kJ]"
DELTAU_melted_ice=m_ice*C_water*(T_2 - 0)"[kJ]"
m_ice_grams=m_ice*convert(kg,g)"[g]"
"Cooling with Cold Water:"
DELTAE_sys = DELTAU_water+DELTAU_ColdWater"[kJ]"
DELTAU_water = m_water*C_water*(T_2_ColdWater - T_1)"[kJ]"
DELTAU_ColdWater = m_ColdWater*C_water*(T_2_ColdWater - T_1_ColdWater)"[kJ]"
m_ColdWater_grams=m_ColdWater*convert(kg,g)"[g]"
```

$T_{1,ice}$ [C]	$m_{ice,grams}$ [g]
-26	45.94
-24	46.42
-22	46.91
-20	47.4
-18	47.91
-16	48.43
-14	48.97
-12	49.51
-10	50.07
-8	50.64
-6	51.22
-4	51.81
-2	52.42
0	53.05



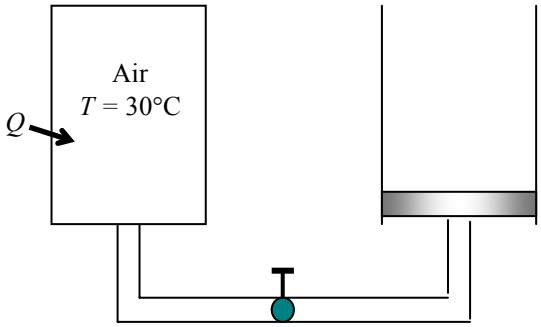
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4-142 A rigid tank filled with air is connected to a cylinder with zero clearance. The valve is opened, and air is allowed to flow into the cylinder. The temperature is maintained at 30°C at all times. The amount of heat transfer with the surroundings is to be determined.

Assumptions **1** Air is an ideal gas. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** There are no work interactions involved other than the boundary work.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1).

Analysis We take the entire air in the tank and the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1) = 0$$

$$Q_{\text{in}} = W_{\text{b,out}}$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$. The initial volume of air is

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow V_2 = \frac{P_1}{P_2}\frac{T_2}{T_1} V_1 = \frac{400 \text{ kPa}}{200 \text{ kPa}} \times 1 \times (0.4 \text{ m}^3) = 0.80 \text{ m}^3$$

The pressure at the piston face always remains constant at 200 kPa. Thus the boundary work done during this process is

$$W_{\text{b,out}} = \int_1^2 P dV = P_2(V_2 - V_1) = (200 \text{ kPa})(0.8 - 0.4)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = 80 \text{ kJ}$$

Therefore, the heat transfer is determined from the energy balance to be

$$W_{\text{b,out}} = Q_{\text{in}} = \mathbf{80\ kJ}$$

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4-143 A well-insulated room is heated by a steam radiator, and the warm air is distributed by a fan. The average temperature in the room after 30 min is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energy changes are negligible. **3** The air pressure in the room remains constant and thus the air expands as it is heated, and some warm air escapes.

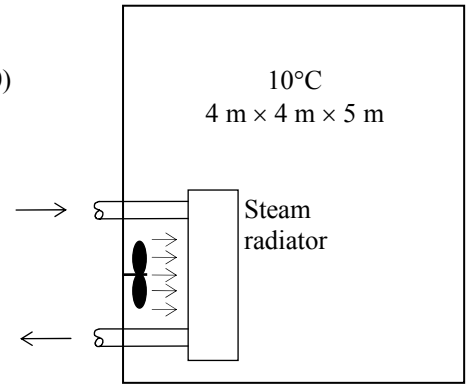
Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). Also, $c_p = 1.005$ kJ/kg.K for air at room temperature (Table A-2).

Analysis We first take the radiator as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = m(u_1 - u_2)$$



Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{matrix} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{matrix}\right\} \begin{matrix} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 100 \text{ kPa} \\ (v_2 = v_1) \end{matrix}\right\} \begin{matrix} v_f = 0.001043, \quad v_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, \quad u_{fg} = 2088.2 \text{ kJ/kg} \end{matrix}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{1.08049 - 0.001043}{1.6941 - 0.001043} = 0.6376$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + 0.6376 \times 2088.2 = 1748.7 \text{ kJ/kg}$$

$$m = \frac{V_1}{v_1} = \frac{0.015 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.0139 \text{ kg}$$

Substituting, $Q_{\text{out}} = (0.0139 \text{ kg})(2654.6 - 1748.7)\text{kJ/kg} = 12.58 \text{ kJ}$

The volume and the mass of the air in the room are $V = 4 \times 4 \times 5 = 80 \text{ m}^3$ and

$$m_{\text{air}} = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(80 \text{ m}^3)}{(0.2870 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(283 \text{ K})} = 98.5 \text{ kg}$$

The amount of fan work done in 30 min is

$$W_{\text{fan,in}} = \dot{W}_{\text{fan,in}} \Delta t = (0.120 \text{ kJ/s})(30 \times 60 \text{ s}) = 216 \text{kJ}$$

We now take the air in the room as the system. The energy balance for this closed system is expressed as

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$Q_{\text{in}} + W_{\text{fan,in}} - W_{\text{b,out}} = \Delta U$$

$$Q_{\text{in}} + W_{\text{fan,in}} = \Delta H \cong mc_p(T_2 - T_1)$$

since the boundary work and ΔU combine into ΔH for a constant pressure expansion or compression process. It can also be expressed as

$$(\dot{Q}_{\text{in}} + \dot{W}_{\text{fan,in}})\Delta t = mc_{p,\text{avg}}(T_2 - T_1)$$

Substituting, $(12.58 \text{ kJ}) + (216 \text{ kJ}) = (98.5 \text{ kg})(1.005 \text{ kJ/kg°C})(T_2 - 10)°\text{C}$

which yields

$T_2 =$ **12.3°C**

Therefore, the air temperature in the room rises from 10°C to 12.3°C in 30 min.

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4-144 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

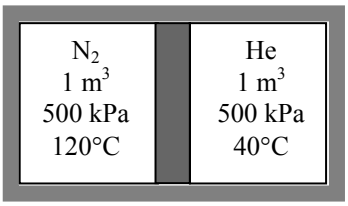
Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

Properties The gas constants and the constant volume specific heats are $R = 0.2968$ kPa.m^3/kg.K is $c_v = 0.743$ kJ/kg·°C for N_2, and $R = 2.0769$ kPa.m^3/kg.K is $c_v = 3.1156$ kJ/kg·°C for He (Tables A-1 and A-2)

Analysis The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(393\ \text{K})} = 4.287\ \text{kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(313\ \text{K})} = 0.7691\ \text{kg}$$



Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He}$$

Substituting,

$$(4.287\ \text{kg})(0.743\ \text{kJ/kg}\cdot°\text{C})(T_f - 120)°\text{C} + (0.7691\ \text{kg})(3.1156\ \text{kJ/kg}\cdot°\text{C})(T_f - 40)°\text{C} = 0$$

It gives

$$T_f = \mathbf{85.7°C}$$

where T_f is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

Discussion Using the relation $PV = NR_uT$, it can be shown that the total number of moles in the cylinder is 0.153 + 0.192 = 0.345 kmol, and the final pressure is 515 kPa.

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4-145 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

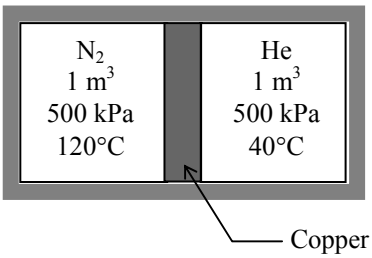
Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

Properties The gas constants and the constant volume specific heats are $R = 0.2968$ kPa.m^3/kg.K is $c_v = 0.743$ kJ/kg·°C for N_2, and $R = 2.0769$ kPa.m^3/kg.K is $c_v = 3.1156$ kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of copper piston is $c = 0.386$ kJ/kg·°C (Table A-3).

Analysis The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(393\ \text{K})} = 4.287\ \text{kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(313\ \text{K})} = 0.7691\ \text{kg}$$



Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He} + [mc(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\,Cu} = (120 + 40)/2 = 80°\text{C}$$

Substituting,

$$(4.287\ \text{kg})(0.743\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_f - 120)°\text{C} + (0.7691\ \text{kg})(3.1156\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_f - 40)°\text{C} + (8\ \text{kg})(0.386\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_f - 80)°\text{C} = 0$$

It gives

$$T_f = \mathbf{83.7°C}$$

where T_f is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

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4-146 Problem 4-145 is reconsidered. The effect of the mass of the copper piston on the final equilibrium temperature as the mass of piston varies from 1 kg to 10 kg is to be investigated. The final temperature is to be plotted against the mass of piston.

Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns:"
R_u=8.314 [kJ/kmol-K]
V_N2[1]=1 [m^3]
Cv_N2=0.743 [kJ/kg-K]  "From Table A-2(a) at 27C"
R_N2=0.2968 [kJ/kg-K]  "From Table A-2(a)"
T_N2[1]=120 [C]
P_N2[1]=500 [kPa]
V_He[1]=1 [m^3]
Cv_He=3.1156 [kJ/kg-K] "From Table A-2(a) at 27C"
T_He[1]=40 [C]
P_He[1]=500 [kPa]
R_He=2.0769 [kJ/kg-K]  "From Table A-2(a)"
m_Pist=8 [kg]
Cv_Pist=0.386 [kJ/kg-K]  "Use Cp for Copper from Table A-3(b) at 27C"

"Solution:"
"mass calculations:"
P_N2[1]*V_N2[1]=m_N2*R_N2*(T_N2[1]+273)
P_He[1]*V_He[1]=m_He*R_He*(T_He[1]+273)

"The entire cylinder is considered to be a closed system, neglecting the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_negPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_neglPist
E_in =0 [kJ]
E_out = 0 [kJ]

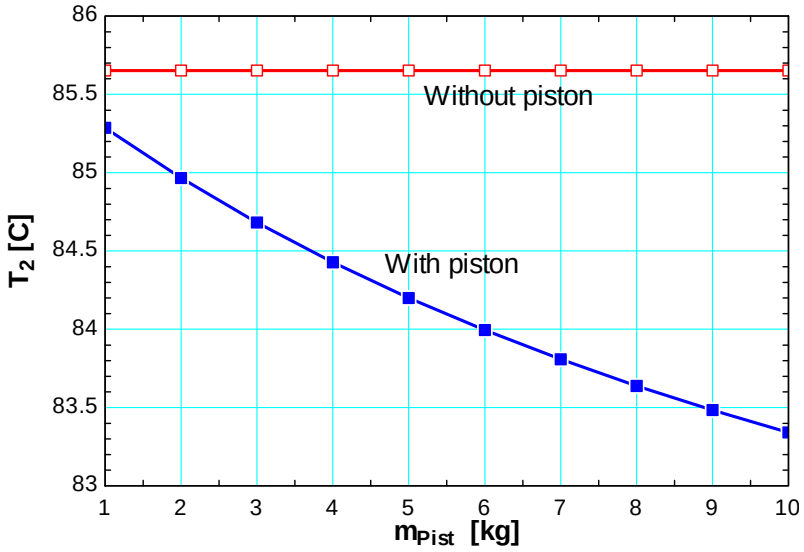
"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_neglPist= m_N2*Cv_N2*(T_2_neglPist-T_N2[1])+m_He*Cv_He*(T_2_neglPist-T_He[1])
"The entire cylinder is considered to be a closed system, including the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_withPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_withPist

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_withPist= m_N2*Cv_N2*(T_2_withPist-T_N2[1])+m_He*Cv_He*(T_2_withPist-
T_He[1])+m_Pist*Cv_Pist*(T_2_withPist-T_Pist[1])
T_Pist[1]=(T_N2[1]+T_He[1])/2
"Total volume of gases:"
V_total=V_N2[1]+V_He[1]
"Final pressure at equilibrium:"
"Neglecting effect of piston, P_2 is:"
P_2_neglPist*V_total=N_total*R_u*(T_2_neglPist+273)

"Including effect of piston, P_2 is:"
N_total=m_N2/molarmass(nitrogen)+m_He/molarmass(Helium)
P_2_withPist*V_total=N_total*R_u*(T_2_withPist+273)
```

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m_{Pist} [kg]	$T_{2,neglPist}$ [C]	$T_{2,withPist}$ [C]
1	85.65	85.29
2	85.65	84.96
3	85.65	84.68
4	85.65	84.43
5	85.65	84.2
6	85.65	83.99
7	85.65	83.81
8	85.65	83.64
9	85.65	83.48
10	85.65	83.34



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4-147 A piston-cylinder device initially contains saturated liquid water. An electric resistor placed in the tank is turned on until the tank contains saturated water vapor. The volume of the tank, the final temperature, and the power rating of the resistor are to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

Properties The initial properties of steam are (Table A-4)

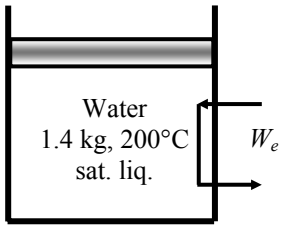
$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} \boldsymbol{\nu}_1 = 0.001157\ \text{m}^3/\text{kg} \\ h_1 = 852.26\ \text{kJ/kg} \\ P_1 = 1554.9\ \text{kPa} \end{array}$$

Analysis (*a*) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{\text{e,in}} = W_{\text{b,out}} + \Delta U = \Delta H = m(h_2 - h_1)$$



since

$$W_{\text{b,out}} + \Delta U = \Delta H \quad \text{for a constant-pressure process.}$$

The initial and final volumes are

$$\boldsymbol{V}_1 = m\boldsymbol{\nu}_1 = (1.4\ \text{kg})(0.001157\ \text{m}^3/\text{kg}) = 0.001619\ \text{m}^3$$

$$\boldsymbol{V}_2 = 4(0.001619\ \text{m}^3) = \mathbf{0.006476\ m^3}$$

(b) Now, the final state can be fixed by calculating specific volume

$$\boldsymbol{\nu}_2 = \frac{\boldsymbol{V}_2}{m} = \frac{0.006476\ \text{m}^3}{1.4\ \text{kg}} = 0.004626\ \text{m}^3/\text{kg}$$

The final state is saturated mixture and both pressure and temperature remain constant during the process. Other properties are

$$\left.\begin{array}{l} P_2 = P_1 = 1554.9\ \text{kPa} \\ \boldsymbol{\nu}_2 = 0.004626\ \text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = T_1 = \mathbf{200°C} \\ h_2 = 905.65\ \text{kJ/kg} \\ x_2 = 0.02752 \end{array} \quad (\text{Table A-4 or EES})$$

(*c*) Substituting,

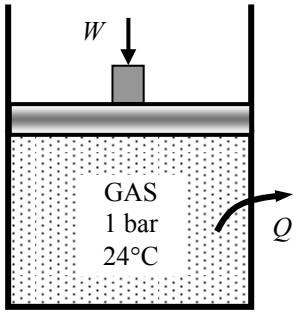
$$W_{\text{e,in}} = (1.4\ \text{kg})(905.65 - 852.26)\text{kJ/kg} = 74.75\ \text{kJ}$$

Finally, the power rating of the resistor is

$$\dot{W}_{\text{e,in}} = \frac{W_{\text{e,in}}}{\Delta t} = \frac{74.75\ \text{kJ}}{20 \times 60\ \text{s}} = \mathbf{0.0623\ kW}$$

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4-148 A piston-cylinder device contains an ideal gas. An external shaft connected to the piston exerts a force. For an isothermal process of the ideal gas, the amount of heat transfer, the final pressure, and the distance that the piston is displaced are to be determined.



Assumptions **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The friction between the piston and the cylinder is negligible.

Analysis (*a*) We take the ideal gas in the cylinder to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U_{\text{ideal gas}} \cong mc_v (T_2 - T_1)_{\text{ideal gas}}) = 0 \quad (\text{since } T_2 = T_1 \text{ and } KE = PE = 0)$$

$$W_{\text{b,in}} = Q_{\text{out}}$$

Thus, the amount of heat transfer is equal to the boundary work input

$$Q_{\text{out}} = W_{\text{b,in}} = \mathbf{0.1\,kJ}$$

(*b*) The relation for the isothermal work of an ideal gas may be used to determine the final volume in the cylinder. But we first calculate initial volume

$$V_1 = \frac{\pi D^2}{4} L_1 = \frac{\pi (0.12\,\text{m})^2}{4} (0.2\,\text{m}) = 0.002262\,\text{m}^3$$

Then,

$$W_{\text{b,in}} = -P_1 V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$0.1\,\text{kJ} = -(100\ \text{kPa})(0.002262\ \text{m}^3)\ln\left(\frac{V_2}{0.002262\ \text{m}^3}\right) \longrightarrow V_2 = 0.001454\ \text{m}^3$$

The final pressure can be determined from ideal gas relation applied for an isothermal process

$$P_1 V_1 = P_2 V_2 \longrightarrow (100\,\text{kPa})(0.002262\,\text{m}^3) = P_2 (0.001454\,\text{m}^3) \longrightarrow P_2 = \mathbf{155.6\,kPa}$$

(*c*) The final position of the piston and the distance that the piston is displaced are

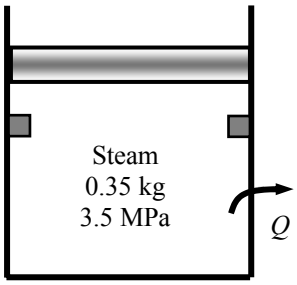
$$V_2 = \frac{\pi D^2}{4} L_2 \longrightarrow 0.001454\,\text{m}^3 = \frac{\pi (0.12\,\text{m})^2}{4} L_2 \longrightarrow L_2 = 0.1285\,\text{m}$$

$$\Delta L = L_1 - L_2 = 0.20 - 0.1285 = 0.07146\,\text{m} = \mathbf{7.1\,cm}$$

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4-149 A piston-cylinder device with a set of stops contains superheated steam. Heat is lost from the steam. The pressure and quality (if mixture), the boundary work, and the heat transfer until the piston first hits the stops and the total heat transfer are to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The friction between the piston and the cylinder is negligible.



Analysis (*a*) We take the steam in the cylinder to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U \quad \text{(since KE = PE = 0)}$$

Denoting when piston first hits the stops as state (2) and the final state as (3), the energy balance relations may be written as

$$W_{\text{b,in}} - Q_{\text{out,1-2}} = m(u_2 - u_1)$$
$$W_{\text{b,in}} - Q_{\text{out,1-3}} = m(u_3 - u_1)$$

The properties of steam at various states are (Tables A-4 through A-6)

$$T_{\text{sat@3.5 MPa}} = 242.56°\text{C}$$
$$T_1 = T_1 + \Delta T_{\text{sat}} = 242.56 + 7.4 = 250°\text{C}$$

$$\left.\begin{matrix} P_1 = 3.5\ \text{MPa} \\ T_1 = 250°\text{C} \end{matrix}\right\} \begin{matrix} v_1 = 0.05875\ \text{m}^3/\text{kg} \\ u_1 = 2623.9\ \text{kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = P_1 = 3.5\ \text{MPa} \\ x_2 = 0 \end{matrix}\right\} \begin{matrix} v_2 = 0.001235\ \text{m}^3/\text{kg} \\ u_2 = 1045.4\ \text{kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} v_3 = v_2 = 0.001235\ \text{m}^3/\text{kg} \\ T_3 = 200°\text{C} \end{matrix}\right\} \begin{matrix} x_3 = \mathbf{0.00062} \\ P_3 = \mathbf{1555\ kPa} \\ u_3 = 851.55\ \text{kJ/kg} \end{matrix}$$

(*b*) Noting that the pressure is constant until the piston hits the stops during which the boundary work is done, it can be determined from its definition as

$$W_{\text{b,in}} = mP_1(v_1 - v_2) = (0.35\ \text{kg})(3500\ \text{kPa})(0.05875 - 0.001235)\text{m}^3 = \mathbf{70.45\ kJ}$$

(*c*) Substituting into energy balance relations,

$$Q_{\text{out,1-2}} = 70.45\ \text{kJ} - (0.35\ \text{kg})(1045.4 - 2623.9)\text{kJ/kg} = \mathbf{622.9\ kJ}$$

(*d*) $$Q_{\text{out,1-3}} = 70.45\ \text{kJ} - (0.35\ \text{kg})(851.55 - 2623.9)\text{kJ/kg} = \mathbf{690.8\ kJ}$$

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4-150 An insulated rigid tank is divided into two compartments, each compartment containing the same ideal gas at different states. The two gases are allowed to mix. The simplest expression for the mixture temperature in a specified format is to be obtained.

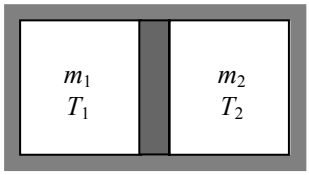
Analysis We take the both compartments together as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U \quad (\text{since } Q = W = \text{KE} = \text{PE} = 0)$$

$$0 = m_1 c_v (T_3 - T_1) + m_2 c_v (T_3 - T_2)$$

$$(m_1 + m_2) T_3 = m_1 T_1 + m_2 T_2$$



and $m_3 = m_1 + m_2$

Solving for final temperature, we find

$$T_3 = \frac{m_1}{m_3} T_1 + \frac{m_2}{m_3} T_2$$

4-151 A relation for the explosive energy of a fluid is given. A relation is to be obtained for the explosive energy of an ideal gas, and the value for air at a specified state is to be evaluated.

Properties The specific heat ratio for air at room temperature is $k = 1.4$.

Analysis The explosive energy per unit volume is given as

$$e_{\text{explosion}} = \frac{u_1 - u_2}{v_1}$$

For an ideal gas,

$$u_1 - u_2 = c_v (T_1 - T_2)$$

$$c_p - c_v = R$$

$$v_1 = \frac{RT_1}{P_1}$$

and thus

$$\frac{c_v}{R} = \frac{c_v}{c_p - c_v} = \frac{1}{c_p / c_v - 1} = \frac{1}{k-1}$$

Substituting,

$$e_{\text{explosion}} = \frac{c_v (T_1 - T_2)}{RT_1 / P_1} = \frac{P_1}{k-1}\left(1 - \frac{T_2}{T_1}\right)$$

which is the desired result.

Using the relation above, the total explosive energy of 20 m³ of air at 5 MPa and 100°C when the surroundings are at 20°C is determined to be

$$E_{\text{explosion}} = V e_{\text{explosion}} = \frac{P_1 V_1}{k-1}\left(1 - \frac{T_2}{T_1}\right) = \frac{(5000\ \text{kPa})(20\ \text{m}^3)}{1.4 - 1}\left(1 - \frac{293\ \text{K}}{373\ \text{K}}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa} \cdot \text{m}^3}\right) = \mathbf{53{,}619\ kJ}$$

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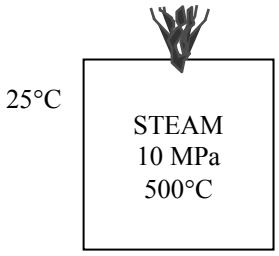
4-152 Using the relation for explosive energy given in the previous problem, the explosive energy of steam and its TNT equivalent at a specified state are to be determined.

Assumptions Steam condenses and becomes a liquid at room temperature after the explosion.

Properties The properties of steam at the initial and the final states are (Table A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.032811 \text{ m}^3/\text{kg} \\ u_1 = 3047.0 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ \text{Comp. liquid} \end{array}\right\} u_2 \cong u_{f@25°\text{C}} = 104.83 \text{ kJ/kg}$$



Analysis The mass of the steam is

$$m = \frac{V}{v_1} = \frac{20 \text{ m}^3}{0.032811 \text{ m}^3/\text{kg}} = 609.6 \text{ kg}$$

Then the total explosive energy of the steam is determined from

$$E_{\text{explosive}} = m(u_1 - u_2) = (609.6 \text{ kg})(3047.0 - 104.83)\text{kJ/kg} = \mathbf{1{,}793{,}436\ kJ}$$

which is equivalent to

$$\frac{1{,}793{,}436 \text{ kJ}}{3250 \text{ kJ/kg of TNT}} = \mathbf{551.8\ kg\ of\ TNT}$$

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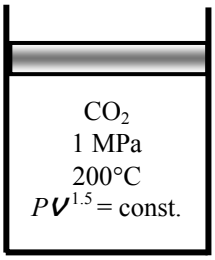
4-153 Carbon dioxide is compressed polytropically in a piston-cylinder device. The final temperature is to be determined treating the carbon dioxide as an ideal gas and a van der Waals gas.

Assumptions The process is quasi-equilibrium.

Properties The gas constant of carbon dioxide is $R = 0.1889$ kJ/kg·K (Table A-1).

Analysis (*a*) The initial specific volume is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.1889\,\text{kJ/kg}\cdot\text{K})(473\,\text{K})}{1000\,\text{kPa}} = 0.08935\,\text{m}^3/\text{kg}$$



From polytropic process expression,

$$v_2 = v_1\left(\frac{P_1}{P_2}\right)^{1/n} = (0.08935\,\text{m}^3/\text{kg})\left(\frac{1000}{3000}\right)^{1/1.5} = 0.04295\,\text{m}^3/\text{kg}$$

The final temperature is then

$$T_2 = \frac{P_2 v_2}{R} = \frac{(3000\,\text{kPa})(0.04295\,\text{m}^3/\text{kg})}{0.1889\,\text{kJ/kg}\cdot\text{K}} = \mathbf{682.1\,K}$$

(b) The van der Waals equation of state for carbon dioxide is

$$\left(P + \frac{365.8}{\bar{v}^2}\right)(\bar{v} - 0.0428) = R_u T$$

When this is applied to the initial state, the result is

$$\left(1000 + \frac{365.8}{\bar{v}_1^2}\right)(\bar{v}_1 - 0.0428) = (8.314)(473)$$

whose solution by iteration or by EES is

$$\bar{v}_1 = 3.882\,\text{m}^3/\text{kmol}$$

The final molar specific volume is then

$$\bar{v}_2 = \bar{v}_1\left(\frac{P_1}{P_2}\right)^{1/n} = (3.882\,\text{m}^3/\text{kmol})\left(\frac{1000}{3000}\right)^{1/1.5} = 1.866\,\text{m}^3/\text{kmol}$$

Substitution of the final molar specific volume into the van der Waals equation of state produces

$$T_2 = \frac{1}{R_u}\left(P + \frac{365.8}{\bar{v}^2}\right)(\bar{v} - 0.0428) = \frac{1}{8.314}\left(3000 + \frac{365.8}{(1.866)^2}\right)(1.866 - 0.0428) = \mathbf{680.9\,K}$$

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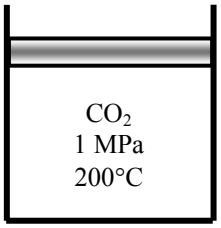
4-154 Carbon dioxide contained in a spring-loaded piston-cylinder device is compressed in a polytropic process. The final temperature is to be determined using the compressibility factor.

Properties The gas constant, the critical pressure, and the critical temperature of CO_2 are, from Table A-1,

$$R = 0.1889 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 304.2 \text{ K}, \qquad P_{cr} = 7.39 \text{ MPa}$$

Analysis From the compressibility chart at the initial state (Fig. A-15 or EES),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{473 \text{ K}}{304.2 \text{ K}} = 1.55 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{1 \text{ MPa}}{7.39 \text{ MPa}} = 0.135 \end{aligned}\right\} \; Z_1 = 0.991$$



The specific volume does not change during the process. Then,

$$v_1 = v_2 = \frac{Z_1 R T_1}{P_1} = \frac{(0.991)(0.1889 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})(473 \text{ K})}{1000 \text{ kPa}} = 0.08855 \text{ m}^3\text{/kg}$$

At the final state,

$$\left.\begin{aligned} P_{R2} &= \frac{P_2}{P_{cr}} = \frac{3 \text{ MPa}}{7.39 \text{ MPa}} = 0.406 \\ v_{R2} &= \frac{v_{2,\text{actual}}}{RT_{cr}/P_{cr}} = \frac{0.08855 \text{ m}^3\text{/kg}}{(0.1889 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})(304.2 \text{ K})/(7390 \text{ kPa})} = 11.4 \end{aligned}\right\} \; Z_2 = 1.0$$

Thus,

$$T_2 = \frac{P_2 v_2}{Z_2 R} = \frac{(3000 \text{ kPa})(0.08855 \text{ m}^3\text{/kg})}{(1.0)(0.1889 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})} = \mathbf{1406 \text{ K}}$$

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4-155E Two adiabatic tanks containing water at different states are connected by a valve. The valve is now opened, allowing the water vapor from the high-pressure tank to move to the low-pressure tank until the pressure in the two becomes equal. The final pressure and the final mass in each tank are to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no heat or work interactions involved

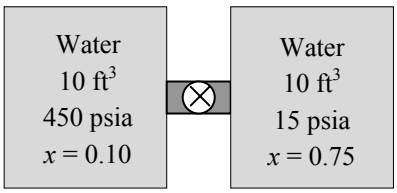
Analysis We take both tanks as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = U_2 - U_1 = 0$$

$$U_1 = U_2$$

$$m_{1,A}u_{1,A} + m_{1,B}u_{1,B} = m_{2,A}u_{2,A} + m_{2,B}u_{2,B}$$



where the high-pressure and low-pressure tanks are denoted by A and B, respectively. The specific volume in each tank is

$$\left.\begin{array}{l} P_{1,A} = 450 \text{ psia} \\ x_{1,A} = 0.10 \end{array}\right\} \begin{array}{l} v_{1,A} = v_f + xv_{fg} = 0.01955 + (0.10)(1.0324 - 0.01955) = 0.12084 \text{ ft}^3/\text{lbm} \\ u_{1,A} = u_f + xu_{fg} = 435.67 + (0.10)(683.52) = 504.02 \text{ Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_{1,B} = 15 \text{ psia} \\ x_{1,B} = 0.75 \end{array}\right\} \begin{array}{l} v_{1,B} = v_f + xv_{fg} = 0.01672 + (0.75)(26.297 - 0.01672) = 19.727 \text{ ft}^3/\text{lbm} \\ u_{1,B} = u_f + xu_{fg} = 181.16 + (0.75)(896.52) = 853.55 \text{ Btu/lbm} \end{array}$$

The total mass contained in the tanks is

$$m = \frac{V_A}{v_{1,A}} + \frac{V_B}{v_{1,B}} = \frac{10 \text{ ft}^3}{0.12084 \text{ ft}^3/\text{lbm}} + \frac{10 \text{ ft}^3}{19.727 \text{ ft}^3/\text{lbm}} = 82.75 + 0.5069 = 83.26 \text{ lbm}$$

Similarly, the initial total internal energy contained in both tanks is

$$U_1 = m_{1,A}u_{1,A} + m_{1,B}u_{1,B} = (82.75)(504.02) + (0.5069)(853.55) = 42{,}155 \text{ kJ}$$

The internal energy and the specific volume are

$$u_1 = u_2 = \frac{U_1}{m} = \frac{42{,}155 \text{ Btu}}{83.26 \text{ lbm}} = 506.3 \text{ Btu/lbm}$$

$$v_1 = v_2 = \frac{V}{m} = \frac{20 \text{ ft}^3}{83.26 \text{ lbm}} = 0.2402 \text{ ft}^3/\text{lbm}$$

Now, the final state is fixed. The pressure in this state may be obtained by iteration in water tables (Table A-5E). We used EES to get the exact result:

$$P_2 = \mathbf{313 \text{ psia}}$$

Since the specific volume is now the same in both tanks, and both tanks have the same volume, the mass is equally divided between the tanks at the end of this process,

$$m_{2,A} = m_{2,B} = \frac{m}{2} = \frac{83.26 \text{ lbm}}{2} = \mathbf{41.63 \text{ lbm}}$$

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Fundamentals of Engineering (FE) Exam Problems

4-156 A frictionless piston-cylinder device and a rigid tank contain 3 kmol of an ideal gas at the same temperature, pressure and volume. Now heat is transferred, and the temperature of both systems is raised by 10°C. The amount of extra heat that must be supplied to the gas in the cylinder that is maintained at constant pressure is

(a) 0 kJ (b) 27 kJ (c) 83 kJ (d) 249 kJ (e) 300 kJ

Answer (d) 249 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"Note that Cp-Cv=R, and thus Q_diff=m*R*dT=N*Ru*dT"
N=3 "kmol"
Ru=8.314 "kJ/kmol.K"
T_change=10
Q_diff=N*Ru*T_change

"Some Wrong Solutions with Common Mistakes:"
W1_Qdiff=0 "Assuming they are the same"
W2_Qdiff=Ru*T_change  "Not using mole numbers"
W3_Qdiff=Ru*T_change/N "Dividing by N instead of multiplying"
W4_Qdiff=N*Rair*T_change; Rair=0.287  "using R instead of Ru"
```

4-157 The specific heat of a material is given in a strange unit to be C = 3.60 kJ/kg.°F. The specific heat of this material in the SI units of kJ/kg.°C is

(a) 2.00 kJ/kg.°C (b) 3.20 kJ/kg.°C (c) 3.60 kJ/kg.°C (d) 4.80 kJ/kg.°C (e) 6.48 kJ/kg.°C

Answer (e) 6.48 kJ/kg.°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.60 "kJ/kg.F"
C_SI=C*1.8 "kJ/kg.C"

"Some Wrong Solutions with Common Mistakes:"
W1_C=C "Assuming they are the same"
W2_C=C/1.8 "Dividing by 1.8 instead of multiplying"
```

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4-158 A 3-m^3 rigid tank contains nitrogen gas at 500 kPa and 300 K. Now heat is transferred to the nitrogen in the tank and the pressure of nitrogen rises to 800 kPa. The work done during this process is

(a) 500 kJ (b) 1500 kJ (c) 0 kJ (d) 900 kJ (e) 2400 kJ

Answer (b) 0 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=3 "m^3"
P1=500 "kPa"
T1=300 "K"
P2=800 "kPa"
W=0 "since constant volume"

"Some Wrong Solutions with Common Mistakes:"
R=0.297
W1_W=V*(P2-P1) "Using W=V*DELTAP"
W2_W=V*P1
W3_W=V*P2
W4_W=R*T1*ln(P1/P2)
```

4-159 A 0.5-m^3 cylinder contains nitrogen gas at 600 kPa and 300 K. Now the gas is compressed isothermally to a volume of 0.1 m^3. The work done on the gas during this compression process is

(a) 720 kJ (b) 483 kJ (c) 240 kJ (d) 175 kJ (e) 143 kJ

Answer (b) 483 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=8.314/28
V1=0.5 "m^3"
V2=0.1 "m^3"
P1=600 "kPa"
T1=300 "K"
P1*V1=m*R*T1
W=m*R*T1* ln(V2/V1) "constant temperature"

"Some Wrong Solutions with Common Mistakes:"
W1_W=R*T1* ln(V2/V1)   "Forgetting m"
W2_W=P1*(V1-V2) "Using V*DeltaP"
P1*V1/T1=P2*V2/T1
W3_W=(V1-V2)*(P1+P2)/2 "Using P_ave*Delta V"
W4_W=P1*V1-P2*V2 "Using W=P1V1-P2V2"
```

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4-160 A well-sealed room contains 60 kg of air at 200 kPa and 25°C. Now solar energy enters the room at an average rate of 0.8 kJ/s while a 120-W fan is turned on to circulate the air in the room. If heat transfer through the walls is negligible, the air temperature in the room in 30 min will be

(a) 25.6°C (b) 49.8°C (c) 53.4°C (d) 52.5°C (e) 63.4°C

Answer (e) 63.4°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=60 "kg"
P1=200 "kPa"
T1=25 "C"
Qsol=0.8 "kJ/s"
time=30*60 "s"
Wfan=0.12 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*(Wfan+Qsol)=m*Cv*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
Cp=1.005 "kJ/kg.K"
time*(Wfan+Qsol)=m*Cp*(W1_T2-T1)  "Using Cp instead of Cv "
time*(-Wfan+Qsol)=m*Cv*(W2_T2-T1) "Subtracting Wfan instead of adding"
time*Qsol=m*Cv*(W3_T2-T1) "Ignoring Wfan"
time*(Wfan+Qsol)/60=m*Cv*(W4_T2-T1) "Using min for time instead of s"
```

4-161 A 2-kW baseboard electric resistance heater in a vacant room is turned on and kept on for 15 min. The mass of the air in the room is 75 kg, and the room is tightly sealed so that no air can leak in or out. The temperature rise of air at the end of 15 min is

(a) 8.5°C (b) 12.4°C (c) 24.0°C (d) 33.4°C (e) 54.8°C

Answer (d) 33.4°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=75 "kg"
time=15*60 "s"
W_e=2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*W_e=m*Cv*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
Cp=1.005 "kJ/kg.K"
time*W_e=m*Cp*W1_DELTAT "Using Cp instead of Cv"
time*W_e/60=m*Cv*W2_DELTAT "Using min for time instead of s"
```

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4-162 A room contains 75 kg of air at 100 kPa and 15°C. The room has a 250-W refrigerator (the refrigerator consumes 250 W of electricity when running), a 120-W TV, a 1.8-kW electric resistance heater, and a 50-W fan. During a cold winter day, it is observed that the refrigerator, the TV, the fan, and the electric resistance heater are running continuously but the air temperature in the room remains constant. The rate of heat loss from the room that day is

(a) 5832 kJ/h (b) 6192 kJ/h (c) 7560 kJ/h (d) 7632 kJ/h (e) 7992 kJ/h

Answer (e) 7992 kJ/h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=75 "kg"
P_1=100 "kPa"
T_1=15 "C"
time=30*60 "s"
W_ref=0.250 "kJ/s"
W_TV=0.120 "kJ/s"
W_heater=1.8 "kJ/s"
W_fan=0.05 "kJ/s"

"Applying energy balance E_in-E_out=dE_system gives E_out=E_in since T=constant and dE=0"
E_gain=W_ref+W_TV+W_heater+W_fan
Q_loss=E_gain*3600 "kJ/h"

"Some Wrong Solutions with Common Mistakes:"
E_gain1=-W_ref+W_TV+W_heater+W_fan "Subtracting Wrefrig instead of adding"
W1_Qloss=E_gain1*3600 "kJ/h"
E_gain2=W_ref+W_TV+W_heater-W_fan "Subtracting Wfan instead of adding"
W2_Qloss=E_gain2*3600 "kJ/h"
E_gain3=-W_ref+W_TV+W_heater-W_fan "Subtracting Wrefrig and Wfan instead of adding"
W3_Qloss=E_gain3*3600 "kJ/h"
E_gain4=W_ref+W_heater+W_fan "Ignoring the TV"
W4_Qloss=E_gain4*3600 "kJ/h"
```

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4-163 A piston-cylinder device contains 5 kg of air at 400 kPa and 30°C. During a quasi-equilibrium isothermal expansion process, 15 kJ of boundary work is done by the system, and 3 kJ of paddle-wheel work is done on the system. The heat transfer during this process is

(a) 12 kJ (b) 18 kJ (c) 2.4 kJ (d) 3.5 kJ (e) 60 kJ

Answer (a) 12 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=5 "kg"
P_1=400 "kPa"
T=30 "C"
Wout_b=15 "kJ"
Win_pw=3 "kJ"
"Noting that T=constant and thus dE_system=0, applying energy balance E_in-E_out=dE_system gives"
Q_in+Win_pw-Wout_b=0

"Some Wrong Solutions with Common Mistakes:"
W1_Qin=Q_in/Cv "Dividing by Cv"
W2_Qin=Win_pw+Wout_b "Adding both quantities"
W3_Qin=Win_pw "Setting it equal to paddle-wheel work"
W4_Qin=Wout_b "Setting it equal to boundaru work"
```

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4-164 A container equipped with a resistance heater and a mixer is initially filled with 3.6 kg of saturated water vapor at 120°C. Now the heater and the mixer are turned on; the steam is compressed, and there is heat loss to the surrounding air. At the end of the process, the temperature and pressure of steam in the container are measured to be 300°C and 0.5 MPa. The net energy transfer to the steam during this process is

(a) 274 kJ (b) 914 kJ (c) 1213 kJ (d) 988 kJ (e) 1291 kJ

Answer (d) 988 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=3.6 "kg"
T1=120 "C"
x1=1 "saturated vapor"
P2=500 "kPa"
T2=300 "C"
u1=INTENERGY(Steam_IAPWS,T=T1,x=x1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
"Noting that Eout=0 and dU_system=m*(u2-u1), applying energy balance E_in-E_out=dE_system gives"
E_out=0
E_in=m*(u2-u1)

"Some Wrong Solutions with Common Mistakes:"
Cp_steam=1.8723 "kJ/kg.K"
Cv_steam=1.4108 "kJ/kg.K"
W1_Ein=m*Cp_Steam*(T2-T1) "Assuming ideal gas and using Cp"
W2_Ein=m*Cv_steam*(T2-T1) "Assuming ideal gas and using Cv"
W3_Ein=u2-u1 "Not using mass"
h1=ENTHALPY(Steam_IAPWS,T=T1,x=x1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
W4_Ein=m*(h2-h1)  "Using enthalpy"
```

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4-165 A 6-pack canned drink is to be cooled from 18°C to 3°C. The mass of each canned drink is 0.355 kg. The drinks can be treated as water, and the energy stored in the aluminum can itself is negligible. The amount of heat transfer from the 6 canned drinks is

(a) 22 kJ (b) 32 kJ (c) 134 kJ (d) 187 kJ (e) 223 kJ

Answer (c) 134 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=6*0.355 "kg"
T1=18 "C"
T2=3 "C"
DELTAT=T2-T1 "C"
"Applying energy balance E_in-E_out=dE_system and noting that dU_system=m*C*DELTAT gives"
-Q_out=m*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
-W1_Qout=m*C*DELTAT/6 "Using one can only"
-W2_Qout=m*C*(T1+T2) "Adding temperatures instead of subtracting"
-W3_Qout=m*1.0*DELTAT  "Using specific heat of air or forgetting specific heat"
```

4-166 A glass of water with a mass of 0.45 kg at 20°C is to be cooled to 0°C by dropping ice cubes at 0°C into it. The latent heat of fusion of ice is 334 kJ/kg, and the specific heat of water is 4.18 kJ/kg.°C. The amount of ice that needs to be added is

(a) 56 g (b) 113 g (c) 124 g (d) 224 g (e) 450 g

Answer (b) 113 g

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
h_melting=334 "kJ/kg.K"
m_w=0.45 "kg"
T1=20 "C"
T2=0 "C"
DELTAT=T2-T1 "C"
"Noting that there is no energy transfer with the surroundings and the latent heat of melting of ice is
transferred form the water, and applying energy balance E_in-E_out=dE_system to ice+water gives"
dE_ice+dE_w=0
dE_ice=m_ice*h_melting
dE_w=m_w*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_mice*h_melting*(T1-T2)+m_w*C*DELTAT=0  "Multiplying h_latent by temperature difference"
W2_mice=m_w "taking mass of water to be equal to the mass of ice"
```

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4-167 A 2-kW electric resistance heater submerged in 5-kg water is turned on and kept on for 10 min. During the process, 300 kJ of heat is lost from the water. The temperature rise of water is

(a) 0.4°C (b) 43.1°C (c) 57.4°C (d) 71.8°C (e) 180.0°C

Answer (b) 43.1°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=5 "kg"
Q_loss=300 "kJ"
time=10*60 "s"
W_e=2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*W_e-Q_loss = dU_system
dU_system=m*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
time*W_e = m*C*W1_T "Ignoring heat loss"
time*W_e+Q_loss = m*C*W2_T  "Adding heat loss instead of subtracting"
time*W_e-Q_loss = m*1.0*W3_T "Using specific heat of air or not using specific heat"
```

4-168 1.5 kg of liquid water initially at 12°C is to be heated to 95°C in a teapot equipped with a 800 W electric heating element inside. The specific heat of water can be taken to be 4.18 kJ/kg.°C, and the heat loss from the water during heating can be neglected. The time it takes to heat the water to the desired temperature is

(a) 5.9 min (b) 7.3 min (c) 10.8 min (d) 14.0 min (e) 17.0 min

Answer (c) 10.8 min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=1.5 "kg"
T1=12 "C"
T2=95 "C"
Q_loss=0 "kJ"
W_e=0.8 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
(time*60)*W_e-Q_loss = dU_system  "time in minutes"
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_time*60*W_e-Q_loss = m*C*(T2+T1) "Adding temperatures instead of subtracting"
W2_time*60*W_e-Q_loss = C*(T2-T1) "Not using mass"
```

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4-169 An ordinary egg with a mass of 0.1 kg and a specific heat of 3.32 kJ/kg.°C is dropped into boiling water at 95°C. If the initial temperature of the egg is 5°C, the maximum amount of heat transfer to the egg is

(a) 12 kJ (b) 30 kJ (c) 24 kJ (d) 18 kJ (e) infinity

Answer (b) 30 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.32 "kJ/kg.K"
m=0.1 "kg"
T1=5 "C"
T2=95 "C"
"Applying energy balance E_in-E_out=dE_system gives"
E_in = dU_system
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_Ein = m*C*T2  "Using T2 only"
W2_Ein=m*(ENTHALPY(Steam_IAPWS,T=T2,x=1)-ENTHALPY(Steam_IAPWS,T=T2,x=0))  "Using h_fg"
```

4-170 An apple with an average mass of 0.18 kg and average specific heat of 3.65 kJ/kg.°C is cooled from 22°C to 5°C. The amount of heat transferred from the apple is

(a) 0.85 kJ (b) 62.1 kJ (c) 17.7 kJ (d) 11.2 kJ (e) 7.1 kJ

Answer (d) 11.2 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.65 "kJ/kg.K"
m=0.18 "kg"
T1=22 "C"
T2=5 "C"
"Applying energy balance E_in-E_out=dE_system gives"
-Q_out = dU_system
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
-W1_Qout =C*(T2-T1) "Not using mass"
-W2_Qout =m*C*(T2+T1) "adding temperatures"
```

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4-171 The specific heat at constant pressure for an ideal gas is given by $c_p = 0.9+(2.7\times10^{-4})T$ (kJ/kg · K) where T is in kelvin. The change in the enthalpy for this ideal gas undergoing a process in which the temperature changes from 27 to 47°C is most nearly

(a) 19.7 kJ/kg (b) 22.0 kJ/kg (c) 25.5 kJ/kg (d) 29.7 kJ/kg (e) 32.1 kJ/kg

Answer (a) 19.7 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=(27+273) [K]
T2=(47+273) [K]
"Performing the necessary integration, we obtain"
DELTAh=0.9*(T2-T1)+2.7E-4/2*(T2^2-T1^2)
```

4–172 The specific heat at constant volume for an ideal gas is given by $c_v = 0.7+(2.7\times10^{-4})T$ (kJ/kg · K) where T is in kelvin. The change in the enthalpy for this ideal gas undergoing a process in which the temperature changes from 27 to 127°C is most nearly

(a) 70 kJ/kg (b) 72.1 kJ/kg (c) 79.5 kJ/kg (d) 82.1 kJ/kg (e) 84.0 kJ/kg

Answer (c) 79.5 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=(27+273) [K]
T2=(127+273) [K]
"Performing the necessary integration, we obtain"
DELTAh=0.7*(T2-T1)+2.7E-4/2*(T2^2-T1^2)
```

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4-173 An ideal gas has a gas constant R = 0.3 kJ/kg·K and a constant-volume specific heat c_v = 0.7 kJ/kg·K. If the gas has a temperature change of 100°C, choose the correct answer for each of the following:

1. The change in enthalpy is, in kJ/kg

(*a*) 30 (*b*) 70 (*c*) 100 (*d*) insufficient information to determine

Answer (c) 100

2. The change in internal energy is, in kJ/kg

(*a*) 30 (*b*) 70 (*c*) 100 (*d*) insufficient information to determine

Answer (b) 70

3. The work done is, in kJ/kg

(*a*) 30 (*b*) 70 (*c*) 100 (*d*) insufficient information to determine

Answer (d) insufficient information to determine

4. The heat transfer is, in kJ/kg

(*a*) 30 (*b*) 70 (*c*) 100 (*d*) insufficient information to determine

Answer (d) insufficient information to determine

5. The change in the pressure-volume product is, in kJ/kg

(*a*) 30 (*b*) 70 (*c*) 100 (*d*) insufficient information to determine

Answer (a) 30

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.3 [kJ/kg-K]
c_v=0.7  [kJ/kg-K]
DELTAT=100 [K]

"(I)"
c_p=R+c_v
DELTAh=c_p*DELTAT
"(II)"
DELTAu=c_v*DELTAT
"(V)"
PV=R*DELTAT
```

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4-174 An ideal gas undergoes a constant temperature (isothermal) process in a closed system. The heat transfer and work are, respectively

(*a*) 0, $-c_v\Delta T$ (*b*) $c_v\Delta T$, 0 (*c*) $c_p\Delta T$, $R\Delta T$ (*d*) $R\ln(T_2/T_1)$, $R\ln(T_2/T_1)$

Answer (*d*) $R\ln(T_2/T_1)$, $R\ln(T_2/T_1)$

4-175 An ideal gas under goes a constant volume (isochoric) process in a closed system. The heat transfer and work are, respectively

(*a*) 0, $-c_v\Delta T$ (*b*) $c_v\Delta T$, 0 (*c*) $c_p\Delta T$, $R\Delta T$ (*d*) $R\ln(T_2/T_1)$, $R\ln(T_2/T_1)$

Answer (*b*) $c_v\Delta T$, 0

4-176 An ideal gas under goes a constant pressure (isobaric) process in a closed system. The heat transfer and work are, respectively

(*a*) 0, $-c_v\Delta T$ (*b*) $c_v\Delta T$, 0 (*c*) $c_p\Delta T$, $R\Delta T$ (*d*) $R\ln(T_2/T_1)$, $R\ln(T_2/T_1)$

Answer (*c*) $c_p\Delta T$, $R\Delta T$

4-177 An ideal gas under goes a constant entropy (isentropic) process in a closed system. The heat transfer and work are, respectively

(*a*) 0, $-c_v\Delta T$ (*b*) $c_v\Delta T$, 0 (*c*) $c_p\Delta T$, $R\Delta T$ (*d*) $R\ln(T_2/T_1)$, $R\ln(T_2/T_1)$

Answer (*a*) 0, $-c_v\Delta T$

4-178 … 4-183 Design and Essay Problems

4-182 A claim that fruits and vegetables are cooled by 6°C for each percentage point of weight loss as moisture during vacuum cooling is to be evaluated.

Analysis Assuming the fruits and vegetables are cooled from 30ºC and 0ºC, the average heat of vaporization can be taken to be 2466 kJ/kg, which is the value at 15ºC, and the specific heat of products can be taken to be 4 kJ/kg.ºC. Then the vaporization of 0.01 kg water will lower the temperature of 1 kg of produce by 24.66/4 = 6ºC. Therefore, the vacuum cooled products will lose 1 percent moisture for each 6ºC drop in temperature. Thus the claim is **reasonable**.



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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 5

MASS AND ENERGY ANALYSIS OF CONTROL VOLUMES

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Conservation of Mass

5-1C Mass flow rate is the amount of mass flowing through a cross-section per unit time whereas the volume flow rate is the amount of volume flowing through a cross-section per unit time.

5-2C Flow through a control volume is steady when it involves no changes with time at any specified position.

5-3C The amount of mass or energy entering a control volume does not have to be equal to the amount of mass or energy leaving during an unsteady-flow process.

5-4C No, a flow with the same volume flow rate at the inlet and the exit is not necessarily steady (unless the density is constant). To be steady, the mass flow rate through the device must remain constant.

5-5E A pneumatic accumulator arranged to maintain a constant pressure as air enters or leaves is considered. The amount of air added is to be determined.

***Assumptions* 1** Air is an ideal gas.

Properties The gas constant of air is $R = 0.3704$ psia·ft^3/lbm·R (Table A-1E).

Analysis At the beginning of the filling, the mass of the air in the container is

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(200\ \text{psia})(0.2\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(80+460\ \text{R})} = 0.200\ \text{lbm}$$

During the process both pressure and temperature remain constant while volume increases by 5 times. Thus,

$$m_2 = \frac{P_2 V_2}{RT_2} = 5m_1 = 5(0.200) = 1.00\ \text{lbm}$$

The amount of air added to the container is then

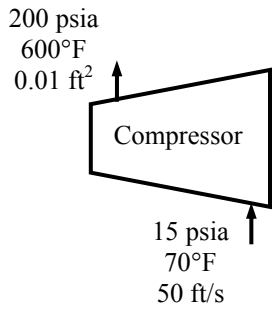
$$\Delta m = m_2 - m_1 = 1.00 - 0.200 = \mathbf{0.8\ lbm}$$

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5-6E Helium at a specified state is compressed to another specified state. The mass flow rate and the inlet area are to be determined.

Assumptions Flow through the compressor is steady.

Properties The gas cosntant of helium is $R = 2.6809$ psia·ft^3/lbm·R (Table A-1E)



Analysis The mass flow rate is determined from

$$\dot{m} = \frac{A_2 V_2}{\nu_2} = \frac{A_2 V_2 P_2}{RT_2} = \frac{(0.01\,\text{ft}^2)(100\,\text{ft/s})(200\,\text{psia})}{(2.6809\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1060\,\text{R})} = \mathbf{0.07038\ lbm/s}$$

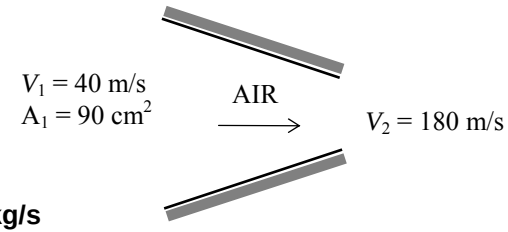
The inlet area is determined from

$$A_1 = \frac{\dot{m}\nu_1}{V_1} = \frac{\dot{m}RT_1}{V_1 P_1} = \frac{(0.07038\,\text{lbm/s})(2.6809\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530\,\text{R})}{(50\,\text{ft/s})(15\,\text{psia})} = \mathbf{0.1333\ ft^2}$$

5-7 Air is accelerated in a nozzle. The mass flow rate and the exit area of the nozzle are to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 2.21 kg/m^3 at the inlet, and 0.762 kg/m^3 at the exit.



Analysis (a) The mass flow rate of air is determined from the inlet conditions to be

$$\dot{m} = \rho_1 A_1 V_1 = (2.21\,\text{kg/m}^3)(0.009\,\text{m}^2)(40\,\text{m/s}) = \mathbf{0.796\ kg/s}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit area of the nozzle is determined to be

$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.796\,\text{kg/s}}{(0.762\,\text{kg/m}^3)(180\,\text{m/s})} = 0.0058\,\text{m}^2 = \mathbf{58\ cm^2}$$

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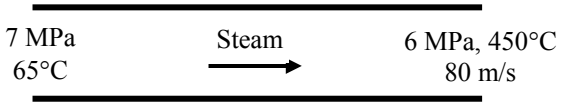
5-8 Water flows through the tubes of a boiler. The velocity and volume flow rate of the water at the inlet are to be determined.

Assumptions Flow through the boiler is steady.

Properties The specific volumes of water at the inlet and exit are (Tables A-6 and A-7)

$$\left.\begin{matrix} P_1 = 7\,\text{MPa} \\ T_1 = 65°\text{C} \end{matrix}\right\} \nu_1 = 0.001017\,\text{m}^3/\text{kg}$$

$$\left.\begin{matrix} P_2 = 6\,\text{MPa} \\ T_2 = 450°\text{C} \end{matrix}\right\} \nu_2 = 0.05217\,\text{m}^3/\text{kg}$$



Analysis The cross-sectional area of the tube is

$$A_c = \frac{\pi D^2}{4} = \frac{\pi(0.13\,\text{m})^2}{4} = 0.01327\,\text{m}^2$$

The mass flow rate through the tube is same at the inlet and exit. It may be determined from exit data to be

$$\dot{m} = \frac{A_c V_2}{\nu_2} = \frac{(0.01327\,\text{m}^2)(80\,\text{m/s})}{0.05217\,\text{m}^3/\text{kg}} = 20.35\,\text{kg/s}$$

The water velocity at the inlet is then

$$V_1 = \frac{\dot{m}\nu_1}{A_c} = \frac{(20.35\,\text{kg/s})(0.001017\,\text{m}^3/\text{kg})}{0.01327\,\text{m}^2} = \mathbf{1.560\,m/s}$$

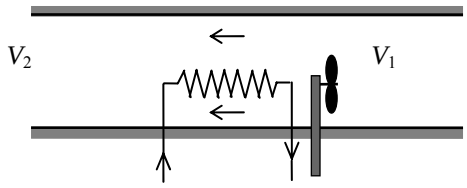
The volumetric flow rate at the inlet is

$$\dot{V}_1 = A_c V_1 = (0.01327\,\text{m}^2)(1.560\,\text{m/s}) = \mathbf{0.0207\,m^3/s}$$

5-9 Air is expanded and is accelerated as it is heated by a hair dryer of constant diameter. The percent increase in the velocity of air as it flows through the drier is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 1.20 kg/m^3 at the inlet, and 1.05 kg/m^3 at the exit.



Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A V_1 = \rho_2 A V_2$$

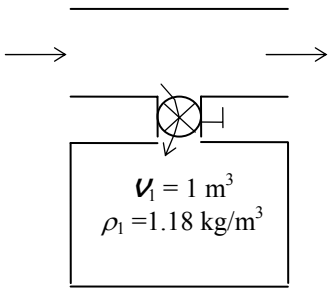
$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20\,\text{kg/m}^3}{0.95\,\text{kg/m}^3} = 1.263 \quad \text{(or, and increase of } \mathbf{26.3\%}\text{)}$$

Therefore, the air velocity increases 26.3% as it flows through the hair drier.

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5-10 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until the density rises to a specified level. The mass of air that entered the tank is to be determined.

Properties The density of air is given to be 1.18 kg/m^3 at the beginning, and 7.20 kg/m^3 at the end.



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. The mass balance for this system can be expressed as

Mass balance:

$$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1 = \rho_2 V - \rho_1 V$$

Substituting,

$$m_i = (\rho_2 - \rho_1)V = [(7.20 - 1.18)\,\text{kg/m}^3](1\,\text{m}^3) = \mathbf{6.02\,kg}$$

Therefore, 6.02 kg of mass entered the tank.

5-11 A cyclone separator is used to remove fine solid particles that are suspended in a gas stream. The mass flow rates at the two outlets and the amount of fly ash collected per year are to be determined.

Assumptions Flow through the separator is steady.

Analysis Since the ash particles cannot be converted into the gas and vice-versa, the mass flow rate of ash into the control volume must equal that going out, and the mass flow rate of flue gas into the control volume must equal that going out. Hence, the mass flow rate of ash leaving is

$$\dot{m}_{\text{ash}} = y_{\text{ash}}\dot{m}_{\text{in}} = (0.001)(10\,\text{kg/s}) = \mathbf{0.01\,kg/s}$$

The mass flow rate of flue gas leaving the separator is then

$$\dot{m}_{\text{flue gas}} = \dot{m}_{\text{in}} - \dot{m}_{\text{ash}} = 10 - 0.01 = \mathbf{9.99\,kg/s}$$

The amount of fly ash collected per year is

$$m_{\text{ash}} = \dot{m}_{\text{ash}}\Delta t = (0.01\,\text{kg/s})(365 \times 24 \times 3600\,\text{s/year}) = \mathbf{315{,}400\ kg/year}$$

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5-12 Air flows through an aircraft engine. The volume flow rate at the inlet and the mass flow rate at the exit are to be determined.

Assumptions **1** Air is an ideal gas. **2** The flow is steady.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1).

Analysis The inlet volume flow rate is

$$\dot{V}_1 = A_1 V_1 = (1\,\text{m}^2)(180\,\text{m/s}) = \mathbf{180\ m^3/s}$$

The specific volume at the inlet is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\,\text{K})}{100\,\text{kPa}} = 0.8409\,\text{m}^3/\text{kg}$$

Since the flow is steady, the mass flow rate remains constant during the flow. Then,

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{180\,\text{m}^3/\text{s}}{0.8409\,\text{m}^3/\text{kg}} = \mathbf{214.1\,kg/s}$$

5-13 A spherical hot-air balloon is considered. The time it takes to inflate the balloon is to be determined.

Assumptions **1** Air is an ideal gas.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1).

Analysis The specific volume of air entering the balloon is

$$v = \frac{RT}{P} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\,\text{K})}{120\,\text{kPa}} = 0.7008\,\text{m}^3/\text{kg}$$

The mass flow rate at this entrance is

$$\dot{m} = \frac{A_c V}{v} = \frac{\pi D^2}{4}\frac{V}{v} = \frac{\pi(1.0\,\text{m})^2}{4}\frac{3\,\text{m/s}}{0.7008\,\text{m}^3/\text{kg}} = 3.362\,\text{kg/s}$$

The initial mass of the air in the balloon is

$$m_i = \frac{V_i}{v} = \frac{\pi D^3}{6v} = \frac{\pi(5\,\text{m})^3}{6(0.7008\,\text{m}^3/\text{kg})} = 93.39\,\text{kg}$$

Similarly, the final mass of air in the balloon is

$$m_f = \frac{V_f}{v} = \frac{\pi D^3}{6v} = \frac{\pi(15\,\text{m})^3}{6(0.7008\,\text{m}^3/\text{kg})} = 2522\,\text{kg}$$

The time it takes to inflate the balloon is determined from

$$\Delta t = \frac{m_f - m_i}{\dot{m}} = \frac{(2522 - 93.39)\,\text{kg}}{3.362\,\text{kg/s}} = 722\,\text{s} = \mathbf{12.0\ min}$$

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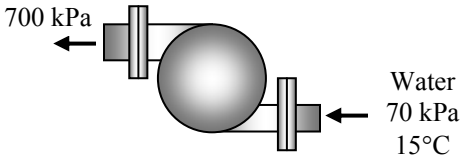
5-14 A water pump increases water pressure. The diameters of the inlet and exit openings are given. The velocity of the water at the inlet and outlet are to be determined.

Assumptions **1** Flow through the pump is steady. **2** The specific volume remains constant.

Properties The inlet state of water is compressed liquid. We approximate it as a saturated liquid at the given temperature. Then, at 15°C and 40°C, we have (Table A-4)

$$\left.\begin{matrix} T = 15°\text{C} \\ x = 0 \end{matrix}\right\} \nu_1 = 0.001001\ \text{m}^3/\text{kg}$$

$$\left.\begin{matrix} T = 40°\text{C} \\ x = 0 \end{matrix}\right\} \nu_1 = 0.001008\ \text{m}^3/\text{kg}$$



Analysis The velocity of the water at the inlet is

$$V_1 = \frac{\dot{m}\nu_1}{A_1} = \frac{4\dot{m}\nu_1}{\pi D_1^2} = \frac{4(0.5\ \text{kg/s})(0.001001\ \text{m}^3/\text{kg})}{\pi(0.01\ \text{m})^2} = \mathbf{6.37\ m/s}$$

Since the mass flow rate and the specific volume remains constant, the velocity at the pump exit is

$$V_2 = V_1 \frac{A_1}{A_2} = V_1\left(\frac{D_1}{D_2}\right)^2 = (6.37\ \text{m/s})\left(\frac{0.01\ \text{m}}{0.015\ \text{m}}\right)^2 = \mathbf{2.83\ m/s}$$

Using the specific volume at 40°C, the water velocity at the inlet becomes

$$V_1 = \frac{\dot{m}\nu_1}{A_1} = \frac{4\dot{m}\nu_1}{\pi D_1^2} = \frac{4(0.5\ \text{kg/s})(0.001008\ \text{m}^3/\text{kg})}{\pi(0.01\ \text{m})^2} = \mathbf{6.42\ m/s}$$

which is a 0.8% increase in velocity.

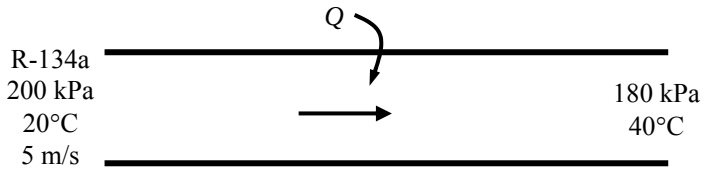
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5-15 Refrigerant-134a flows through a pipe. Heat is supplied to R-134a. The volume flow rates of air at the inlet and exit, the mass flow rate, and the velocity at the exit are to be determined.

Properties The specific volumes of R-134a at the inlet and exit are (Table A-13)

$$\left.\begin{matrix} P_1 = 200\ \text{kPa} \\ T_1 = 20°\text{C} \end{matrix}\right\} \nu_1 = 0.1142\ \text{m}^3/\text{kg} \qquad \left.\begin{matrix} P_1 = 180\ \text{kPa} \\ T_1 = 40°\text{C} \end{matrix}\right\} \nu_2 = 0.1374\ \text{m}^3/\text{kg}$$

Analysis



(*a*) (*b*) The volume flow rate at the inlet and the mass flow rate are

$$\dot{V}_1 = A_c V_1 = \frac{\pi D^2}{4} V_1 = \frac{\pi (0.28\ \text{m})^2}{4} (5\ \text{m/s}) = \mathbf{0.3079\ m^3/s}$$

$$\dot{m} = \frac{1}{\nu_1} A_c V_1 = \frac{1}{\nu_1} \frac{\pi D^2}{4} V_1 = \frac{1}{0.1142\ \text{m}^3/\text{kg}} \frac{\pi (0.28\ \text{m})^2}{4} (5\ \text{m/s}) = \mathbf{2.696\ kg/s}$$

(c) Noting that mass flow rate is constant, the volume flow rate and the velocity at the exit of the pipe are determined from

$$\dot{V}_2 = \dot{m} \nu_2 = (2.696\ \text{kg/s})(0.1374\ \text{m}^3/\text{kg}) = \mathbf{0.3705\ m^3/s}$$

$$V_2 = \frac{\dot{V}_2}{A_c} = \frac{0.3705\ \text{m}^3/\text{s}}{\frac{\pi (0.28\ \text{m})^2}{4}} = \mathbf{6.02\ m/s}$$

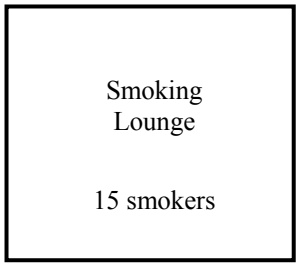
5-16 A smoking lounge that can accommodate 15 smokers is considered. The required minimum flow rate of air that needs to be supplied to the lounge and the diameter of the duct are to be determined.

Assumptions Infiltration of air into the smoking lounge is negligible.

Properties The minimum fresh air requirements for a smoking lounge is given to be 30 L/s per person.

Analysis The required minimum flow rate of air that needs to be supplied to the lounge is determined directly from

$$\dot{V}_{\text{air}} = \dot{V}_{\text{air per person}} (\text{No. of persons})$$
$$= (30\ \text{L/s} \cdot \text{person})(15\ \text{persons}) = 450\ \text{L/s} = \mathbf{0.45\ m^3/s}$$



The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2 / 4)$$

Solving for the diameter *D* and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.45\ \text{m}^3/\text{s})}{\pi (8\ \text{m/s})}} = \mathbf{0.268\ m}$$

Therefore, the diameter of the fresh air duct should be at least 26.8 cm if the velocity of air is not to exceed 8 m/s.

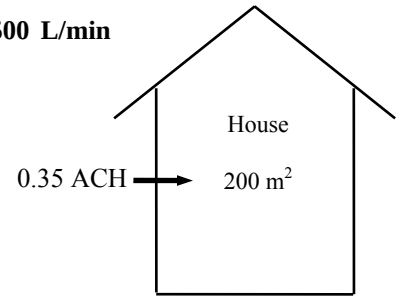
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5-17 The minimum fresh air requirements of a residential building is specified to be 0.35 air changes per hour. The size of the fan that needs to be installed and the diameter of the duct are to be determined.

Analysis The volume of the building and the required minimum volume flow rate of fresh air are

$$\boldsymbol{V}_{\text{room}} = (3.0\text{ m})(200\text{ m}^2) = 600\text{ m}^3$$

$$\dot{\boldsymbol{V}} = \boldsymbol{V}_{\text{room}} \times \text{ACH} = (600\text{ m}^3)(0.35/\text{h}) = 210\text{ m}^3/\text{h} = 210{,}000\text{ L/h} = \mathbf{3500\ L/min}$$



The volume flow rate of fresh air can be expressed as

$$\dot{\boldsymbol{V}} = VA = V(\pi D^2/4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(210/3600\text{ m}^3/\text{s})}{\pi(4\text{ m/s})}} = \mathbf{0.136\ m}$$

Therefore, the diameter of the fresh air duct should be at least 13.6 cm if the velocity of air is not to exceed 4 m/s.

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Flow Work and Energy Transfer by Mass

5-18C Energy can be transferred to or from a control volume as heat, various forms of work, and by mass.

5-19C Flow energy or flow work is the energy needed to push a fluid into or out of a control volume. Fluids at rest do not possess any flow energy.

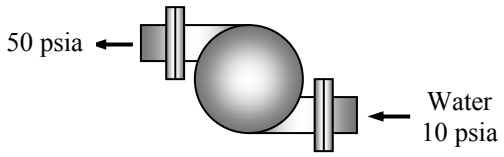
5-20C Flowing fluids possess flow energy in addition to the forms of energy a fluid at rest possesses. The total energy of a fluid at rest consists of internal, kinetic, and potential energies. The total energy of a flowing fluid consists of internal, kinetic, potential, and flow energies.

5-21E A water pump increases water pressure. The flow work required by the pump is to be determined.

Assumptions **1** Flow through the pump is steady. **2** The state of water at the pump inlet is saturated liquid. **3** The specific volume remains constant.

Properties The specific volume of saturated liquid water at 10 psia is

$$\nu = \nu_{f@10\,\text{psia}} = 0.01659\ \text{ft}^3/\text{lbm} \quad \text{(Table A-5E)}$$



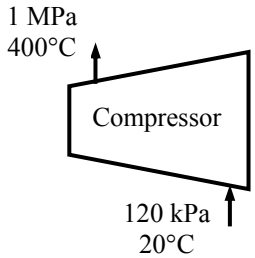
Then the flow work relation gives

$$\begin{aligned} w_{\text{flow}} &= P_2\nu_2 - P_1\nu_1 = \nu(P_2 - P_1) \\ &= (0.01659\ \text{ft}^3/\text{lbm})(50-10)\text{psia}\left(\frac{1\,\text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &= \mathbf{0.1228\ Btu/lbm} \end{aligned}$$

5-22 An air compressor compresses air. The flow work required by the compressor is to be determined.

Assumptions **1** Flow through the compressor is steady. **2** Air is an ideal gas.

Properties The gas constant of air is $R = 0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1).



Analysis Combining the flow work expression with the ideal gas equation of state gives

$$\begin{aligned} w_{\text{flow}} &= P_2\nu_2 - P_1\nu_1 \\ &= R(T_2 - T_1) \\ &= (0.287\ \text{kJ/kg}\cdot\text{K})(400-20)\text{K} \\ &= \mathbf{109\ kJ/kg} \end{aligned}$$

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5-23E Steam is leaving a pressure cooker at a specified pressure. The velocity, flow rate, the total and flow energies, and the rate of energy transfer by mass are to be determined.

Assumptions **1** The flow is steady, and the initial start-up period is disregarded. **2** The kinetic and potential energies are negligible, and thus they are not considered. **3** Saturation conditions exist within the cooker at all times so that steam leaves the cooker as a saturated vapor at 20 psia.

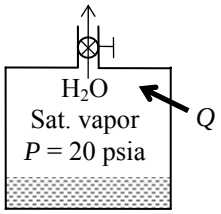
Properties The properties of saturated liquid water and water vapor at 20 psia are $\nu_f = 0.01683$ ft^3/lbm, $\nu_g = 20.093$ ft^3/lbm, $u_g = 1081.8$ Btu/lbm, and $h_g = 1156.2$ Btu/lbm (Table A-5E).

Analysis (*a*) Saturation conditions exist in a pressure cooker at all times after the steady operating conditions are established. Therefore, the liquid has the properties of saturated liquid and the exiting steam has the properties of saturated vapor at the operating pressure. The amount of liquid that has evaporated, the mass flow rate of the exiting steam, and the exit velocity are

$$m = \frac{\Delta V_{\text{liquid}}}{\nu_f} = \frac{0.6\text{ gal}}{0.01683\text{ ft}^3/\text{lbm}}\left(\frac{0.13368\text{ ft}^3}{1\text{ gal}}\right) = 4.766\text{ lbm}$$

$$\dot{m} = \frac{m}{\Delta t} = \frac{4.766\text{ lbm}}{45\text{ min}} = 0.1059\text{ lbm/min} = \mathbf{1.765 \times 10^{-3}\ lbm/s}$$

$$V = \frac{\dot{m}}{\rho_g A_c} = \frac{\dot{m}\nu_g}{A_c} = \frac{(1.765\times10^{-3}\text{ lbm/s})(20.093\text{ ft}^3/\text{lbm})}{0.15\text{ in}^2}\left(\frac{144\text{ in}^2}{1\text{ ft}^2}\right) = \mathbf{34.1\ ft/s}$$



(*b*) Noting that $h = u + P\nu$ and that the kinetic and potential energies are disregarded, the flow and total energies of the exiting steam are

$$e_{\text{flow}} = P\nu = h - u = 1156.2 - 1081.8 = \mathbf{74.4\ Btu/lbm}$$

$$\theta = h + ke + pe \cong h = \mathbf{1156.2\ Btu/lbm}$$

Note that the kinetic energy in this case is ke = $V^2/2$ = (34.1 ft/s)^2 /2 = 581 ft^2/s^2 = 0.0232 Btu/lbm, which is very small compared to enthalpy.

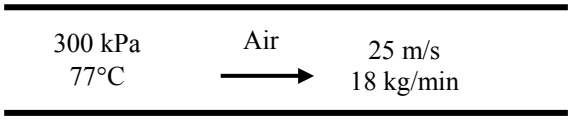
(*c*) The rate at which energy is leaving the cooker by mass is simply the product of the mass flow rate and the total energy of the exiting steam per unit mass,

$$\dot{E}_{\text{mass}} = \dot{m}\theta = (1.765\times10^{-3}\text{ lbm/s})(1156.2\text{ Btu/lbm}) = \mathbf{2.04\ Btu/s}$$

Discussion The numerical value of the energy leaving the cooker with steam alone does not mean much since this value depends on the reference point selected for enthalpy (it could even be negative). The significant quantity is the difference between the enthalpies of the exiting vapor and the liquid inside (which is h_{fg}) since it relates directly to the amount of energy supplied to the cooker.

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5-24 Air flows steadily in a pipe at a specified state. The diameter of the pipe, the rate of flow energy, and the rate of energy transport by mass are to be determined. Also, the error involved in the determination of energy transport by mass is to be determined.



Properties The properties of air are R = 0.287 kJ/kg.K and c_p = 1.008 kJ/kg.K (at 350 K from Table A-2b)

Analysis (*a*) The diameter is determined as follows

$$\nu = \frac{RT}{P} = \frac{(0.287\ \text{kJ/kg.K})(77+273\ \text{K})}{(300\ \text{kPa})} = 0.3349\ \text{m}^3/\text{kg}$$

$$A = \frac{\dot{m}\nu}{V} = \frac{(18/60\ \text{kg/s})(0.3349\ \text{m}^3/\text{kg})}{25\ \text{m/s}} = 0.004018\ \text{m}^2$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(0.004018\ \text{m}^2)}{\pi}} = \mathbf{0.0715\ m}$$

(*b*) The rate of flow energy is determined from

$$\dot{W}_{\text{flow}} = \dot{m}P\nu = (18/60\ \text{kg/s})(300\ \text{kPa})(0.3349\ \text{m}^3/\text{kg}) = \mathbf{30.14\ kW}$$

(*c*) The rate of energy transport by mass is

$$\begin{aligned}\dot{E}_{\text{mass}} &= \dot{m}(h+ke) = \dot{m}\left(c_pT + \frac{1}{2}V^2\right) \\ &= (18/60\ \text{kg/s})\left[(1.008\ \text{kJ/kg.K})(77+273\ \text{K}) + \frac{1}{2}(25\ \text{m/s})^2\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right] \\ &= \mathbf{105.94\ kW}\end{aligned}$$

(*d*) If we neglect kinetic energy in the calculation of energy transport by mass

$$\dot{E}_{\text{mass}} = \dot{m}h = \dot{m}c_pT = (18/60\ \text{kg/s})(1.005\ \text{kJ/kg.K})(77+273\ \text{K}) = 105.84\ \text{kW}$$

Therefore, the error involved if neglect the kinetic energy is only **0.09%**.

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Steady Flow Energy Balance: Nozzles and Diffusers

5-25C No.

5-26C It is mostly converted to internal energy as shown by a rise in the fluid temperature.

5-27C The kinetic energy of a fluid increases at the expense of the internal energy as evidenced by a decrease in the fluid temperature.

5-28C Heat transfer to the fluid as it flows through a nozzle is desirable since it will probably increase the kinetic energy of the fluid. Heat transfer from the fluid will decrease the exit velocity.

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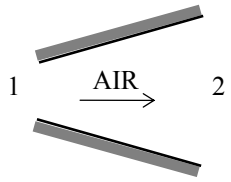
5-29 Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpy of air at the inlet temperature of 400 K is h_1 = 400.98 kJ/kg (Table A-17).

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \quad ,$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 400.98 \text{ kJ/kg} - \frac{(30 \text{ m/s})^2 - (230 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 426.98 \text{ kJ/kg}$$

From Table A-17,

T_2 = **425.6 K**

(*b*) The specific volume of air at the diffuser exit is

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(425.6 \text{ K})}{(100 \text{ kPa})} = 1.221 \text{ m}^3/\text{kg}$$

From conservation of mass,

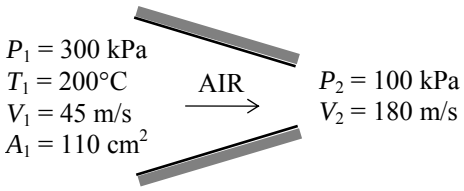
$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}\nu_2}{V_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = \mathbf{0.0678\ m^2}$$

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5-30 Air is accelerated in a nozzle from 45 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is c_p = 1.02 kJ/kg.°C (Table A-2).



Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\ \text{K})}{300\ \text{kPa}} = 0.4525\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 = \frac{1}{0.4525\ \text{m}^3/\text{kg}}(0.0110\ \text{m}^2)(45\ \text{m/s}) = \mathbf{1.094\ kg/s}$$

(*b*) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\nearrow 0\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,ave}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$0 = (1.02\ \text{kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) + \frac{(180\ \text{m/s})^2 - (45\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

It yields

$$T_2 = \mathbf{185.2°C}$$

(*c*) The specific volume of air at the nozzle exit is

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(185.2 + 273\ \text{K})}{100\ \text{kPa}} = 1.315\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow 1.094\ \text{kg/s} = \frac{1}{1.315\ \text{m}^3/\text{kg}} A_2 (180\ \text{m/s}) \rightarrow A_2 = 0.00799\ \text{m}^2 = \mathbf{79.9\ cm^2}$$

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5-31 Problem 5-30 is reconsidered. The effect of the inlet area on the mass flow rate, exit velocity, and the exit area as the inlet area varies from 50 cm^2 to 150 cm^2 is to be investigated, and the final results are to be plotted against the inlet area.

Analysis The problem is solved using EES, and the solution is given below.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'Air' = WorkFluid$  then
       HCal:=ENTHALPY(Air,T=Tx) "Ideal gas equ."
   else
      HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"
"Property relation: Air is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"
"Knowns - obtain from the input diagram"
WorkFluid$ = 'Air'
T[1] = 200  [C]
P[1] = 300   [kPa]
Vel[1] = 45  [m/s]
P[2] = 100  [kPa]
Vel[2] = 180  [m/s]
A[1]=110 [cm^2]
Am[1]=A[1]*convert(cm^2,m^2)

"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

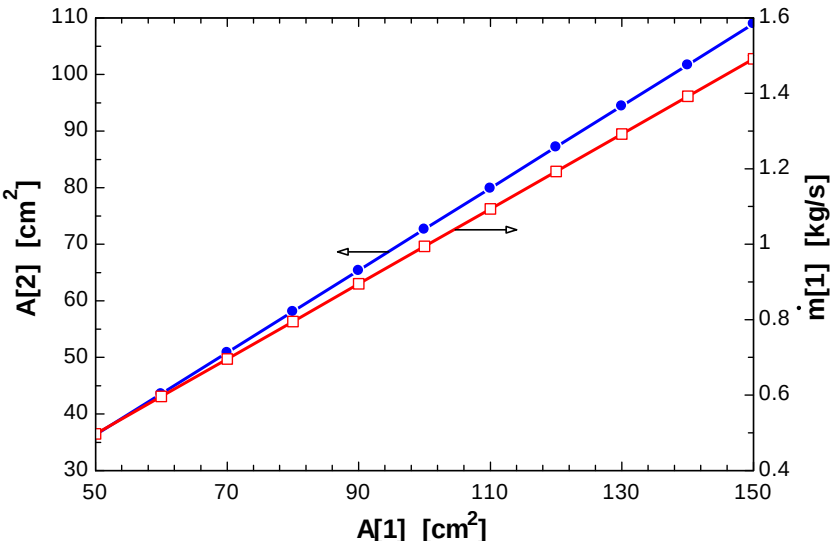
"Conservation of mass: "
m_dot[1]= m_dot[2]
"Mass flow rate"
m_dot[1]=Am[1]*Vel[1]/v[1]
m_dot[2]= Am[2]*Vel[2]/v[2]

"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)

"Definition"
A_ratio=A[1]/A[2]
A[2]=Am[2]*convert(m^2,cm^2)
```

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A_1 [cm^2]	A_2 [cm^2]	m_1 [kg/s]	T_2 [C]
50	36.32	0.497	185.2
60	43.59	0.5964	185.2
70	50.85	0.6958	185.2
80	58.12	0.7952	185.2
90	65.38	0.8946	185.2
100	72.65	0.9941	185.2
110	79.91	1.093	185.2
120	87.18	1.193	185.2
130	94.44	1.292	185.2
140	101.7	1.392	185.2
150	109	1.491	185.2



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5-32E Air is accelerated in an adiabatic nozzle. The velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The nozzle is adiabatic.

Properties The specific heat of air at the average temperature of (700+645)/2=672.5°F is c_p = 0.253 Btu/lbm·R (Table A-2E*b*).

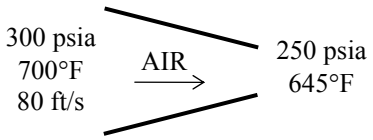
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$



Solving for exit velocity,

$$V_2 = \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} = \left[V_1^2 + 2c_p(T_1 - T_2)\right]^{0.5}$$

$$= \left[(80 \text{ ft/s})^2 + 2(0.253 \text{ Btu/lbm} \cdot \text{R})(700 - 645)\text{R}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)\right]^{0.5}$$

$$= \mathbf{838.6 \text{ ft/s}}$$

5-33 Air is decelerated in an adiabatic diffuser. The velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The diffuser is adiabatic.

Properties The specific heat of air at the average temperature of (20+90)/2=55°C =328 K is c_p = 1.007 kJ/kg·K (Table A-2*b*).

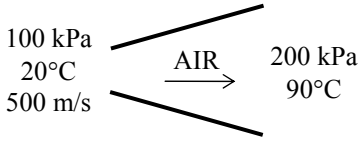
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$



Solving for exit velocity,

$$V_2 = \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} = \left[V_1^2 + 2c_p(T_1 - T_2)\right]^{0.5}$$

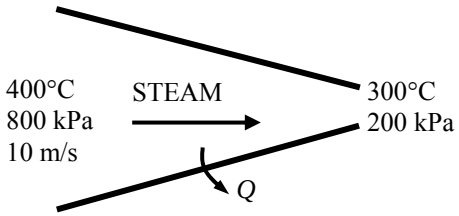
$$= \left[(500 \text{ m/s})^2 + 2(1.007 \text{ kJ/kg} \cdot \text{K})(20 - 90)\text{K}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)\right]^{0.5}$$

$$= \mathbf{330.2 \text{ m/s}}$$

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5-34 Heat is lost from the steam flowing in a nozzle. The velocity and the volume flow rate at the nozzle exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions.



Analysis We take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{\text{out}} \quad \text{since } \dot{W} \cong \Delta \text{pe} \cong 0)$$

or

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + \frac{\dot{Q}_{\text{out}}}{\dot{m}}$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \upsilon_1 = 0.38429 \text{ m}^3/\text{kg} \\ h_1 = 3267.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} \upsilon_2 = 1.31623 \text{ m}^3/\text{kg} \\ h_2 = 3072.1 \text{ kJ/kg} \end{array}$$

The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\upsilon_1} A_1 V_1 = \frac{1}{0.38429 \text{ m}^3/\text{s}} (0.08 \text{ m}^2)(10 \text{ m/s}) = 2.082 \text{ kg/s}$$

Substituting,

$$3267.7 \text{ kJ/kg} + \frac{(10 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3072.1 \text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) + \frac{25 \text{ kJ/s}}{2.082 \text{ kg/s}}$$

$$\longrightarrow V_2 = \mathbf{606 \text{ m/s}}$$

The volume flow rate at the exit of the nozzle is

$$\dot{V}_2 = \dot{m}\upsilon_2 = (2.082 \text{ kg/s})(1.31623 \text{ m}^3/\text{kg}) = \mathbf{2.74 \text{ m}^3/\text{s}}$$

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5-35 Steam is accelerated in a nozzle from a velocity of 40 m/s to 300 m/s. The exit temperature and the ratio of the inlet-to-exit area of the nozzle are to be determined.

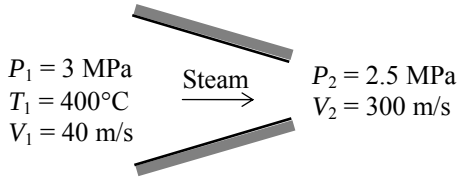
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-6),

$$\left.\begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \nu_1 = 0.09938 \text{ m}^3/\text{kg} \\ h_1 = 3231.7 \text{ kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 3231.7 \text{ kJ/kg} - \frac{(300 \text{ m/s})^2 - (40 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3187.5 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_2 = 2.5 \text{ MPa} \\ h_2 = 3187.5 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{376.6°C} \\ \nu_2 = 0.11533 \text{ m}^3/\text{kg} \end{array}$$

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\nu_2} A_2 V_2 = \frac{1}{\nu_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\nu_1}{\nu_2}\frac{V_2}{V_1} = \frac{(0.09938 \text{ m}^3/\text{kg})(300 \text{ m/s})}{(0.11533 \text{ m}^3/\text{kg})(40 \text{ m/s})} = \mathbf{6.46}$$

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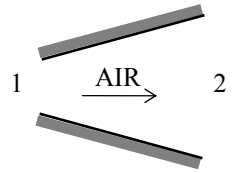
5-36E Air is decelerated in a diffuser from 600 ft/s to a low velocity. The exit temperature and the exit velocity of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The enthalpy of air at the inlet temperature of 50°F is h_1 = 121.88 Btu/lbm (Table A-17E).

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta pe \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \quad ,$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 121.88 \text{ Btu/lbm} - \frac{0 - (600 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 129.07 \text{ Btu/lbm}$$

From Table A-17E,

T_2 = **540 R**

(*b*) The exit velocity of air is determined from the conservation of mass relation,

$$\frac{1}{\nu_2} A_2 V_2 = \frac{1}{\nu_1} A_1 V_1 \longrightarrow \frac{1}{RT_2/P_2} A_2 V_2 = \frac{1}{RT_1/P_1} A_1 V_1$$

Thus,

$$V_2 = \frac{A_1 T_2 P_1}{A_2 T_1 P_2} V_1 = \frac{1}{4}\frac{(540 \text{ R})(13 \text{ psia})}{(510 \text{ R})(14.5 \text{ psia})}(600 \text{ ft/s}) = \mathbf{142 \text{ ft/s}}$$

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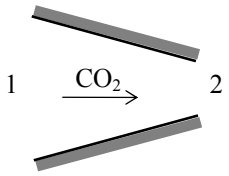
5-37 CO_2 gas is accelerated in a nozzle to 450 m/s. The inlet velocity and the exit temperature are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** CO_2 is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant and molar mass of CO_2 are 0.1889 kPa.m^3/kg.K and 44 kg/kmol (Table A-1). The enthalpy of CO_2 at 500°C is $\bar{h}_1 = 30{,}797$ kJ/kmol (Table A-20).

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume is determined to be

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(773\ \text{K})}{1000\ \text{kPa}} = 0.146\ \text{m}^3/\text{kg}$$



Thus,

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 \longrightarrow V_1 = \frac{\dot{m}\nu_1}{A_1} = \frac{(6000/3600\ \text{kg/s})(0.146\ \text{m}^3/\text{kg})}{40\times10^{-4}\ \text{m}^2} = \mathbf{60.8\ m/s}$$

(b) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}}^{\nearrow 0\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$\bar{h}_2 = \bar{h}_1 - \frac{V_2^2 - V_1^2}{2} M$$

$$= 30{,}797\ \text{kJ/kmol} - \frac{(450\ \text{m/s})^2 - (60.8\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)(44\ \text{kg/kmol})$$

$$= 26{,}423\ \text{kJ/kmol}$$

Then the exit temperature of CO_2 from Table A-20 is obtained to be $T_2 = \mathbf{685.8\ K}$

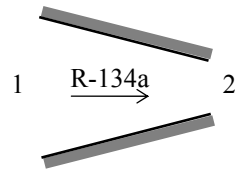
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5-38 R-134a is accelerated in a nozzle from a velocity of 20 m/s. The exit velocity of the refrigerant and the ratio of the inlet-to-exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the refrigerant tables (Table A-13)

$$\left.\begin{array}{l} P_1 = 700 \text{ kPa} \\ T_1 = 120°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.043358 \text{ m}^3/\text{kg} \\ h_1 = 358.90 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 400 \text{ kPa} \\ T_2 = 30°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.056796 \text{ m}^3/\text{kg} \\ h_2 = 275.07 \text{ kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$0 = (275.07 - 358.90)\text{kJ/kg} + \frac{V_2^2 - (20 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

It yields

V_2 **= 409.9 m/s**

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_1}{v_2}\frac{V_2}{V_1} = \frac{(0.043358 \text{ m}^3/\text{kg})(409.9 \text{ m/s})}{(0.056796 \text{ m}^3/\text{kg})(20 \text{ m/s})} = \mathbf{15.65}$$

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5-39 Nitrogen is decelerated in a diffuser from 275 m/s to a lower velocity. The exit velocity of nitrogen and the ratio of the inlet-to-exit area are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Nitrogen is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

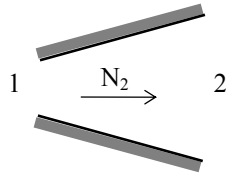
Properties The molar mass of nitrogen is $M = 28$ kg/kmol (Table A-1). The enthalpies are (Table A-18)

$$T_1 = 7°\text{C} = 280\ \text{K} \ \rightarrow\ \bar{h}_1 = 8141\ \text{kJ/kmol}$$
$$T_2 = 27°\text{C} = 300\ \text{K} \ \rightarrow\ \bar{h}_2 = 8723\ \text{kJ/kmol}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0\ \text{(steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$



$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$
$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = \frac{\bar{h}_2 - \bar{h}_1}{M} + \frac{V_2^2 - V_1^2}{2},$$

Substituting,

$$0 = \frac{(8723 - 8141)\ \text{kJ/kmol}}{28\ \text{kg/kmol}} + \frac{V_2^2 - (275\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

It yields

V_2 **= 185 m/s**

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\nu_2} A_2 V_2 = \frac{1}{\nu_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\nu_1}{\nu_2}\frac{V_2}{V_1} = \left(\frac{RT_1/P_1}{RT_2/P_2}\right)\frac{V_2}{V_1}$$

or,

$$\frac{A_1}{A_2} = \left(\frac{T_1/P_1}{T_2/P_2}\right)\frac{V_2}{V_1} = \frac{(280\ \text{K/60 kPa})(185\ \text{m/s})}{(300\ \text{K/85 kPa})(200\ \text{m/s})} = \mathbf{0.887}$$

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5-40 Problem 5-39 is reconsidered. The effect of the inlet velocity on the exit velocity and the ratio of the inlet-to-exit area as the inlet velocity varies from 210 m/s to 350 m/s is to be investigated. The final results are to be plotted against the inlet velocity.

Analysis The problem is solved using EES, and the solution is given below.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'N2' = WorkFluid$  then
       HCal:=ENTHALPY(WorkFluid$,T=Tx) "Ideal gas equ."
   else
      HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"
"Property relation: Nitrogen is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"

"Knowns"
WorkFluid$ = 'N2'
T[1] = 7  [C]
P[1] = 60   [kPa]
“Vel[1] = 275  [m/s]”
P[2] = 85  [kPa]
T[2] = 27 [C]

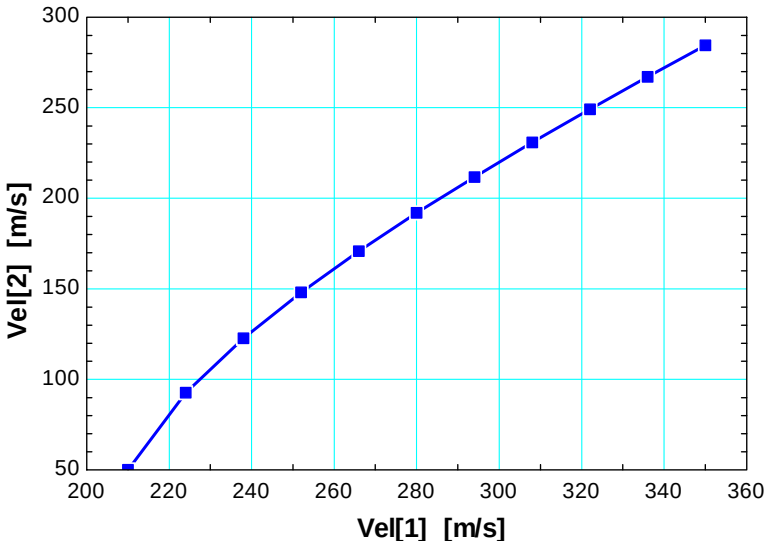
"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

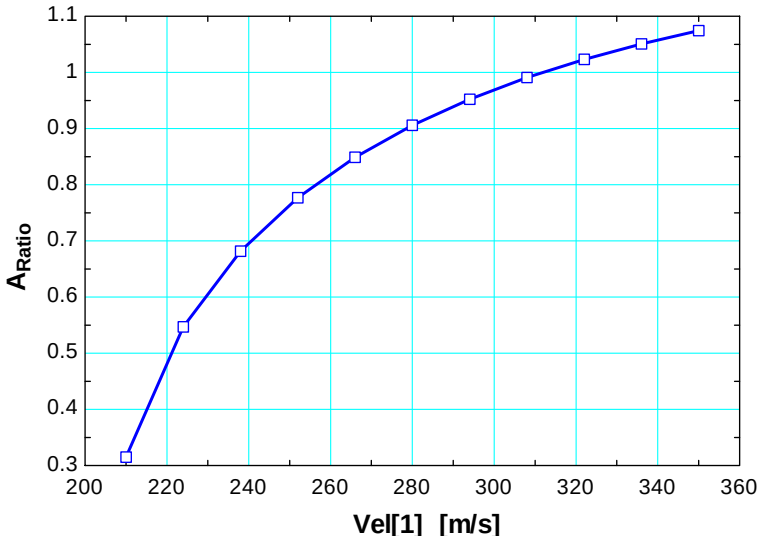
"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

"From the definition of mass flow rate, m_dot = A*Vel/v and conservation of mass the area ratio A_Ratio =
A_1/A_2 is:"
A_Ratio*Vel[1]/v[1] =Vel[2]/v[2]
"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)
```

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Vel_1 [m/s]	Vel_2 [m/s]	A_{Ratio}
210	50.01	0.3149
224	92.61	0.5467
238	122.7	0.6815
252	148	0.7766
266	170.8	0.8488
280	191.8	0.9059
294	211.7	0.9523
308	230.8	0.9908
322	249.2	1.023
336	267	1.051
350	284.4	1.075





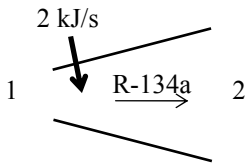
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5-41 R-134a is decelerated in a diffuser from a velocity of 120 m/s. The exit velocity of R-134a and the mass flow rate of the R-134a are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

Properties From the R-134a tables (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ \textit{sat.vapor} \end{array}\right\} \begin{array}{l} \nu_1 = 0.025621 \text{ m}^3/\text{kg} \\ h_1 = 267.29 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ T_2 = 40°\text{C} \end{array}\right\} \begin{array}{l} \nu_2 = 0.023375 \text{ m}^3/\text{kg} \\ h_2 = 274.17 \text{ kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity of R-134a is determined from the steady-flow mass balance to be

$$\frac{1}{\nu_2} A_2 V_2 = \frac{1}{\nu_1} A_1 V_1 \longrightarrow V_2 = \frac{\nu_2}{\nu_1}\frac{A_1}{A_2} V_1 = \frac{1}{1.8}\frac{(0.023375 \text{ m}^3/\text{kg})}{(0.025621 \text{ m}^3/\text{kg})}(120 \text{ m/s}) = \mathbf{60.8 \text{ m/s}}$$

(*b*) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate of the refrigerant is determined to be

$$2 \text{ kJ/s} = \dot{m}\left((274.17 - 267.29)\text{kJ/kg} + \frac{(60.8 \text{ m/s})^2 - (120 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

It yields

$$\dot{m} = \mathbf{1.308 \text{ kg/s}}$$

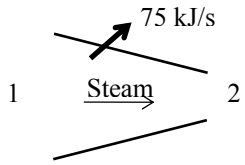
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5-42 Steam is accelerated in a nozzle from a velocity of 60 m/s. The mass flow rate, the exit velocity, and the exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

Properties From the steam tables (Table A-6)

$$\left.\begin{array}{l} P_1 = 4 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \nu_1 = 0.07343 \text{ m}^3/\text{kg} \\ h_1 = 3214.5 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 2 \text{ MPa} \\ T_2 = 300°\text{C} \end{array}\right\} \begin{array}{l} \nu_2 = 0.12551 \text{ m}^3/\text{kg} \\ h_2 = 3024.2 \text{ kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The mass flow rate of steam is

$$\dot{m} = \frac{1}{\nu_1} V_1 A_1 = \frac{1}{0.07343 \text{ m}^3/\text{kg}} (60 \text{ m/s})(50 \times 10^{-4} \text{ m}^2) = \mathbf{4.085 \text{ kg/s}}$$

(*b*) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta pe \cong 0)$$

$$-\dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Substituting, the exit velocity of the steam is determined to be

$$-75 \text{ kJ/s} = (4.085 \text{ kg/s})\left(3024.2 - 3214.5 + \frac{V_2^2 - (60 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right)$$

It yields

$$V_2 = \mathbf{589.5 \text{ m/s}}$$

(*c*) The exit area of the nozzle is determined from

$$\dot{m} = \frac{1}{\nu_2} V_2 A_2 \longrightarrow A_2 = \frac{\dot{m}\nu_2}{V_2} = \frac{(4.085 \text{ kg/s})(0.12551 \text{ m}^3/\text{kg})}{589.5 \text{ m/s}} = \mathbf{8.70 \times 10^{-4} \text{ m}^2}$$

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Turbines and Compressors

5-43C Yes.

5-44C The volume flow rate at the compressor inlet will be greater than that at the compressor exit.

5-45C Yes. Because energy (in the form of shaft work) is being added to the air.

5-46C No.

5-47 R-134a at a given state is compressed to a specified state. The mass flow rate and the power input are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

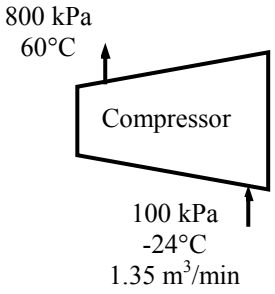
Analysis We take the compressor as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the compressor, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_2)$$



From R134a tables (Tables A-11, A-12, A-13)

$$\left.\begin{aligned} P_1 &= 100 \text{ kPa} \\ T_1 &= -24°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 236.33 \text{ kJ/kg} \\ v_1 &= 0.1947 \text{ m}^3/\text{kg} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 800 \text{ kPa} \\ T_2 &= 60°\text{C} \end{aligned}\right\} h_2 = 296.81 \text{ kJ/kg}$$

The mass flow rate is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{(1.35/60) \text{ m}^3/\text{s}}{0.1947 \text{ m}^3/\text{kg}} = \mathbf{0.1155 \text{ kg/s}}$$

Substituting,

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.1155 \text{ kg/s})(296.81 - 236.33)\text{kJ/kg} = \mathbf{6.99 \text{ kW}}$$

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5-48 Saturated R-134a vapor is compressed to a specified state. The power input is given. The exit temperature is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer with the surroundings is negligible.

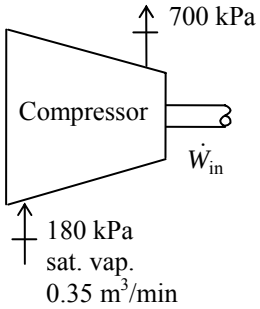
Analysis We take the compressor as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the compressor, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_2)$$



From R134a tables (Table A-12)

$$\left.\begin{array}{l} P_1 = 180 \text{ kPa} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 242.86 \text{ kJ/kg} \\ v_1 = 0.1104 \text{ m}^3/\text{kg} \end{array}$$

The mass flow rate is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{(0.35/60) \text{ m}^3/\text{s}}{0.1104 \text{ m}^3/\text{kg}} = 0.05283 \text{ kg/s}$$

Substituting for the exit enthalpy,

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

$$2.35 \text{ kW} = (0.05283 \text{ kg/s})(h_2 - 242.86)\text{kJ/kg} \longrightarrow h_2 = 287.34 \text{ kJ/kg}$$

From Table A-13,

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ h_2 = 287.34 \text{ kJ/kg} \end{array}\right\} T_2 = \mathbf{48.8°C}$$

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5-49 Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

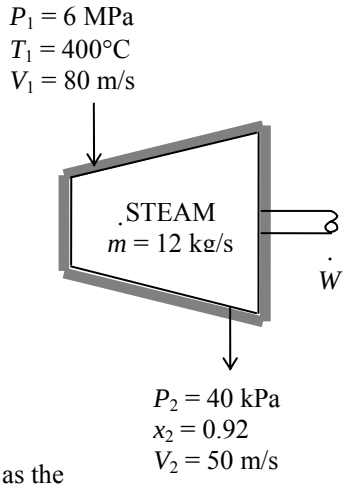
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{matrix} P_1 = 6\ \text{MPa} \\ T_1 = 400°\text{C} \end{matrix}\right\} \begin{matrix} \nu_1 = 0.047420\ \text{m}^3/\text{kg} \\ h_1 = 3178.3\ \text{kJ/kg} \end{matrix}$$

and

$$\left.\begin{matrix} P_2 = 40\ \text{kPa} \\ x_2 = 0.92 \end{matrix}\right\} h_2 = h_f + x_2 h_{fg} = 317.62 + 0.92 \times 2392.1 = 2318.5\ \text{kJ/kg}$$



Analysis (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(50\ \text{m/s})^2 - (80\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{-1.95\ kJ/kg}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{\text{out}} = -(20\ \text{kg/s})(2318.5 - 3178.3 - 1.95)\text{kJ/kg} = 14{,}590\ \text{kW} = \mathbf{14.6\ MW}$$

(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 \longrightarrow A_1 = \frac{\dot{m}\nu_1}{V_1} = \frac{(20\ \text{kg/s})(0.047420\ \text{m}^3/\text{kg})}{80\ \text{m/s}} = \mathbf{0.0119\ m^2}$$

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5-50 Problem 5-49 is reconsidered. The effect of the turbine exit pressure on the power output of the turbine as the exit pressure varies from 10 kPa to 200 kPa is to be investigated. The power output is to be plotted against the exit pressure.

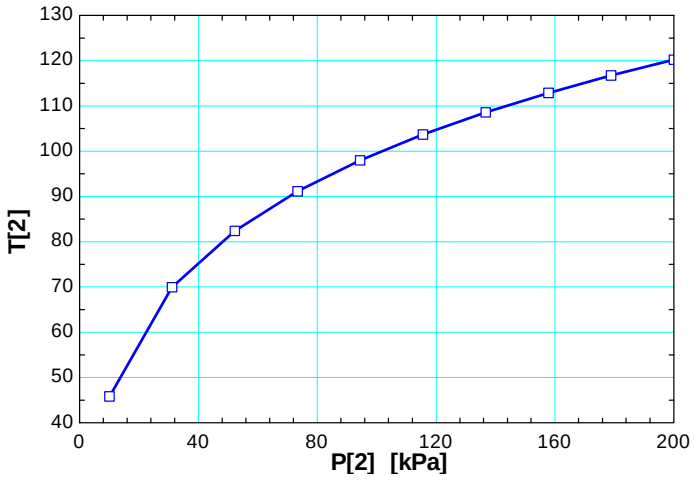
Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns "
T[1] = 450 [C]
P[1] = 6000  [kPa]
Vel[1] = 80  [m/s]
P[2] = 40  [kPa]
X_2=0.92
Vel[2] = 50  [m/s]
m_dot[1]=12 [kg/s]
Fluid$='Steam_IAPWS'

"Property Data"
h[1]=enthalpy(Fluid$,T=T[1],P=P[1])
h[2]=enthalpy(Fluid$,P=P[2],x=x_2)
T[2]=temperature(Fluid$,P=P[2],x=x_2)
v[1]=volume(Fluid$,T=T[1],p=P[1])
v[2]=volume(Fluid$,P=P[2],x=x_2)

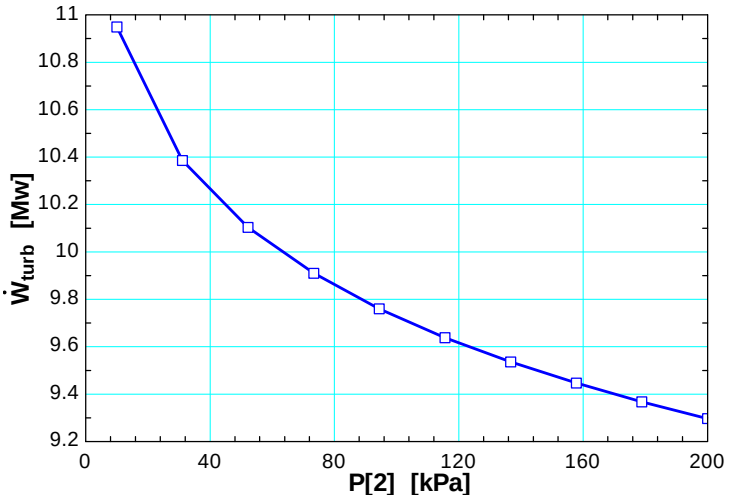
"Conservation of mass: "
m_dot[1]= m_dot[2]

"Mass flow rate"
m_dot[1]=A[1]*Vel[1]/v[1]
m_dot[2]= A[2]*Vel[2]/v[2]
```




```
"Conservation of Energy - Steady Flow energy balance"
m_dot[1]*(h[1]+Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)) = m_dot[2]*(h[2]+Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+W_dot_turb*convert(MW,kJ/s)
DELTAke=Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)-Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)
```

P_2 [kPa]	W_{turb} [MW]	T_2 [C]
10	10.95	45.81
31.11	10.39	69.93
52.22	10.1	82.4
73.33	9.909	91.16
94.44	9.76	98.02
115.6	9.638	103.7
136.7	9.535	108.6
157.8	9.446	112.9
178.9	9.367	116.7
200	9.297	120.2



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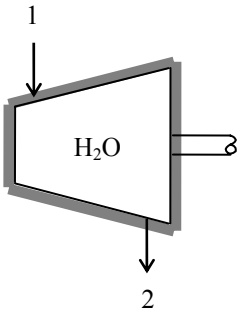
5-51 Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{matrix} P_1 = 10 \text{ MPa} \\ T_1 = 500°\text{C} \end{matrix}\right\} h_1 = 3375.1 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 10 \text{ kPa} \\ x_2 = 0.90 \end{matrix}\right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.90 \times 2392.1 = 2344.7 \text{ kJ/kg}$$



Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}(h_2 - h_1)$$

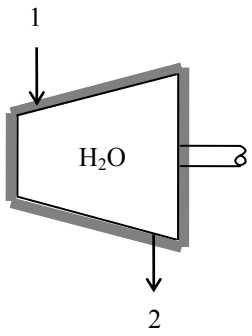
Substituting, the required mass flow rate of the steam is determined to be

$$5000 \text{ kJ/s} = -\dot{m}(2344.7 - 3375.1) \text{ kJ/kg} \longrightarrow \dot{m} = \mathbf{4.852 \text{ kg/s}}$$

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5-52E Steam expands in a turbine. The rate of heat loss from the steam for a power output of 4 MW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.



Properties From the steam tables (Tables A-4E through 6E)

$$\left.\begin{matrix} P_1 = 1000 \text{ psia} \\ T_1 = 900°\text{F} \end{matrix}\right\} h_1 = 1448.6 \text{ Btu/lbm}$$

$$\left.\begin{matrix} P_2 = 5 \text{ psia} \\ sat.vapor \end{matrix}\right\} h_2 = 1130.7 \text{ Btu/lbm}$$

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{out}} = -\dot{m}(h_2 - h_1) - \dot{W}_{\text{out}}$$

Substituting,

$$\dot{Q}_{\text{out}} = -(45000/3600 \text{ lbm/s})(1130.7 - 1448.6)\text{Btu/lbm} - 4000 \text{ kJ/s}\left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}}\right) = \mathbf{182.0 \text{ Btu/s}}$$

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5-53 Air is compressed at a rate of 10 L/s by a compressor. The work required per unit mass and the power required are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of (20+300)/2=160°C=433 K is c_p = 1.018 kJ/kg·K (Table A-2b). The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1).

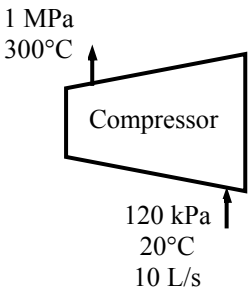
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



Thus,

$$w_{\text{in}} = c_p(T_2 - T_1) = (1.018\,\text{kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{285.0\ kJ/kg}$$

(*b*) The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273\,\text{K})}{120\,\text{kPa}} = 0.7008\,\text{m}^3/\text{kg}$$

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.010\,\text{m}^3/\text{s}}{0.7008\,\text{m}^3/\text{kg}} = 0.01427\,\text{kg/s}$$

Then the power input is determined from the energy balance equation to be

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = (0.01427\,\text{kg/s})(1.018\,\text{kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{4.068\ kW}$$

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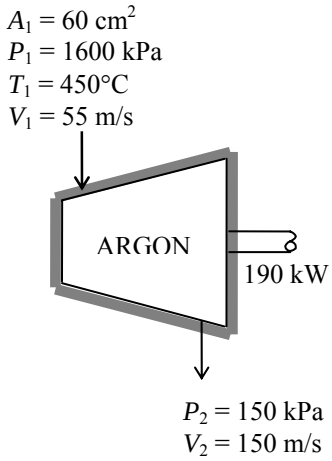
5-54 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 190 kW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

Properties The gas constant of Ar is $R = 0.2081$ kPa.m^3/kg.K. The constant pressure specific heat of Ar is $c_p = 0.5203$ kJ/kg·°C (Table A-2a)

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of argon and its mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.2081\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(723\ \text{K})}{1600\ \text{kPa}} = 0.09404\ \text{m}^3/\text{kg}$$



Thus,

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 = \frac{1}{0.09404\ \text{m}^3/\text{kg}}(0.006\ \text{m}^2)(55\ \text{m/s}) = 3.509\ \text{kg/s}$$

We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting,

$$190\ \text{kJ/s} = -(3.509\ \text{kg/s})\left[(0.5203\ \text{kJ/kg}\cdot\text{°C})(T_2 - 450\text{°C}) + \frac{(150\ \text{m/s})^2 - (55\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

It yields

$T_2 =$ **327°C**

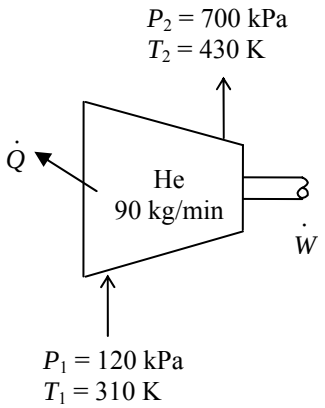
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5-55 Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of helium is $c_p = 5.1926$ kJ/kg·K (Table A-2a).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

$$\begin{aligned}\dot{W}_{\text{in}} &= \dot{Q}_{\text{out}} + \dot{m}c_p(T_2 - T_1) \\ &= (90/6\,0 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg} \cdot \text{K})(430 - 310)\text{K} \\ &= \mathbf{965\ kW}\end{aligned}$$

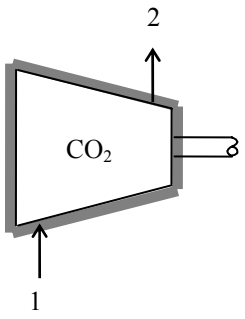
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5-56 CO_2 is compressed by a compressor. The volume flow rate of CO_2 at the compressor inlet and the power input to the compressor are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with variable specific heats. **4** The device is adiabatic and thus heat transfer is negligible.

Properties The gas constant of CO_2 is $R = 0.1889$ kPa.m^3/kg.K, and its molar mass is $M = 44$ kg/kmol (Table A-1). The inlet and exit enthalpies of CO_2 are (Table A-20)

$$T_1 = 300 \text{ K} \quad \rightarrow \quad \bar{h}_1 = 9{,}431 \text{ kJ/kmol}$$
$$T_2 = 450 \text{ K} \quad \rightarrow \quad \bar{h}_2 = 15{,}483 \text{ kJ/kmol}$$



Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its volume flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})}{100 \text{ kPa}} = 0.5667 \text{ m}^3/\text{kg}$$

$$\dot{V} = \dot{m}\nu_1 = (0.5 \text{ kg/s})(0.5667 \text{ m}^3/\text{kg}) = \mathbf{0.283 \text{ m}^3/\text{s}}$$

(*b*) We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}(\bar{h}_2 - \bar{h}_1)/M$$

Substituting

$$\dot{W}_{in} = \frac{(0.5 \text{ kg/s})(15{,}483 - 9{,}431 \text{ kJ/kmol})}{44 \text{ kg/kmol}} = \mathbf{68.8 \text{ kW}}$$

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5-57 Air is expanded in an adiabatic turbine. The mass flow rate of the air and the power produced are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The turbine is well-insulated, and thus there is no heat transfer. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of (500+127)/2=314°C=587 K is c_p = 1.048 kJ/kg·K (Table A-2*b*). The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1).

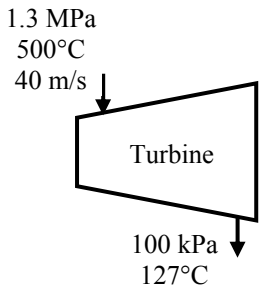
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}\left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right) = \dot{m}\left(c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right)$$



The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500 + 273\ \text{K})}{1300\ \text{kPa}} = 0.1707\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{(0.2\ \text{m}^2)(40\ \text{m/s})}{0.1707\ \text{m}^3/\text{kg}} = \mathbf{46.88\ kg/s}$$

Similarly at the outlet,

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(127 + 273\ \text{K})}{100\ \text{kPa}} = 1.148\ \text{m}^3/\text{kg}$$

$$V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{(46.88\ \text{kg/s})(1.148\ \text{m}^3/\text{kg})}{1\ \text{m}^2} = 53.82\ \text{m/s}$$

(*b*) Substituting into the energy balance equation gives

$$\dot{W}_{\text{out}} = \dot{m}\left(c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right)$$

$$= (46.88\ \text{kg/s})\left[(1.048\ \text{kJ/kg}\cdot\text{K})(500 - 127)\text{K} + \frac{(40\ \text{m/s})^2 - (53.82\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{18{,}300\ kW}$$

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5-58E Air is expanded in an adiabatic turbine. The mass flow rate of the air and the power produced are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The turbine is well-insulated, and thus there is no heat transfer. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of (800+250)/2=525°F is $c_p = 0.2485$ Btu/lbm·R (Table A-2E*b*). The gas constant of air is $R = 0.3704$ psia·ft^3/lbm·R (Table A-1E).

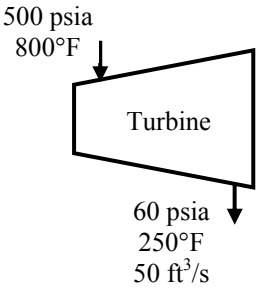
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}\left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right) = \dot{m}\left(c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right)$$



The specific volume of air at the exit and the mass flow rate are

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(250+460\ \text{R})}{60\ \text{psia}} = 4.383\ \text{ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_2}{\nu_2} = \frac{50\ \text{ft}^3/\text{s}}{4.383\ \text{ft}^3/\text{lbm}} = 11.41\ \text{kg/s}$$

$$V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{(11.41\ \text{lbm/s})(4.383\ \text{ft}^3/\text{lbm})}{1.2\ \text{ft}^2} = 41.68\ \text{ft/s}$$

Similarly at the inlet,

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(800+460\ \text{R})}{500\ \text{psia}} = 0.9334\ \text{ft}^3/\text{lbm}$$

$$V_1 = \frac{\dot{m}\nu_1}{A_1} = \frac{(11.41\ \text{lbm/s})(0.9334\ \text{ft}^3/\text{lbm})}{0.6\ \text{ft}^2} = 17.75\ \text{ft/s}$$

Substituting into the energy balance equation gives

$$\dot{W}_{\text{out}} = \dot{m}\left(c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right)$$

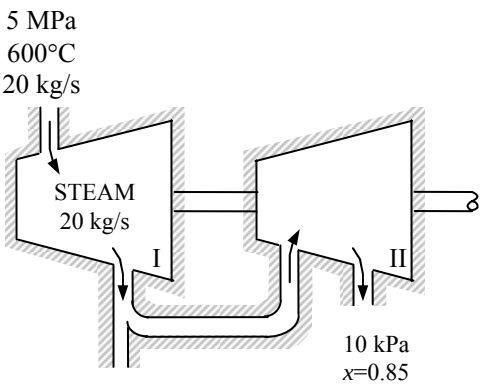
$$= (11.41\ \text{lbm/s})\left[(0.2485\ \text{Btu/lbm}\cdot\text{R})(800-250)\text{R} + \frac{(17.75\ \text{ft/s})^2 - (41.68\ \text{m/s})^2}{2}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{1559\ Btu/s} = \mathbf{1645\ kW}$$

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5-59 Steam expands in a two-stage adiabatic turbine from a specified state to another state. Some steam is extracted at the end of the first stage. The power output of the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible.



Properties From the steam tables (Tables A-5 and A-6)

$$\left.\begin{matrix} P_1 = 5\ \text{MPa} \\ T_1 = 600°\text{C} \end{matrix}\right\} h_1 = 3666.9\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_2 = 0.5\ \text{MPa} \\ x_2 = 1 \end{matrix}\right\} h_2 = 2748.1\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 10\ \text{kPa} \\ x_2 = 0.85 \end{matrix}\right\} \begin{matrix} h_3 = h_f + xh_{fg} \\ = 191.81 + (0.85)(2392.1) = 2225.1\ \text{kJ/kg} \end{matrix}$$

Analysis We take the entire turbine, including the connection part between the two stages, as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters the turbine and two fluid streams leave, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}_1 (h_1 - 0.1h_2 - 0.9h_3)$$

Substituting, the power output of the turbine is

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m}_1 (h_1 - 0.1h_2 - 0.9h_3) \\ &= (20\ \text{kg/s})(3666.9 - 0.1 \times 2748.1 - 0.9 \times 2225.1)\ \text{kJ/kg} \\ &= \mathbf{27{,}790\ kW} \end{aligned}$$

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5-60 Steam is expanded in a turbine. The power output is given. The rate of heat transfer is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties From the steam tables (Table A-4, A5, A-6)

$$\left.\begin{matrix} P_1 = 6\ \text{MPa} \\ T_1 = 600°\text{C} \end{matrix}\right\} h_1 = 3658.8\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_2 = 0.5\ \text{MPa} \\ T_2 = 200°\text{C} \end{matrix}\right\} h_2 = 2855.8\ \text{kJ/kg}$$

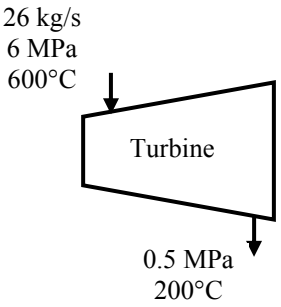
Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the compressor, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{W}_{\text{out}} + \dot{Q}_{\text{out}} \quad (\text{since } \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = -\dot{W}_{\text{out}} + \dot{m}\left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right)$$



Substituting,

$$\dot{Q}_{\text{out}} = -\dot{W}_{\text{out}} + \dot{m}\left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right)$$

$$= 20{,}000\ \text{kW} + (26\ \text{kg/s})\left[(3658.8 - 2855.8)\text{kJ/kg} + \frac{(0 - 180\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{455\ kW}$$

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5-61 Helium at a specified state is compressed to another specified state. The power input is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas.

Properties The properties of helium are c_p = 5.1926 kJ/kg·K and R = 2.0769 kPa·m^3/kg·K (Table A-2*a*).

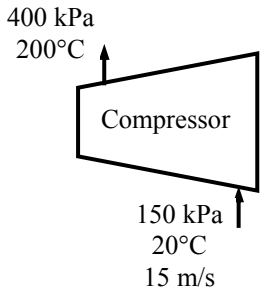
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



The mass flow rate is determined from

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{A_1 V_1 P_1}{RT_1} = \frac{(0.1\,\text{m}^2)(15\,\text{m/s})(150\,\text{kPa})}{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})} 0.3697\,\text{kg/s}$$

Substituting,

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = (0.3697\,\text{kg/s})(5.1926\,\text{kJ/kg}\cdot\text{K})(200 - 20)\text{K} = \mathbf{346\ kW}$$

The flow power input is determined from

$$\dot{W}_{fw} = \dot{m}(P_2\nu_2 - P_1\nu_1) = \dot{m}R(T_2 - T_1) = (0.3697\,\text{kg/s})(2.0769\,\text{kJ/kg}\cdot\text{K})(200 - 20)\text{K} = \mathbf{138\ kW}$$

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Throttling Valves

5-62C The temperature of a fluid can increase, decrease, or remain the same during a throttling process. Therefore, this claim is valid since no thermodynamic laws are violated.

5-63C No. Because air is an ideal gas and $h = h(T)$ for ideal gases. Thus if h remains constant, so does the temperature.

5-64C If it remains in the liquid phase, no. But if some of the liquid vaporizes during throttling, then yes.

5-65C Yes.

5-66 Refrigerant-134a is throttled by a capillary tube. The quality of the refrigerant at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

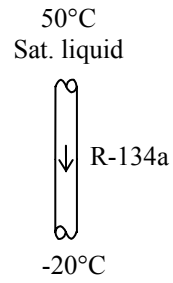
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0$.

The inlet enthalpy of R-134a is, from the refrigerant tables (Table A-11),

$$\left.\begin{array}{l} T_1 = 50°\text{C} \\ \text{sat. liquid} \end{array}\right\} h_1 = h_f = 123.49 \text{ kJ/kg}$$

The exit quality is

$$\left.\begin{array}{l} T_2 = -20°\text{C} \\ h_2 = h_1 \end{array}\right\} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{123.49 - 25.49}{212.91} = \mathbf{0.460}$$

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5-67 Steam is throttled from a specified pressure to a specified state. The quality at the inlet is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

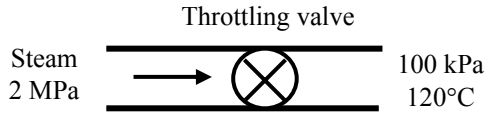
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



Since $\dot{Q} \cong \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0$.

The enthalpy of steam at the exit is (Table A-6),

$$\left.\begin{array}{l} P_2 = 100\,\text{kPa} \\ T_2 = 120°\text{C} \end{array}\right\} h_2 = 2716.1\,\text{kJ/kg}$$

The quality of the steam at the inlet is (Table A-5)

$$\left.\begin{array}{l} P_1 = 2000\,\text{kPa} \\ h_1 = h_2 = 2716.1\,\text{kJ/kg} \end{array}\right\} x_1 = \frac{h_2 - h_f}{h_{fg}} = \frac{2716.1 - 908.47}{1889.8} = \mathbf{0.957}$$

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5-68 Refrigerant-134a is throttled by a valve. The pressure and internal energy after expansion are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of R-134a is, from the refrigerant tables (Tables A-11 through 13),

$$\left.\begin{matrix} P_1 = 0.8 \text{ MPa} \\ T_1 = 25°\text{C} \end{matrix}\right\} h_1 \cong h_{f@25°\text{C}} = 86.41 \text{ kJ/kg}$$

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

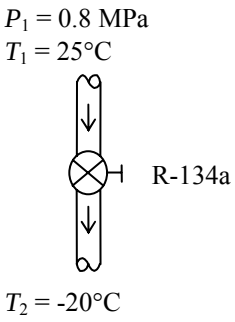
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left.\begin{matrix} T_2 = -20°\text{C} \\ (h_2 = h_1) \end{matrix}\right\} \begin{matrix} h_f = 25.49 \text{ kJ/kg}, & u_f = 25.39 \text{ kJ/kg} \\ h_g = 238.41 \text{ kJ/kg} & u_g = 218.84 \text{ kJ/kg} \end{matrix}$$



Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state, and thus

$$P_2 = P_{\text{sat @ -20°C}} = \mathbf{132.82 \text{ kPa}}$$

Also,

$$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{86.41 - 25.49}{212.91} = 0.2861$$

Thus,

$$u_2 = u_f + x_2 u_{fg} = 25.39 + 0.2861 \times 193.45 = \mathbf{80.74 \text{ kJ/kg}}$$

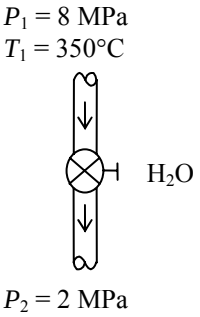
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5-69 Steam is throttled by a well-insulated valve. The temperature drop of the steam after the expansion is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of steam is (Tables A-6),

$$\left.\begin{matrix} P_1 = 8 \text{ MPa} \\ T_1 = 350°\text{C} \end{matrix}\right\} h_1 = 2988.1 \text{ kJ/kg}$$



Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then the exit temperature of steam becomes

$$\left.\begin{matrix} P_2 = 2 \text{ MPa} \\ (h_2 = h_1) \end{matrix}\right\} T_2 = \mathbf{285°C}$$

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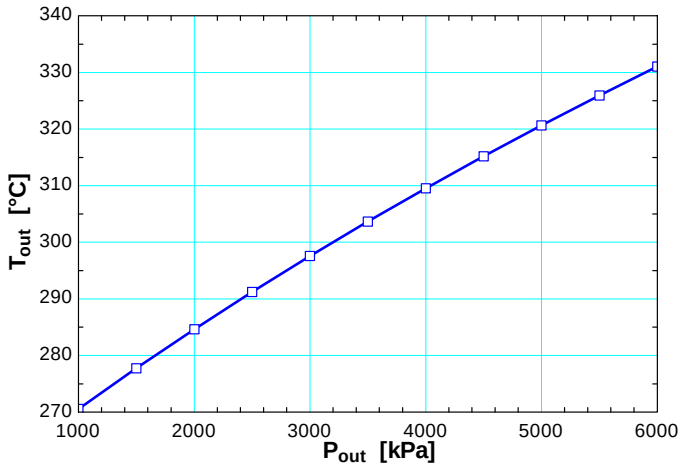
5-70 Problem 5-69 is reconsidered. The effect of the exit pressure of steam on the exit temperature after throttling as the exit pressure varies from 6 MPa to 1 MPa is to be investigated. The exit temperature of steam is to be plotted against the exit pressure.

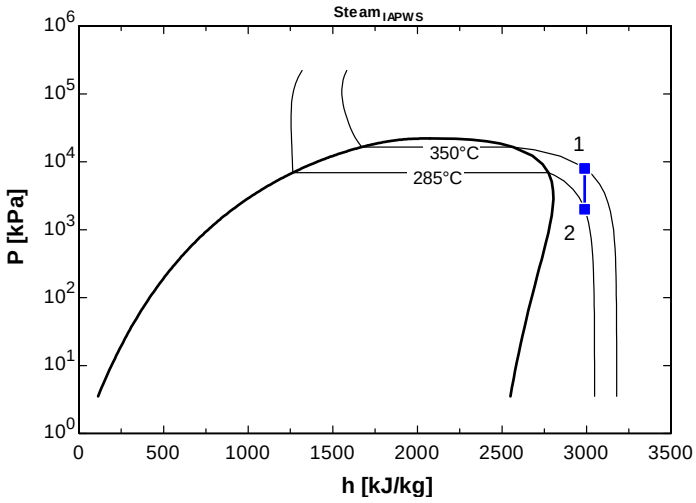
Analysis The problem is solved using EES, and the solution is given below.

```
WorkingFluid$='Steam_iapws' "WorkingFluid can be changed to ammonia or other fluids"
P_in=8000 [kPa]
T_in=350  [C]
P_out=2000  [kPa]

"Analysis"
m_dot_in=m_dot_out  "steady-state mass balance"
m_dot_in=1  "mass flow rate is arbitrary"
m_dot_in*h_in+Q_dot-W_dot-m_dot_out*h_out=0  "steady-state energy balance"
Q_dot=0  "assume the throttle to operate adiabatically"
W_dot=0  "throttles do not have any means of producing power"
h_in=enthalpy(WorkingFluid$,T=T_in,P=P_in)  "property table lookup"
T_out=temperature(WorkingFluid$,P=P_out,h=h_out)  "property table lookup"
x_out=quality(WorkingFluid$,P=P_out,h=h_out)  "x_out is the quality at the outlet"
P[1]=P_in;  P[2]=P_out;  h[1]=h_in;  h[2]=h_out  "use arrays to place points on property plot"
```

P_{out} [kPa]	T_{out} [C]
1000	270.5
1500	277.7
2000	284.6
2500	291.2
3000	297.6
3500	303.7
4000	309.5
4500	315.2
5000	320.7
5500	325.9
6000	331





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5-71E Refrigerant-134a is throttled by a valve. The temperature and internal energy change are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

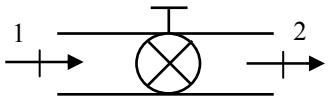
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. The properties are (Tables A-11E through 13E),

$$\left.\begin{array}{l} P_1 = 160 \text{ psia} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 48.52 \text{ Btu/lbm} \\ u_1 = 48.10 \text{ Btu/lbm} \\ T_1 = 109.5°\text{F} \end{array}$$

$$\left.\begin{array}{l} P_2 = 30 \text{ psia} \\ h_2 = h_1 = 48.52 \text{ Btu/lbm} \end{array}\right\} \begin{array}{l} T_2 = 15.4°\text{F} \\ u_2 = 45.41 \text{ Btu/lbm} \end{array}$$

$$\Delta T = T_2 - T_1 = 15.4 - 109.5 = \mathbf{-94.1°F}$$

$$\Delta u = u_2 - u_1 = 45.41 - 48.10 = \mathbf{-2.7 \text{ Btu/lbm}}$$

That is, the temperature drops by 94.1°F and internal energy drops by 2.7 Btu/lbm.

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Mixing Chambers and Heat Exchangers

5-72C Under the conditions of no heat and work interactions between the mixing chamber and the surrounding medium.

5-73C Under the conditions of no heat and work interactions between the heat exchanger and the surrounding medium.

5-74C Yes, if the mixing chamber is losing heat to the surrounding medium.

5-75 Hot and cold streams of a fluid are mixed in a mixing chamber. Heat is lost from the chamber. The energy carried from the mixing chamber is to be determined.

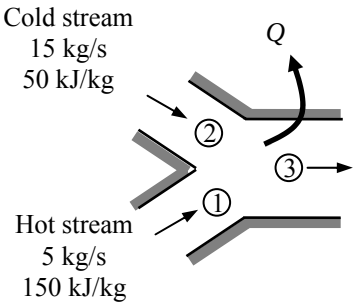
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions.

Analysis We take the mixing device as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 e_1 + \dot{m}_2 e_2 = \dot{m}_3 e_3 + \dot{Q}_{\text{out}}$$



From a mass balance

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 5 + 15 = 20 \text{ kg/s}$$

Substituting into the energy balance equation solving for the exit enthalpy gives

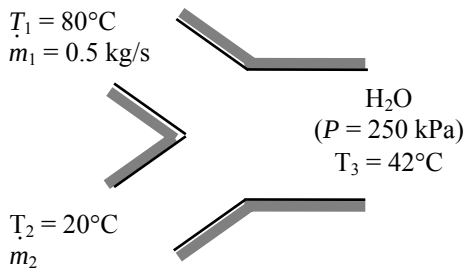
$$\dot{m}_1 e_1 + \dot{m}_2 e_2 = \dot{m}_3 e_3 + \dot{Q}_{\text{out}}$$

$$e_3 = \frac{\dot{m}_1 e_1 + \dot{m}_2 e_2 - \dot{Q}_{\text{out}}}{\dot{m}_3} = \frac{(5 \text{ kg/s})(150 \text{ kJ/kg} + (15 \text{ kg/s})(50 \text{ kJ/kg}) - 5.5 \text{ kW}}{20 \text{ kg/s}} = \mathbf{74.7 \text{ kJ/kg}}$$

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5-76 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There are no work interactions.



Properties Noting that $T < T_{\text{sat @ 250 kPa}} = 127.41°C$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,

$$h_1 \cong h_{f\,@\,80°C} = 335.02 \text{ kJ/kg}$$
$$h_2 \cong h_{f\,@\,20°C} = 83.915 \text{ kJ/kg}$$
$$h_3 \cong h_{f\,@\,42°C} = 175.90 \text{ kJ/kg}$$

Analysis We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two relations and solving for $\dot{m}_2$ gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2} \dot{m}_1$$

Substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{(335.02 - 175.90)\text{ kJ/kg}}{(175.90 - 83.915)\text{ kJ/kg}} (0.5 \text{ kg/s}) = \mathbf{0.865 \text{ kg/s}}$$

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5-77E Liquid water is heated in a chamber by mixing it with saturated water vapor. If both streams enter at the same rate, the temperature and quality (if saturated) of the exit stream is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From steam tables (Tables A-5E through A-6E),

$$h_1 \cong h_{f\,@\,65°\text{F}} = 33.08 \text{ Btu/lbm}$$

$$h_2 = h_{g\,@\,20 \text{ psia}} = 1156.2 \text{ Btu/lbm}$$

Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

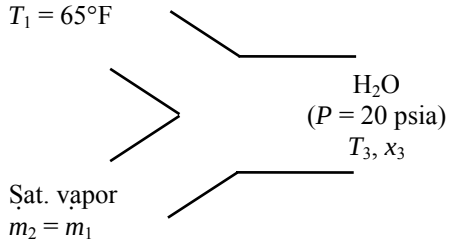
Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 2\dot{m}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$



Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two gives

$$\dot{m}h_1 + \dot{m}h_2 = 2\dot{m}h_3 \text{ or } h_3 = (h_1 + h_2)/2$$

Substituting,

$$h_3 = (33.08 + 1156.2)/2 = 594.6 \text{ Btu/lbm}$$

At 20 psia, h_f = 196.27 Btu/lbm and h_g = 1156.2 Btu/lbm. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{\text{sat @ 20 psia}} = \mathbf{228°F}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{594.6 - 196.27}{1156.2 - 196.27} = \mathbf{0.415}$$

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5-78 Two streams of refrigerant-134a are mixed in a chamber. If the cold stream enters at twice the rate of the hot stream, the temperature and quality (if saturated) of the exit stream are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From R-134a tables (Tables A-11 through A-13),

$$h_1 \cong h_{f\,@\,20°C} = 79.32 \text{ kJ/kg}$$

$$h_2 = h_{@\,1 \text{ MPa},\ 80°C} = 314.25 \text{ kJ/kg}$$

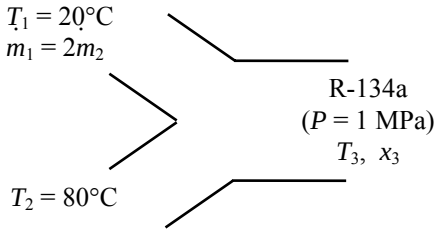
Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2 \quad \text{since } \dot{m}_1 = 2\dot{m}_2$$



Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two gives

$$2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3 \ \textit{or}\ h_3 = (2h_1 + h_2)/3$$

Substituting,

$$h_3 = (2 \times 79.32 + 314.25)/3 = 157.63 \text{ kJ/kg}$$

At 1 MPa, $h_f = 107.32$ kJ/kg and $h_g = 270.99$ kJ/kg. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{\text{sat @ 1 MPa}} = \mathbf{39.37°C}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{157.63 - 107.32}{270.99 - 107.32} = \mathbf{0.307}$$

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5-79 Problem 5-78 is reconsidered. The effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream as the ratio of the mass flow rate of the cold stream to that of the hot stream varies from 1 to 4 is to be investigated. The mixture temperature and quality are to be plotted against the cold-to-hot mass flow rate ratio.

Analysis The problem is solved using EES, and the solution is given below.

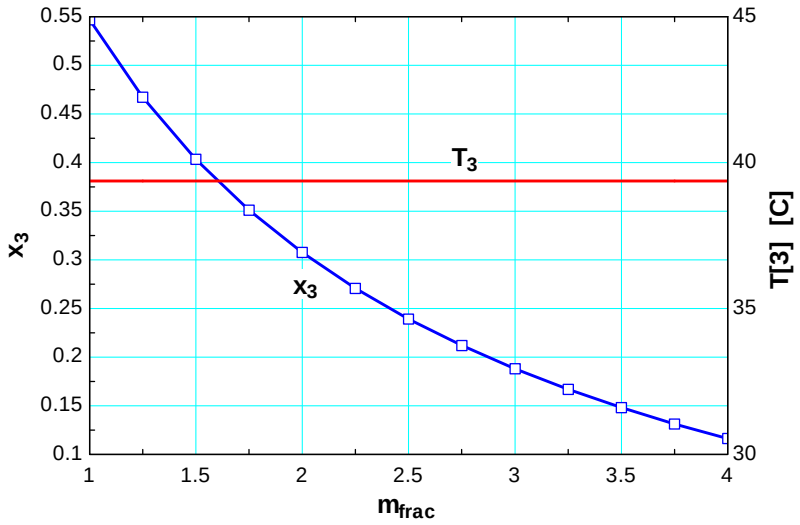
```
"Input Data"
m_frac = 2  "m_frac =m_dot_cold/m_dot_hot= m_dot_1/m_dot_2"
T[1]=20 [C]
P[1]=1000 [kPa]
T[2]=80 [C]
P[2]=1000 [kPa]
m_dot_1=m_frac*m_dot_2
P[3]=1000 [kPa]
m_dot_1=1

"Conservation of mass for the R134a: Sum of m_dot_in=m_dot_out"
m_dot_1+ m_dot_2 =m_dot_3

"Conservation of Energy for steady-flow: neglect changes in KE and PE"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0   "Steady-flow requirement"
E_dot_in=m_dot_1*h[1] + m_dot_2*h[2]
E_dot_out=m_dot_3*h[3]

"Property data are given by:"
h[1] =enthalpy(R134a,T=T[1],P=P[1])
h[2] =enthalpy(R134a,T=T[2],P=P[2])
T[3] =temperature(R134a,P=P[3],h=h[3])
x_3=QUALITY(R134a,h=h[3],P=P[3])
```

m_{frac}	T_3 [C]	x_3
1	39.37	0.5467
1.25	39.37	0.467
1.5	39.37	0.4032
1.75	39.37	0.351
2	39.37	0.3075
2.25	39.37	0.2707
2.5	39.37	0.2392
2.75	39.37	0.2119
3	39.37	0.188
3.25	39.37	0.1668
3.5	39.37	0.1481
3.75	39.37	0.1313
4	39.37	0.1162

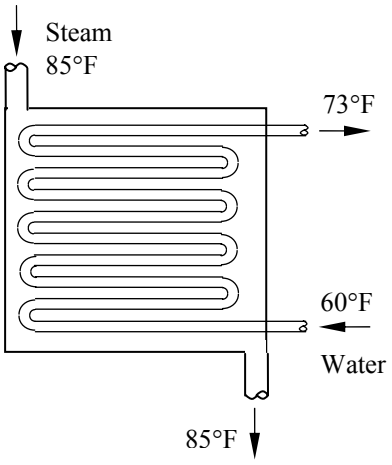


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5-80E Steam is condensed by cooling water in a condenser. The rate of heat transfer in the heat exchanger and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy of vaporization of water at 85°F is 1045.2 Btu/lbm (Table A-4E).



Analysis We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (138 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73°\text{F} - 60°\text{F}) = \mathbf{1794 \text{ Btu/s}}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1794 \text{ Btu/s}}{1045.2 \text{ Btu/lbm}} = \mathbf{1.72 \text{ lbm/s}}$$

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5-81 Steam is condensed by cooling water in the condenser of a power plant. If the temperature rise of the cooling water is not to exceed 10°C, the minimum mass flow rate of the cooling water required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Liquid water is an incompressible substance with constant specific heats at room temperature.

Properties The cooling water exists as compressed liquid at both states, and its specific heat at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3). The enthalpies of the steam at the inlet and the exit states are (Tables A-5 and A-6)

$$\left.\begin{array}{l} P_3 = 20 \text{ kPa} \\ x_3 = 0.95 \end{array}\right\} h_3 = h_f + x_3 h_{fg} = 251.42 + 0.95 \times 2357.5 = 2491.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_4 \cong h_{\text{f@20 kPa}} = 251.42 \text{ kJ/kg}$$

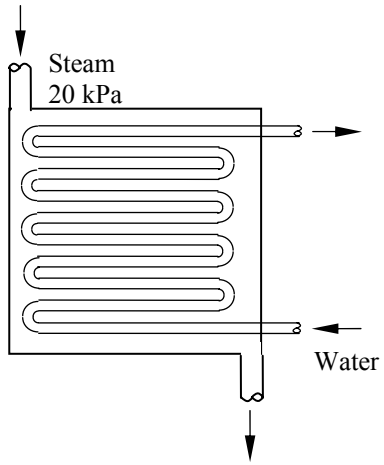
Analysis We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_w \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$



Energy balance (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two,

$$\dot{m}_w (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$$

Solving for $\dot{m}_w$:

$$\dot{m}_w = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \frac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$$

Substituting,

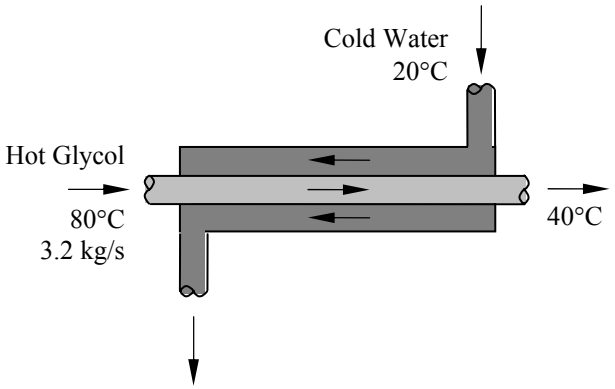
$$\dot{m}_w = \frac{(2491.1 - 251.42) \text{kJ/kg}}{(4.18 \text{ kJ/kg·}^\circ\text{C})(10^\circ\text{C})} (20{,}000/3600 \text{ kg/s}) = \mathbf{297.7 \text{ kg/s}}$$

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5-82 Ethylene glycol is cooled by water in a heat exchanger. The rate of heat transfer in the heat exchanger and the mass flow rate of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg.°C, respectively.



Analysis (*a*) We take the ethylene glycol tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{glycol}} = (3.2 \text{ kg/s})(2.56 \text{ kJ/kg.°C})(80°\text{C} - 40°\text{C}) = \mathbf{327.7\ kW}$$

(*b*) The rate of heat transfer from glycol must be equal to the rate of heat transfer to the water. Then,

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{327.7 \text{ kJ/s}}{(4.18 \text{ kJ/kg.°C})(70°\text{C} - 20°\text{C})} = \mathbf{1.57\ kg/s}$$

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5-83 Problem 5-82 is reconsidered. The effect of the inlet temperature of cooling water on the mass flow rate of water as the inlet temperature varies from 10°C to 40°C at constant exit temperature) is to be investigated. The mass flow rate of water is to be plotted against the inlet temperature.

Analysis The problem is solved using EES, and the solution is given below.

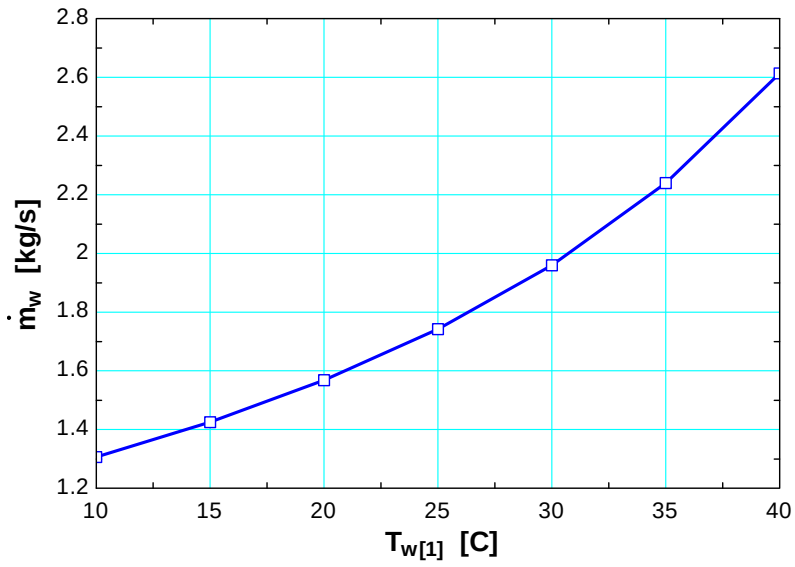
```
"Input Data"
T_w[1]=20 [C]
T_w[2]=70 [C]  "w: water"
m_dot_eg=2 [kg/s] "eg: ethylene glycol"
T_eg[1]=80 [C]
T_eg[2]=40 [C]
C_p_w=4.18 [kJ/kg-K]
C_p_eg=2.56 [kJ/kg-K]

"Conservation of mass for the water: m_dot_w_in=m_dot_w_out=m_dot_w"
"Conservation of mass for the ethylene glycol: m_dot_eg_in=m_dot_eg_out=m_dot_eg"

"Conservation of Energy for steady-flow: neglect changes in KE and PE in each mass steam"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0 "Steady-flow requirement"
E_dot_in=m_dot_w*h_w[1] + m_dot_eg*h_eg[1]
E_dot_out=m_dot_w*h_w[2] + m_dot_eg*h_eg[2]
Q_exchanged =m_dot_eg*h_eg[1] - m_dot_eg*h_eg[2]

"Property data are given by:"
h_w[1] =C_p_w*T_w[1]  "liquid approximation applied for water and ethylene glycol"
h_w[2] =C_p_w*T_w[2]
h_eg[1] =C_p_eg*T_eg[1]
h_eg[2] =C_p_eg*T_eg[2]
```

$T_{w,1}$ [C]	m_w [kg/s]
10	1.307
15	1.425
20	1.568
25	1.742
30	1.96
35	2.24
40	2.613

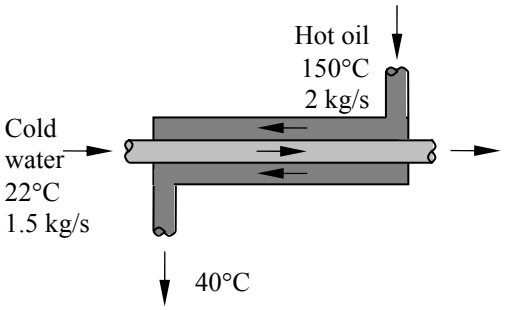


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5-84 Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer in the heat exchanger and the exit temperature of water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg.°C, respectively.



Analysis We take the oil tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the oil becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} = (2\text{ kg/s})(2.2\text{ kJ/kg.°C})(150°\text{C} - 40°\text{C}) = \mathbf{484\ kW}$$

Noting that the heat lost by the oil is gained by the water, the outlet temperature of the water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}_{\text{water}}c_p} = 22°\text{C} + \frac{484\text{ kJ/s}}{(1.5\text{ kg/s})(4.18\text{ kJ/kg.°C})} = \mathbf{99.2°C}$$

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5-85 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the exit temperature of hot water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

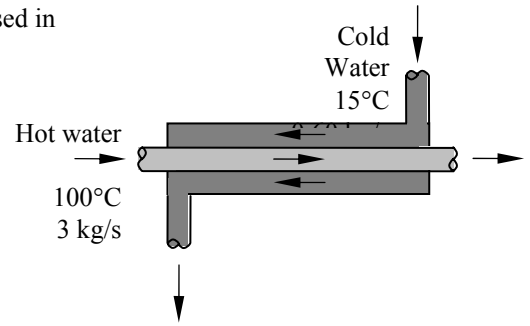
Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cold water}} = (0.60 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) = \mathbf{75.24\ kW}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{hot water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 100°\text{C} - \frac{75.24 \text{ kW}}{(3 \text{ kg/s})(4.19 \text{ kJ/kg.°C})} = \mathbf{94.0°C}$$

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5-86 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the outlet temperature of the air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

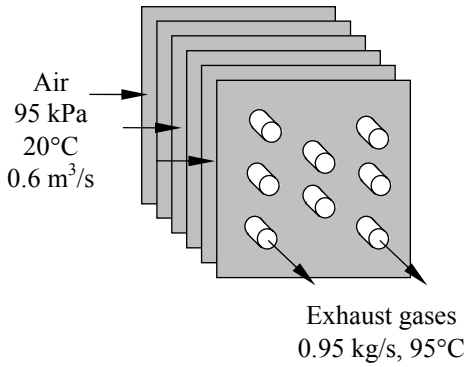
Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively.

Analysis We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Then the rate of heat transfer from the exhaust gases becomes

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{gas}} \\ &= (0.95\ \text{kg/s})(1.1\ \text{kJ/kg.°C})(160°\text{C} - 95°\text{C}) \\ &= \mathbf{67.93\ kW}\end{aligned}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95\ \text{kPa})(0.6\ \text{m}^3/\text{s})}{(0.287\ \text{kPa.m}^3/\text{kg.K}) \times 293\ \text{K}} = 0.6778\ \text{kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

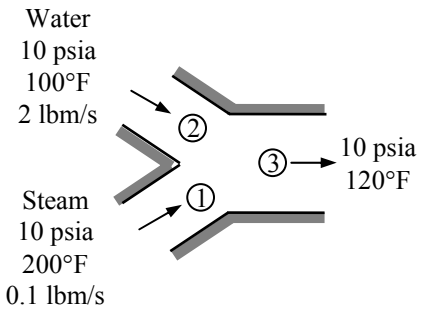
$$\dot{Q} = \dot{m}c_p(T_{\text{c,out}} - T_{\text{c,in}}) \longrightarrow T_{\text{c,out}} = T_{\text{c,in}} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{67.93\ \text{kW}}{(0.6778\ \text{kg/s})(1.005\ \text{kJ/kg.°C})} = \mathbf{120°C}$$

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5-87E An adiabatic open feedwater heater mixes steam with feedwater. The outlet mass flow rate and the outlet velocity are to be determined for two exit temperatures.

Assumptions Steady operating conditions exist.

Analysis From a mass balance



$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.1 + 2 = \mathbf{2.1\ lbm/s}$$

The specific volume at the exit is (Table A-4E)

$$\left.\begin{array}{l} P_3 = 10\ \text{psia} \\ T_3 = 120°\text{F} \end{array}\right\} \nu_3 \cong \nu_{f\,@120°\text{F}} = 0.01620\ \text{ft}^3/\text{lbm}$$

The exit velocity is then

$$V_3 = \frac{\dot{m}_3 \nu_3}{A_3} = \frac{4\dot{m}_3 \nu_3}{\pi D^2}$$

$$= \frac{4(2.1\ \text{lbm/s})(0.01620\ \text{ft}^3/\text{lbm})}{\pi(0.5\ \text{ft})^2}$$

$$= \mathbf{0.1733\ ft/s}$$

When the temperature at the exit is 180°F, we have

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.1 + 2 = \mathbf{2.1\ lbm/s}$$

$$\left.\begin{array}{l} P_3 = 10\ \text{psia} \\ T_3 = 180°\text{F} \end{array}\right\} \nu_3 \cong \nu_{f\,@180°\text{F}} = 0.01651\ \text{ft}^3/\text{lbm}$$

$$V_3 = \frac{\dot{m}_3 \nu_3}{A_3} = \frac{4\dot{m}_3 \nu_3}{\pi D^2} = \frac{4(2.1\ \text{lbm/s})(0.01651\ \text{ft}^3/\text{lbm})}{\pi(0.5\ \text{ft})^2} = \mathbf{0.1766\ ft/s}$$

The mass flow rate at the exit is same while the exit velocity slightly increases when the exit temperature is 180°F instead of 120°F.

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5-88E Air is heated in a steam heating system. For specified flow rates, the volume flow rate of air at the inlet is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm.R (Table A-1E). The constant pressure specific heat of air is c_p = 0.240 Btu/lbm·°F (Table A-2E). The enthalpies of steam at the inlet and the exit states are (Tables A-4E through A-6E)

$$\left.\begin{matrix} P_3 = 30 \text{ psia} \\ T_3 = 400°\text{F} \end{matrix}\right\} h_3 = 1237.9 \text{ Btu/lbm}$$

$$\left.\begin{matrix} P_4 = 25 \text{ psia} \\ T_4 = 212°\text{F} \end{matrix}\right\} h_4 \cong h_{f@212°\text{F}} = 180.21 \text{ Btu/lbm}$$

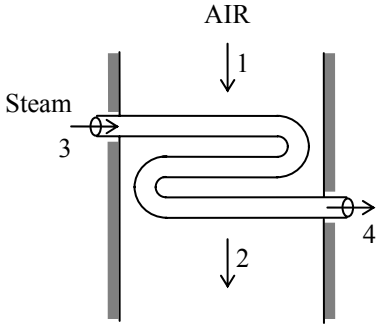
Analysis We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_a \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$



Energy balance (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_a (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$

Solving for $\dot{m}_a$:

$$\dot{m}_a = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \frac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$$

Substituting,

$$\dot{m}_a = \frac{(1237.9 - 180.21)\text{Btu/lbm}}{(0.240 \text{ Btu/lbm} \cdot °\text{F})(130 - 80)°\text{F}} (15 \text{ lbm/min}) = 1322 \text{ lbm/min} = 22.04 \text{ lbm/s}$$

Also,

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(540 \text{ R})}{14.7 \text{ psia}} = 13.61 \text{ ft}^3/\text{lbm}$$

Then the volume flow rate of air at the inlet becomes

$$\dot{V}_1 = \dot{m}_a \nu_1 = (22.04 \text{ lbm/s})(13.61 \text{ ft}^3/\text{lbm}) = \mathbf{300 \text{ ft}^3/\text{s}}$$

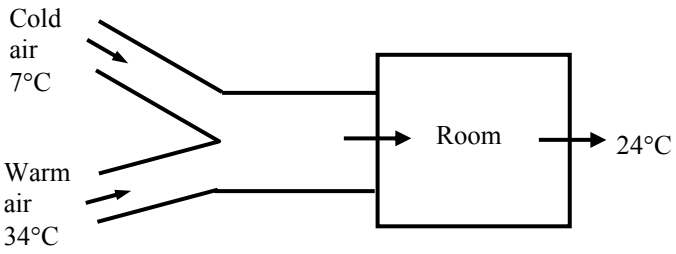
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5-89 Two streams of cold and warm air are mixed in a chamber. If the ratio of hot to cold air is 1.6, the mixture temperature and the rate of heat gain of the room are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K. The enthalpies of air are obtained from air table (Table A-17) as

$$h_1 = h_{@280\text{ K}} = 280.13 \text{ kJ/kg}$$
$$h_2 = h_{@307\text{ K}} = 307.23 \text{ kJ/kg}$$
$$h_{\text{room}} = h_{@297\text{ K}} = 297.18 \text{ kJ/kg}$$



Analysis (*a*) We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + 1.6\dot{m}_1 = \dot{m}_3 = 2.6\dot{m}_1 \quad \text{since} \quad \dot{m}_2 = 1.6\dot{m}_1$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two gives $\dot{m}_1 h_1 + 2.2\dot{m}_1 h_2 = 3.2\dot{m}_1 h_3$ *or* $h_3 = (h_1 + 2.2h_2)/3.2$

Substituting,

$$h_3 = (280.13 + 2.2 \times 307.23)/3.2 = 298.76 \text{ kJ/kg}$$

From air table at this enthalpy, the mixture temperature is

$$T_3 = T_{@\, h = 298.76 \text{ kJ/kg}} = 298.6 \text{ K} = \mathbf{25.6°C}$$

(*b*) The mass flow rates are determined as follows

$$v_1 = \frac{RT_1}{P} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(7 + 273 \text{ K})}{105 \text{ kPa}} = 0.7654 \text{ m}^3/\text{kg}$$
$$\dot{m}_1 = \frac{\dot{V}_1}{v_1} = \frac{0.75 \text{ m}^3/\text{s}}{0.7654 \text{ m}^3/\text{kg}} = 0.9799 \text{ kg/s}$$
$$\dot{m}_3 = 3.2\dot{m}_1 = 3.2(0.9799 \text{ kg/s}) = 3.136 \text{ kg/s}$$

The rate of heat gain of the room is determined from

$$\dot{Q}_{\text{gain}} = \dot{m}_3 (h_{\text{room}} - h_3) = (3.136 \text{ kg/s})(297.18 - 298.76) \text{ kJ/kg} = \mathbf{-4.93 \text{ kW}}$$

The negative sign indicates that the room actually loses heat at a rate of 4.93 kW.

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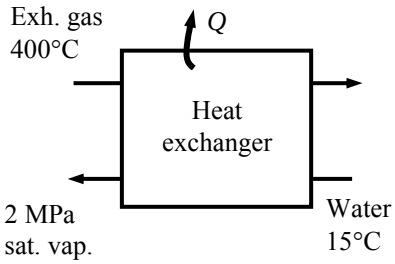
5-90 A heat exchanger that is not insulated is used to produce steam from the heat given up by the exhaust gases of an internal combustion engine. The temperature of exhaust gases at the heat exchanger exit and the rate of heat transfer to the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Exhaust gases are assumed to have air properties with constant specific heats.

Properties The constant pressure specific heat of the exhaust gases is taken to be $c_p = 1.045$ kJ/kg·°C (Table A-2). The inlet and exit enthalpies of water are (Tables A-4 and A-5)

$$\left.\begin{matrix} T_{w,in} = 15°C \\ x = 0\,(\text{sat. liq.}) \end{matrix}\right\} h_{w,in} = 62.98 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_{w,out} = 2 \text{ MPa} \\ x = 1\,(\text{sat. vap.}) \end{matrix}\right\} h_{w,out} = 2798.3 \text{ kJ/kg}$$



Analysis We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (*for each fluid stream*):

$$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{m}_{in} = \dot{m}_{out}$$

Energy balance (for the entire heat exchanger):

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{exh} h_{exh,in} + \dot{m}_w h_{w,in} = \dot{m}_{exh} h_{exh,out} + \dot{m}_w h_{w,out} + \dot{Q}_{out} \quad (\text{since } \dot{W} = \Delta ke \cong \Delta pe \cong 0)$$

or $$\dot{m}_{exh} c_p T_{exh,in} + \dot{m}_w h_{w,in} = \dot{m}_{exh} c_p T_{exh,out} + \dot{m}_w h_{w,out} + \dot{Q}_{out}$$

Noting that the mass flow rate of exhaust gases is 15 times that of the water, substituting gives

$$\begin{aligned} &15\dot{m}_w (1.045 \text{ kJ/kg.°C})(400°C) + \dot{m}_w (62.98 \text{ kJ/kg}) \\ &\qquad = 15\dot{m}_w (1.045 \text{ kJ/kg.°C}) T_{exh,out} + \dot{m}_w (2798.3 \text{ kJ/kg}) + \dot{Q}_{out} \end{aligned} \quad (1)$$

The heat given up by the exhaust gases and heat picked up by the water are

$$\dot{Q}_{exh} = \dot{m}_{exh} c_p (T_{exh,in} - T_{exh,out}) = 15\dot{m}_w (1.045 \text{ kJ/kg.°C})(400 - T_{exh,out})°C \quad (2)$$

$$\dot{Q}_w = \dot{m}_w (h_{w,out} - h_{w,in}) = \dot{m}_w (2798.3 - 62.98) \text{kJ/kg} \quad (3)$$

The heat loss is

$$\dot{Q}_{out} = f_{\text{heat loss}} \dot{Q}_{exh} = 0.1 \dot{Q}_{exh} \quad (4)$$

The solution may be obtained by a trial-error approach. Or, solving the above equations simultaneously using EES software, we obtain

$$T_{exh,out} = \mathbf{206.1°C}, \dot{Q}_w = \mathbf{97.26 \text{ kW}}, \ \dot{m}_w = 0.03556 \text{ kg/s}, \ \dot{m}_{exh} = 0.5333 \text{ kg/s}$$

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5-91 A chilled-water heat-exchange unit is designed to cool air by water. The maximum water outlet temperature is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2*a*). The specific heat of water is 4.18 kJ/kg·K (Table A-3).

Analysis The water temperature at the heat exchanger exit will be maximum when all the heat released by the air is picked up by the water. First, the inlet specific volume and the mass flow rate of air are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303\ \text{K})}{100\ \text{kPa}} = 0.8696\ \text{m}^3/\text{kg}$$
$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{5\ \text{m}^3/\text{s}}{0.8696\ \text{m}^3/\text{kg}} = 5.750\ \text{kg/s}$$

We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_3 = \dot{m}_a \text{ and } \dot{m}_2 = \dot{m}_4 = \dot{m}_w$$

Energy balance (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two,

$$\dot{m}_a (h_1 - h_3) = \dot{m}_w (h_4 - h_2)$$
$$\dot{m}_a c_{p,a} (T_1 - T_3) = \dot{m}_w c_{p,w} (T_4 - T_2)$$

Solving for the exit temperature of water,

$$T_4 = T_2 + \frac{\dot{m}_a c_{p,a} (T_1 - T_3)}{\dot{m}_w c_{p,w}} = 8°\text{C} + \frac{(5.750\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot°\text{C})(30-18)°\text{C}}{(2\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})} = \mathbf{16.3°C}$$

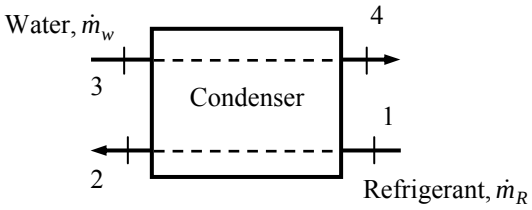
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5-92 Refrigerant-134a is condensed in a condenser by cooling water. The rate of heat transfer to the water and the mass flow rate of water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions between the condenser and the surroundings.

Analysis We take the condenser as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_R h_1 + \dot{m}_w h_3 = \dot{m}_R h_2 + \dot{m}_w h_4$$
$$\dot{m}_R (h_1 - h_2) = \dot{m}_w (h_4 - h_3) = \dot{m}_w c_p (T_4 - T_3)$$

If we take the refrigerant as the system, the energy balance can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_R h_1 = \dot{m}_R h_2 + \dot{Q}_{\text{out}}$$
$$\dot{Q}_{\text{out}} = \dot{m}_R (h_1 - h_2)$$

(a) The properties of refrigerant at the inlet and exit states of the condenser are (from Tables A-11 through A-13)

$$\left.\begin{aligned} P_1 &= 1200 \text{ kPa} \\ T_1 &= 85°\text{C} \end{aligned}\right\} h_1 = 316.73 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 1200 \text{ kPa} \\ T_2 &= T_{\text{sat @ 1200 kPa}} - \Delta T_{\text{subcool}} = 46.3 - 6.3 = 40°\text{C} \end{aligned}\right\} h_2 \cong h_{f\,@\,40°\text{C}} = 108.26 \text{ kJ/kg}$$

The rate of heat rejected to the water is

$$\dot{Q}_{\text{out}} = \dot{m}_R (h_1 - h_2) = (0.042 \text{ kg/s})(316.73 - 108.26)\text{kJ/kg} = 8.76 \text{ kW} = \mathbf{525 \text{ kJ/min}}$$

(b) The mass flow rate of water can be determined from the energy balance on the condenser:

$$\dot{Q}_{\text{out}} = \dot{m}_w c_p \Delta T_w$$
$$8.76 \text{ kW} = \dot{m}_w (4.18 \text{ kJ/kg} \cdot °\text{C})(12°\text{C})$$
$$\dot{m}_w = 0.175 \text{ kg/s} = \mathbf{10.5 \text{ kg/min}}$$

The specific heat of water is taken as 4.18 kJ/kg·°C (Table A-3).

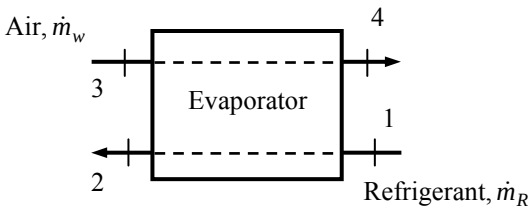
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5-93 Refrigerant-22 is evaporated in an evaporator by air. The rate of heat transfer from the air and the temperature change of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions between the evaporator and the surroundings.

Analysis We take the condenser as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_R h_1 + \dot{m}_a h_3 = \dot{m}_R h_2 + \dot{m}_a h_4$$
$$\dot{m}_R (h_2 - h_1) = \dot{m}_a (h_3 - h_4) = \dot{m}_a c_p \Delta T_a$$

If we take the refrigerant as the system, the energy balance can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_R h_1 + \dot{Q}_{\text{in}} = \dot{m}_R h_2$$
$$\dot{Q}_{\text{in}} = \dot{m}_R (h_2 - h_1)$$

(a) The mass flow rate of the refrigerant is

$$\dot{m}_R = \frac{\dot{V}_1}{v_1} = \frac{(2.25/3600)\,\text{m}^3/\text{s}}{0.0253\,\text{m}^3/\text{kg}} = 0.02472\,\text{kg/s}$$

The rate of heat absorbed from the air is

$$\dot{Q}_{\text{in}} = \dot{m}_R (h_2 - h_1) = (0.02472\,\text{kg/s})(398.0 - 220.2)\text{kJ/kg} = \mathbf{4.39\,kW}$$

(b) The temperature change of air can be determined from an energy balance on the evaporator:

$$\dot{Q}_L = \dot{m}_R (h_3 - h_2) = \dot{m}_a c_p (T_{a1} - T_{a2})$$
$$4.39\,\text{kW} = (0.5\,\text{kg/s})(1.005\,\text{kJ/kg}\cdot{}^\circ\text{C})\Delta T_a$$
$$\Delta T_a = \mathbf{8.7^\circ C}$$

The specific heat of air is taken as 1.005 kJ/kg·°C (Table A-2).

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5-94 Two mass streams of the same idela gas are mixed in a mixing chamber. Heat is transferred to the chamber. Three expressions as functions of other parameters are to be obtained.

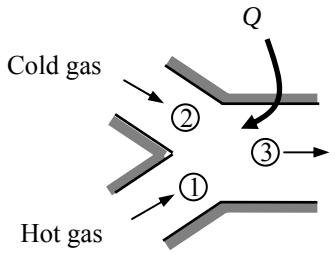
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis (a) We take the mixing device as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 + \dot{Q}_{\text{in}} = \dot{m}_3 h_3$$



From a mass balance,

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

Since $h = c_p T$,

Then

$$\dot{m}_1 c_p T_1 + \dot{m}_2 c_p T_2 + \dot{Q}_{\text{in}} = \dot{m}_3 c_p T_3$$

$$T_3 = \frac{\dot{m}_1}{\dot{m}_3} T_1 + \frac{\dot{m}_2}{\dot{m}_3} T_2 + \frac{\dot{Q}_{\text{in}}}{\dot{m}_3 c_p}$$

(b) Expression for volume flow rate:

$$\dot{\boldsymbol{V}}_3 = \dot{m}_3 \boldsymbol{v}_3 = \dot{m}_3 \frac{RT_3}{P_3}$$

$$\dot{\boldsymbol{V}}_3 = \frac{\dot{m}_3 R}{P_3}\left(\frac{\dot{m}_1}{\dot{m}_3} T_1 + \frac{\dot{m}_2}{\dot{m}_3} T_2 + \frac{\dot{Q}_{\text{in}}}{\dot{m}_3 c_p} \right)$$

$$P_3 = P_1 = P_2 = P$$

$$\dot{\boldsymbol{V}}_3 = \frac{\dot{m}_1 RT_1}{P_1} + \frac{\dot{m}_2 RT_2}{P_2} + \frac{R\dot{Q}_{\text{in}}}{P_3 c_p}$$

$$\dot{\boldsymbol{V}}_3 = \dot{\boldsymbol{V}}_1 + \dot{\boldsymbol{V}}_2 + \frac{R\dot{Q}_{\text{in}}}{Pc_p}$$

(c) If the process is adiabatic, then

$$\dot{\boldsymbol{V}}_3 = \dot{\boldsymbol{V}}_1 + \dot{\boldsymbol{V}}_2$$

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Pipe and duct Flow

5-95 Heat is supplied to the argon as it flows in a heater. The exit temperature of argon and the volume flow rate at the exit are to be determined.

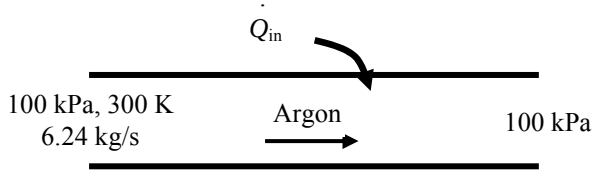
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of argon is 0.2081 kPa.m^3/kg.K. The constant pressure specific heat of air at room temperature is c_p = 0.5203 kJ/kg·°C (Table A-2*a*).

Analysis (a) We take the pipe(heater) in which the argon is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 + \dot{Q}_{\text{in}} = \dot{m}h_2$$
$$\dot{Q}_{\text{in}} = \dot{m}(h_2 - h_1)$$
$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Substituting and solving for the exit temperature,

$$T_2 = T_1 + \frac{\dot{Q}_{\text{in}}}{\dot{m}c_p} = 300 \text{ K} + \frac{150 \text{ kW}}{(6.24 \text{ kg/s})(0.5203 \text{ kJ/kg} \cdot \text{K})} = 346.2 \text{ K} = \mathbf{73.2°C}$$

(b) The exit specific volume and the volume flow rate are

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.2081 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(346.2 \text{ K})}{100 \text{ kPa}} = 0.7204 \text{ m}^3/\text{kg}$$
$$\dot{V}_2 = \dot{m}\nu_2 = (6.24 \text{ kg/s})(0.8266 \text{ m}^3/\text{kg}) = \mathbf{4.50 \text{ m}^3/\text{s}}$$

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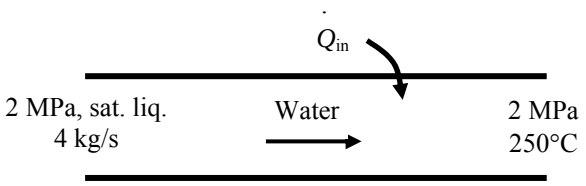
5-96 Saturated liquid water is heated in a steam boiler at a specified rate. The rate of heat transfer in the boiler is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which the water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 + \dot{Q}_{\text{in}} = \dot{m}h_2$$
$$\dot{Q}_{\text{in}} = \dot{m}(h_2 - h_1)$$



The enthalpies of water at the inlet and exit of the boiler are (Table A-5, A-6).

$$\left.\begin{matrix} P_1 = 2 \text{ MPa} \\ x = 0 \end{matrix}\right\} h_1 \cong h_{f\,@\,2\text{ MPa}} = 908.47 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 2 \text{ MPa} \\ T_2 = 250°\text{C} \end{matrix}\right\} h_2 = 2903.3 \text{ kJ/kg}$$

Substituting,

$$\dot{Q}_{\text{in}} = (4 \text{ kg/s})(2903.3 - 908.47)\text{kJ/kg} = \mathbf{7980\ kW}$$

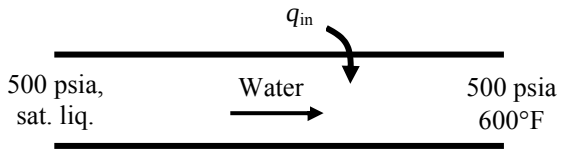
5-97E Saturated liquid water is heated in a steam boiler. The heat transfer per unit mass is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which the water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 + \dot{Q}_{\text{in}} = \dot{m}h_2$$
$$\dot{Q}_{\text{in}} = \dot{m}(h_2 - h_1)$$
$$q_{\text{in}} = h_2 - h_1$$



The enthalpies of water at the inlet and exit of the boiler are (Table A-5E, A-6E).

$$\left.\begin{matrix} P_1 = 500 \text{ psia} \\ x = 0 \end{matrix}\right\} h_1 \cong h_{f\,@\,500\text{ psia}} = 449.51 \text{ Btu/lbm}$$

$$\left.\begin{matrix} P_2 = 500 \text{ psia} \\ T_2 = 600°\text{F} \end{matrix}\right\} h_2 = 1298.6 \text{ Btu/lbm}$$

Substituting,

$$q_{\text{in}} = 1298.6 - 449.51 = \mathbf{849.1\ Btu/lbm}$$

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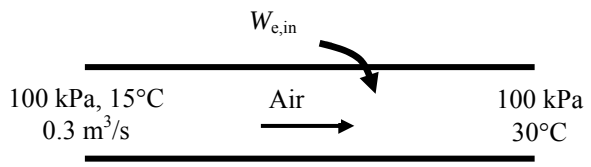
5-98 Air at a specified rate is heated by an electrical heater. The current is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The heat losses from the air is negligible.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2*a*).

Analysis We take the pipe in which the air is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 + \dot{W}_{\text{e,in}} = \dot{m}h_2$$
$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1)$$
$$\mathbf{V}I = \dot{m}c_p(T_2 - T_1)$$

The inlet specific volume and the mass flow rate of air are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{100\ \text{kPa}} = 0.8266\ \text{m}^3/\text{kg}$$
$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.3\ \text{m}^3/\text{s}}{0.8266\ \text{m}^3/\text{kg}} = 0.3629\ \text{kg/s}$$

Substituting into the energy balance equation and solving for the current gives

$$I = \frac{\dot{m}c_p(T_2 - T_1)}{\mathbf{V}} = \frac{(0.3629\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})(30-15)\text{K}}{110\ \text{V}}\left(\frac{1000\ \text{VI}}{1\ \text{kJ/s}}\right) = \mathbf{49.7\ Amperes}$$

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5-99E The cooling fan of a computer draws air, which is heated in the computer by absorbing the heat of PC circuits. The electrical power dissipated by the circuits is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** All the heat dissipated by the circuits are picked up by the air drawn by the fan.

Properties The gas constant of air is 0.3704 psia·ft^3/lbm·R (Table A-1E). The constant pressure specific heat of air at room temperature is c_p = 0.240 Btu/lbm·°F (Table A-2E*a*).

Analysis We take the pipe in which the air is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

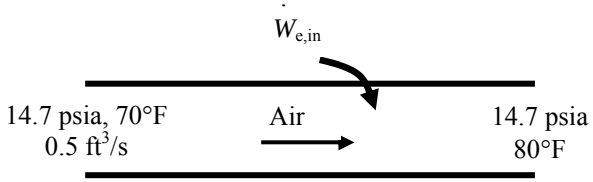
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{e,in}} = \dot{m}h_2$$

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1)$$

$$\dot{W}_{\text{e,in}} = \dot{m}c_p(T_2 - T_1)$$



The inlet specific volume and the mass flow rate of air are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530\ \text{R})}{14.7\ \text{psia}} = 13.35\ \text{ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.5\ \text{ft}^3/\text{s}}{13.35\ \text{ft}^3/\text{lbm}} = 0.03745\ \text{lbm/s}$$

Substituting,

$$\dot{W}_{\text{e,out}} = (0.03745\ \text{lbm/s})(0.240\ \text{Btu/lbm}\cdot\text{R})(80-70)\text{Btu/lbm}\left(\frac{1\ \text{kW}}{0.94782\ \text{Btu/s}}\right) = \mathbf{0.0948\ kW}$$

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5-100 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of T_{avg} = (45+60)/2 = 52.5°C = 325.5 K is c_p = 1.0065 kJ/kg.°C. The gas constant for air is R = 0.287 kJ/kg.K (Table A-2).

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the required mass flow rate of air to absorb heat at a rate of 60 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{60\text{ W}}{(1006.5\text{ J/kg.°C})(60 - 45)\text{°C}} = 0.00397\text{ kg/s} = 0.238\text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63\text{ kPa}}{(0.287\text{ kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972\text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.238\text{ kg/min}}{0.6972\text{ kg/m}^3} = \mathbf{0.341\,m^3/min}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \rightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.341\text{ m}^3\text{/min})}{\pi(110\text{ m/min})}} = 0.063\text{ m} = \mathbf{6.3\text{ cm}}$$

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5-101 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of T_{ave} = (45+60)/2 = 52.5°C is c_p = 1.0065 kJ/kg.°C The gas constant for air is R = 0.287 kJ/kg.K (Table A-2).

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the required mass flow rate of air to absorb heat at a rate of 100 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}})$$

$$\dot{m} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{100 \text{ W}}{(1006.5 \text{ J/kg.°C})(60 - 45)°\text{C}} = 0.006624 \text{ kg/s} = 0.397 \text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.57 \text{ m}^3\text{/min}}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.57 \text{ m}^3\text{/min})}{\pi(110 \text{ m/min})}} = 0.081 \text{ m} = \mathbf{8.1 \text{ cm}}$$

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5-102E Electronic devices mounted on a cold plate are cooled by water. The amount of heat generated by the electronic devices is to be determined.

Assumptions **1** Steady operating conditions exist. **2** About 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation. **3** Kinetic and potential energy changes are negligible.

Properties The properties of water at room temperature are $\rho = 62.1$ lbm/ft^3 and $c_p = 1.00$ Btu/lbm.°F (Table A-3E).

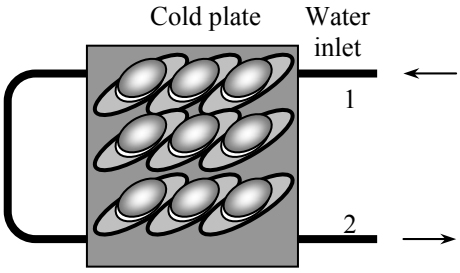
Analysis We take the tubes of the cold plate to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then mass flow rate of water and the rate of heat removal by the water are determined to be

$$\dot{m} = \rho AV = \rho \frac{\pi D^2}{4} V = (62.1\,\text{lbm/ft}^3)\frac{\pi(0.25/12\,\text{ft})^2}{4}(40\,\text{ft/min}) = 0.8483\,\text{lbm/min} = 50.9\,\text{lbm/h}$$

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}}) = (50.9\,\text{lbm/h})(1.00\,\text{Btu/lbm.°F})(105 - 70)°\text{F} = 1781\,\text{Btu/h}$$

which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

$$\dot{Q} = \frac{1781\,\text{Btu/h}}{0.85} = \mathbf{2096\ Btu/h} = \mathbf{614\ W}$$

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5-103 The components of an electronic device located in a horizontal duct of rectangular cross section are cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

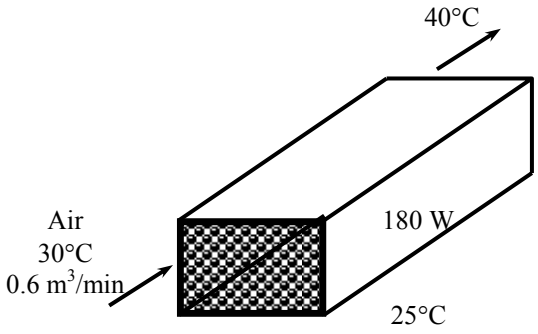
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible

Properties The gas constant of air is $R = 0.287$ kJ/kg.°C (Table A-1). The specific heat of air at room temperature is $c_p = 1.005$ kJ/kg.°C (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\,\text{kPa}}{(0.287\,\text{kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\,\text{kg/m}^3$$

$$\dot{m} = \rho \dot{V} = (1.165\,\text{kg/m}^3)(0.6\,\text{m}^3/\text{min}) = 0.700\,\text{kg/min}$$



We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{\text{air}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} = (0.700/60\,\text{kg/s})(1.005\,\text{kJ/kg.°C})(40-30)\text{°C} = 0.117\,\text{kW} = 117\,\text{W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63\ W}$$

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5-104 The components of an electronic device located in a horizontal duct of circular cross section is cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible

Properties The gas constant of air is $R = 0.287$ kJ/kg.°C (Table A-1). The specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\,\text{kPa}}{(0.287\,\text{kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\,\text{kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.165\,\text{kg/m}^3)(0.6\,\text{m}^3/\text{min}) = 0.700\,\text{kg/min}$$

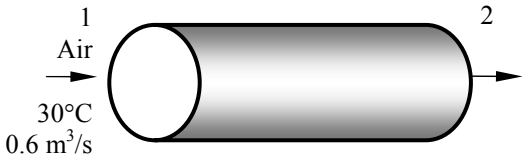
We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{\text{air}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} = (0.700/60\,\text{kg/s})(1.005\,\text{kJ/kg.°C})(40-30)°\text{C} = 0.117\,\text{kW} = 117\,\text{W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

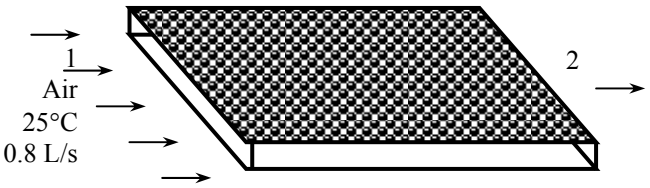
$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63\ W}$$

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5-105 Air enters a hollow-core printed circuit board. The exit temperature of the air is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** The local atmospheric pressure is 1 atm. **4** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is $R = 0.287$ kJ/kg.°C (Table A-1). The specific heat of air at room temperature is $c_p = 1.005$ kJ/kg.°C (Table A-2).



Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\,\text{kPa}}{(0.287\,\text{kPa.m}^3/\text{kg.K})(25+273)\text{K}} = 1.185\,\text{kg/m}^3$$

$$\dot{m} = \rho \dot{V} = (1.185\,\text{kg/m}^3)(0.0008\,\text{m}^3/\text{s}) = 0.0009477\,\text{kg/s}$$

We take the hollow core to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the exit temperature of air leaving the hollow core becomes

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{\dot{Q}_{\text{in}}}{\dot{m}c_p} = 25\,^\circ\text{C} + \frac{15\,\text{J/s}}{(0.0009477\,\text{kg/s})(1005\,\text{J/kg.°C})} = \mathbf{46.0°C}$$

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5-106 A computer is cooled by a fan blowing air through the case of the computer. The required flow rate of the air and the fraction of the temperature rise of air that is due to heat generated by the fan are to be determined.

Assumptions **1** Steady flow conditions exist. **2** Air is an ideal gas with constant specific heats. **3** The pressure of air is 1 atm. **4** Kinetic and potential energy changes are negligible

Properties The specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

Analysis (*a*) We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Noting that the fan power is 25 W and the 8 PCBs transfer a total of 80 W of heat to air, the mass flow rate of air is determined to be

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q}_{\text{in}} + \dot{W}_{\text{in}}}{c_p(T_e - T_i)} = \frac{(8 \times 10)\text{ W} + 25\text{ W}}{(1005\text{ J/kg.°C})(10°\text{C})} = \mathbf{0.0104\,kg/s}$$

(*b*) The fraction of temperature rise of air that is due to the heat generated by the fan and its motor can be determined from

$$\dot{Q} = \dot{m}c_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{25\text{ W}}{(0.0104\text{ kg/s})(1005\text{ J/kg.°C})} = 2.4°\text{C}$$

$$f = \frac{2.4°\text{C}}{10°\text{C}} = 0.24 = \mathbf{24\%}$$

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5-107 Hot water enters a pipe whose outer surface is exposed to cold air in a basement. The rate of heat loss from the water is to be determined.

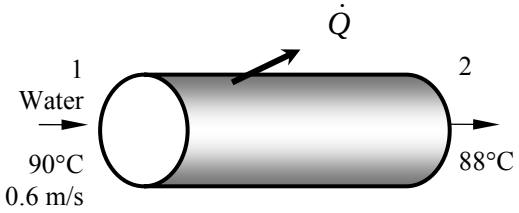
Assumptions **1** Steady flow conditions exist. **2** Water is an incompressible substance with constant specific heats. **3** The changes in kinetic and potential energies are negligible.

Properties The properties of water at the average temperature of (90+88)/2 = 89°C are $\rho = 965\ \text{kg/m}^3$ and $c_p = 4.21$ kJ/kg.°C (Table A-3).

Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c V = (965\ \text{kg/m}^3)\frac{\pi(0.025\ \text{m})^2}{4}(0.6\ \text{m/s}) = 0.2842\ \text{kg/s}$$

We take the section of the pipe in the basement to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the hot water to the surrounding air becomes

$$\dot{Q}_{\text{out}} = \dot{m}c_p[T_{\text{in}} - T_{\text{out}}]_{\text{water}} = (0.2842\ \text{kg/s})(4.21\ \text{kJ/kg.°C})(90 - 88)°\text{C} = \mathbf{2.39\ kW}$$

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5-108 Problem 5-107 is reconsidered. The effect of the inner pipe diameter on the rate of heat loss as the pipe diameter varies from 1.5 cm to 7.5 cm is to be investigated. The rate of heat loss is to be plotted against the diameter.

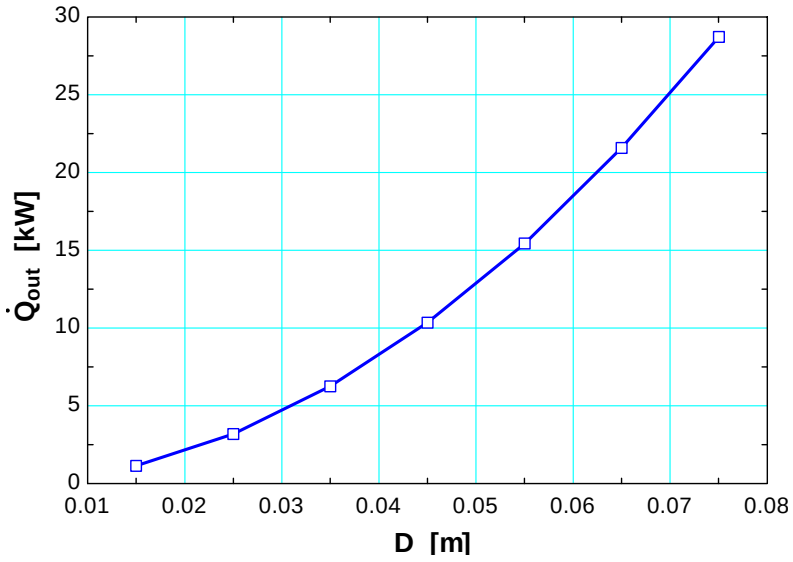
Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns:"
D = 0.025 [m]
rho = 965 [kg/m^3]
Vel = 0.6 [m/s]
T_1 = 90 [C]
T_2 = 88 [C]
c_p = 4.21 [kJ/kg-C]

"Analysis:"
"The mass flow rate of water is:"
Area = pi*D^2/4
m_dot = rho*Area*Vel

"We take the section of the pipe in the basement to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as"
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 "Steady-flow assumption"
E_dot_in = m_dot*h_in
E_dot_out =Q_dot_out+m_dot*h_out
h_in = c_p * T_1
h_out = c_p * T_2
```

D [m]	Q_{out} [kW]
0.015	1.149
0.025	3.191
0.035	6.254
0.045	10.34
0.055	15.44
0.065	21.57
0.075	28.72



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5-109 A room is to be heated by an electric resistance heater placed in a duct in the room. The power rating of the electric heater and the temperature rise of air as it passes through the heater are to be determined.

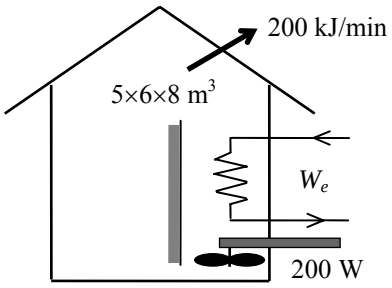
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the room.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 and c_v = 0.718 kJ/kg·K (Table A-2).

Analysis (*a*) The total mass of air in the room is

$$V = 5\times 6\times 8 \text{ m}^3 = 240 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(98 \text{ kPa})(240 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288 \text{ K})} = 284.6 \text{ kg}$$



We first take the *entire room* as our system, which is a closed system since no mass leaks in or out. The power rating of the electric heater is determined by applying the conservation of energy relation to this constant volume closed system:

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} + W_{\text{fan,in}} - Q_{out} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}\right) = mc_{v,\text{avg}}(T_2 - T_1)$$

Solving for the electrical work input gives

$$\begin{aligned}\dot{W}_{\text{e,in}} &= \dot{Q}_{\text{out}} - \dot{W}_{\text{fan,in}} + mc_v(T_2 - T_1)/\Delta t \\ &= (200/60 \text{ kJ/s}) - (0.2 \text{ kJ/s}) + (284.6 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(25-15)^\circ\text{C}/(15\times 60 \text{ s}) \\ &= \mathbf{5.40\ kW}\end{aligned}$$

(*b*) We now take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this adiabatic steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

$$\Delta T = T_2 - T_1 = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{\dot{m}c_p} = \frac{(5.40 + 0.2) \text{ kJ/s}}{(50/60 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot\text{K})} = \mathbf{6.7^\circ C}$$

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5-110E Water is heated in a parabolic solar collector. The required length of parabolic collector is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat loss from the tube is negligible so that the entire solar energy incident on the tube is transferred to the water. **3** Kinetic and potential energy changes are negligible

Properties The specific heat of water at room temperature is c_p = 1.00 Btu/lbm.°F (Table A-3E).

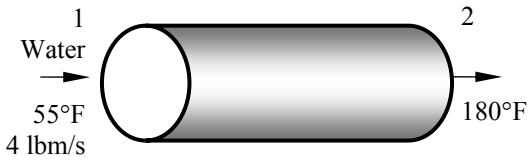
Analysis We take the thin aluminum tube to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$



Then the total rate of heat transfer to the water flowing through the tube becomes

$$\dot{Q}_{\text{total}} = \dot{m}c_p (T_e - T_i) = (4 \text{ lbm/s})(1.00 \text{ Btu/lbm.°F})(180 - 55)°\text{F} = 500 \text{ Btu/s} = 1{,}800{,}000 \text{ Btu/h}$$

The length of the tube required is

$$L = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{1{,}800{,}000 \text{ Btu/h}}{400 \text{ Btu/h.ft}} = \mathbf{4500 \text{ ft}}$$

5-111 A house is heated by an electric resistance heater placed in a duct. The power rating of the electric heater is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·K (Table A-2)

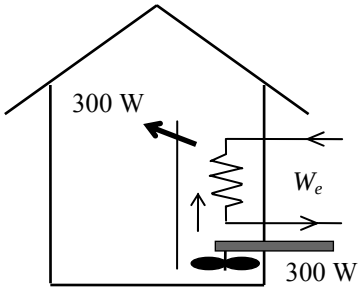
Analysis We take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} = \dot{Q}_{\text{out}} + \dot{m}(h_2 - h_1) = \dot{Q}_{\text{out}} + \dot{m}c_p (T_2 - T_1)$$



Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} + \dot{m}c_p \Delta T - \dot{W}_{\text{fan,in}} = (0.3 \text{ kJ/s}) + (0.6 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot °\text{C})(7°\text{C}) - 0.3 \text{ kW} = \mathbf{4.22 \text{ kW}}$$

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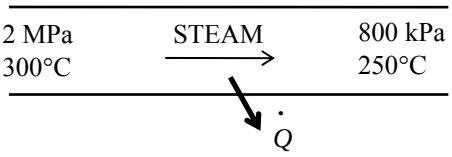
5-112 Steam pipes pass through an unheated area, and the temperature of steam drops as a result of heat losses. The mass flow rate of steam and the rate of heat loss from are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

Properties From the steam tables (Table A-6),

$$\left.\begin{array}{l} P_1 = 2\ \text{MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} \nu_1 = 0.12551\ \text{m}^3/\text{kg} \\ h_1 = 3024.2\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ T_2 = 250°\text{C} \end{array}\right\} h_2 = 2950.4\ \text{kJ/kg}$$



Analysis (*a*) The mass flow rate of steam is determined directly from

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 = \frac{1}{0.12551\ \text{m}^3/\text{kg}} \left[\pi (0.06\ \text{m})^2\right](3\ \text{m/s}) = \mathbf{0.270\ kg/s}$$

(*b*) We take the *steam pipe* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \nearrow^{0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2)$$

Substituting, the rate of heat loss is determined to be

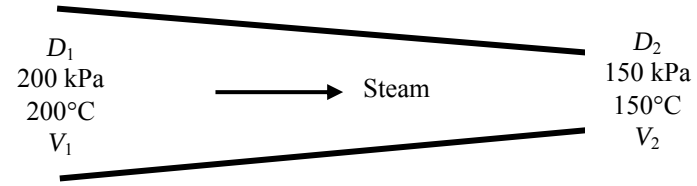
$$\dot{Q}_{\text{loss}} = (0.270\ \text{kg/s})(3024.2 - 2950.4)\ \text{kJ/kg} = \mathbf{19.9\ kJ/s}$$

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5-113 Steam flows through a non-constant cross-section pipe. The inlet and exit velocities of the steam are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Analysis We take the pipe as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as



Mass balance:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{in} = \dot{m}_{out} \longrightarrow A_1 \frac{V_1}{\nu_1} = A_1 \frac{V_1}{\nu_1} \longrightarrow \frac{\pi D_1^2}{4} \frac{V_1}{\nu_1} = \frac{\pi D_2^2}{4} \frac{V_2}{\nu_2}$$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{matrix} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{matrix}\right\} \begin{matrix} \nu_1 = 1.0805 \text{ m}^3/\text{kg} \\ h_1 = 2870.7 \text{ kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 150 \text{ kPa} \\ T_1 = 150°\text{C} \end{matrix}\right\} \begin{matrix} \nu_2 = 1.2855 \text{ m}^3/\text{kg} \\ h_2 = 2772.9 \text{ kJ/kg} \end{matrix}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting,

$$\frac{\pi (1.8 \text{ m})^2}{4} \frac{V_1}{(1.0805 \text{ m}^3/\text{kg})} = \frac{\pi (1.0 \text{ m})^2}{4} \frac{V_2}{(1.2855 \text{ m}^3/\text{kg})} \qquad (1)$$

$$2870.7 \text{ kJ/kg} + \frac{V_1^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 2772.9 \text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) \qquad (2)$$

There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

$V_1 =$ **118.8 m/s**

$V_2 =$ **458.0 m/s**

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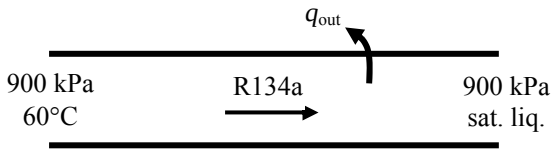
5-114 R-134a is condensed in a condenser. The heat transfer per unit mass is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which R-134a is condensed as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 = \dot{m}h_2 + \dot{Q}_{\text{out}}$$
$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2)$$
$$q_{\text{out}} = h_1 - h_2$$



The enthalpies of R-134a at the inlet and exit of the condenser are (Table A-12, A-13).

$$\left.\begin{array}{l} P_1 = 900 \text{ kPa} \\ T_1 = 60°\text{C} \end{array}\right\} h_1 = 295.13 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ x = 0 \end{array}\right\} h_2 = h_{f\,@\,900\text{ kPa}} = 101.61 \text{ kJ/kg}$$

Substituting,

$$q_{\text{out}} = 295.13 - 101.61 = \mathbf{193.5\ kJ/kg}$$

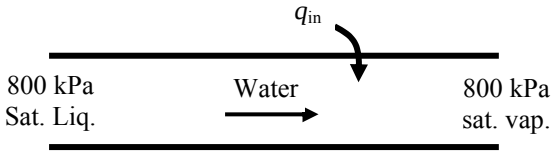
5-115 Water is heated at constant pressure so that it changes a state from saturated liquid to saturated vapor. The heat transfer per unit mass is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Analysis We take the pipe in which water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 + \dot{Q}_{\text{in}} = \dot{m}h_2$$
$$\dot{Q}_{\text{in}} = \dot{m}(h_2 - h_1)$$
$$q_{\text{in}} = h_2 - h_1 = h_{fg}$$



where $h_{fg\,@\,800\text{ kPa}} = 2047.5 \text{ kJ/kg}$ (Table A-5)

Thus,

$$q_{\text{in}} = \mathbf{2047.5\ kJ/kg}$$

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5-116 Electrical work is supplied to the air as it flows in a hair dryer. The mass flow rate of air and the volume flow rate at the exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the dryer is negligible.

Properties The gas constant of argon is 0.287 kPa.m^3/kg.K. The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2*a*).

Analysis (a) We take the pipe as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

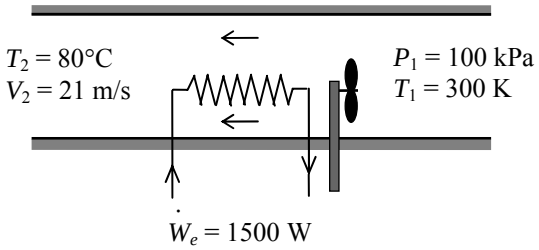
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) + \dot{W}_{\text{in}} = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$\dot{W}_{\text{in}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$\dot{W}_{\text{in}} = \dot{m}\left(c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}\right)$$



Substituting and solving for the mass flow rate,

$$\dot{m} = \frac{\dot{W}_{\text{in}}}{c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}}$$

$$= \frac{1.50\,\text{kW}}{(1.005\,\text{kJ/kg}\cdot\text{K})(353 - 300)\text{K} + \frac{(21\,\text{m/s})^2 - 0}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right)}$$

$$= \mathbf{0.0280\ kg/s}$$

(b) The exit specific volume and the volume flow rate are

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\,\text{K})}{100\ \text{kPa}} = 1.013\,\text{m}^3/\text{kg}$$

$$\dot{V}_2 = \dot{m}\nu_2 = (0.02793\,\text{kg/s})(1.013\,\text{m}^3/\text{kg}) = \mathbf{0.0284\ m^3/s}$$

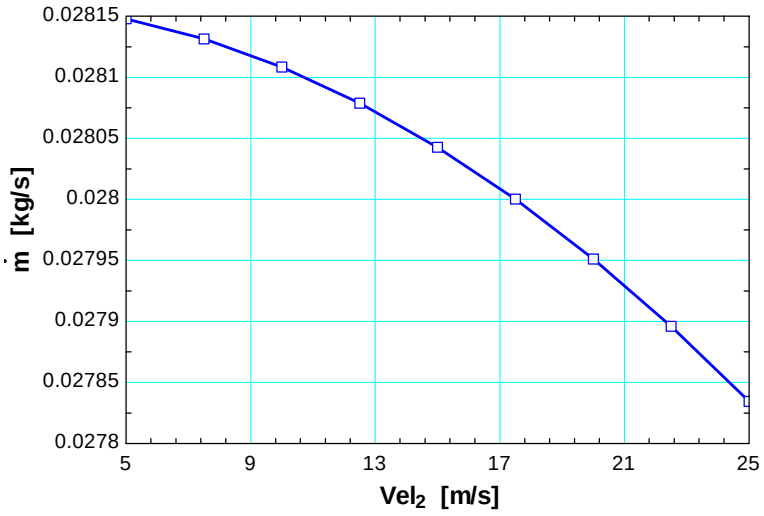
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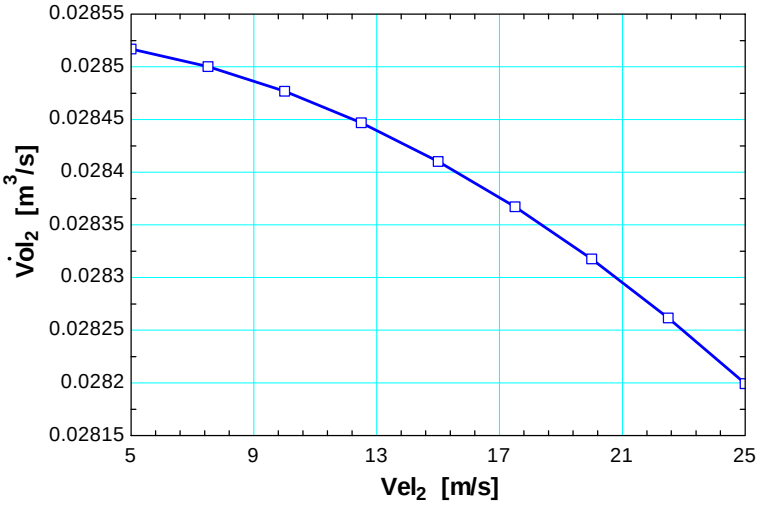
5-117 Problem 5-116 is reconsidered. The effect of the exit velocity on the mass flow rate and the exit volume flow rate is to be investiagted.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
T1=300 [K]
P=100 [kPa]
Vel_1=0 [m/s]
W_dot_e_in=1.5 [kW]
T2=(80+273) [K]
Vel_2=21 [m/s]
"Properties"
c_p=1.005 [kJ/kg-K]
R=0.287 [kJ/kg-K]
"Analysis"
W_dot_e_in=m_dot*c_p*(T2-T1)+m_dot*(Vel_2^2-Vel_1^2)*Convert(m^2/s^2,kJ/kg)  "energy balance on hair dryer"
v2=(R*T2)/P
Vol_dot_2=m_dot*v2
```

Vel_2 [m/s]	m [kg/s]	Vol_2 [m^3/s]
5	0.02815	0.02852
7.5	0.02813	0.0285
10	0.02811	0.02848
12.5	0.02808	0.02845
15	0.02804	0.02841
17.5	0.028	0.02837
20	0.02795	0.02832
22.5	0.0279	0.02826
25	0.02783	0.0282





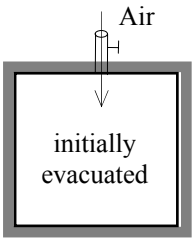
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Charging and Discharging Processes

5-118 An insulated rigid tank is evacuated. A valve is opened, and air is allowed to fill the tank until mechanical equilibrium is established. The final temperature in the tank is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The device is adiabatic and thus heat transfer is negligible.

Properties The specific heat ratio for air at room temperature is $k = 1.4$ (Table A-2).



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = m_2 u_2 \ (\text{since } Q \cong W \cong E_{\text{out}} = E_{\text{initial}} = ke \cong pe \cong 0)$$

Combining the two balances:

$$u_2 = h_i \rightarrow c_v T_2 = c_p T_i \rightarrow T_2 = (c_p / c_v) T_i = kT_i$$

Substituting,

$$T_2 = 1.4 \times 290 \text{ K} = 406 \text{ K} = \mathbf{133^\circ C}$$

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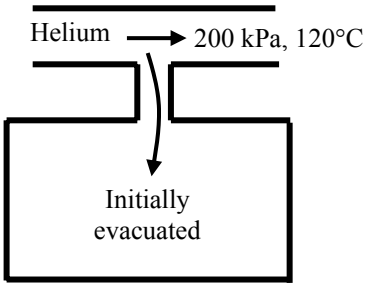
5-119 Helium flows from a supply line to an initially evacuated tank. The flow work of the helium in the supply line and the final temperature of the helium in the tank are to be determined.

Properties The properties of helium are R = 2.0769 kJ/kg.K, c_p = 5.1926 kJ/kg.K, c_v = 3.1156 kJ/kg.K (Table A-2a).

Analysis The flow work is determined from its definition but we first determine the specific volume

$$v = \frac{RT_{\text{line}}}{P} = \frac{(2.0769\ \text{kJ/kg.K})(120 + 273\ \text{K})}{(200\ \text{kPa})} = 4.0811\ \text{m}^3/\text{kg}$$

$$w_{\text{flow}} = Pv = (200\ \text{kPa})(4.0811\ \text{m}^3/\text{kg}) = \mathbf{816.2\ kJ/kg}$$



Noting that the flow work in the supply line is converted to sensible internal energy in the tank, the final helium temperature in the tank is determined as follows

$$u_{\text{tank}} = h_{\text{line}}$$

$$h_{\text{line}} = c_p T_{\text{line}} = (5.1926\ \text{kJ/kg.K})(120 + 273\ \text{K}) = 2040.7\ \text{kJ/kg}$$

$$u_{\text{-tank}} = c_v T_{\text{tank}} \longrightarrow 2040.7\ \text{kJ/kg} = (3.1156\ \text{kJ/kg.K}) T_{\text{tank}} \longrightarrow T_{\text{tank}} = \mathbf{655.0\ K}$$

Alternative Solution: Noting the definition of specific heat ratio, the final temperature in the tank can also be determined from

$$T_{\text{tank}} = kT_{\text{line}} = 1.667(120 + 273\ \text{K}) = \mathbf{655.1K}$$

which is practically the same result.

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5-120 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

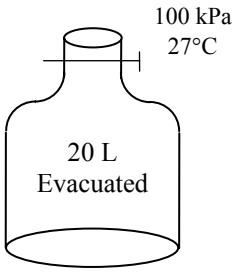
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1).

Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = ke \cong pe \cong 0)$$

Combining the two balances:

$$Q_{\text{in}} = m_2(u_2 - h_i)$$

where

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(100 \text{ kPa})(0.020 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})} = 0.02323 \text{ kg}$$

$$T_i = T_2 = 300 \text{ K} \xrightarrow{\text{Table A-17}} \begin{matrix} h_i = 300.19 \text{ kJ/kg} \\ u_2 = 214.07 \text{ kJ/kg} \end{matrix}$$

Substituting,

$$Q_{\text{in}} = (0.02323 \text{ kg})(214.07 - 300.19) \text{ kJ/kg} = -2.0 \text{ kJ}$$

or

$$Q_{\text{out}} = \mathbf{2.0 \text{ kJ}}$$

Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

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5-121 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until mechanical equilibrium is established. The mass of air that entered and the amount of heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the tank (will be verified).

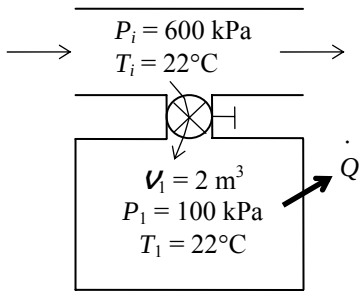
Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The properties of air are (Table A-17)

$$T_i = 295 \text{ K} \longrightarrow h_i = 295.17 \text{ kJ/kg}$$
$$T_1 = 295 \text{ K} \longrightarrow u_1 = 210.49 \text{ kJ/kg}$$
$$T_2 = 350 \text{ K} \longrightarrow u_2 = 250.02 \text{ kJ/kg}$$

Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(295 \text{ K})} = 2.362 \text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(600 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(350 \text{ K})} = 11.946 \text{ kg}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = 11.946 - 2.362 = \mathbf{9.584 \text{ kg}}$$

(*b*) The heat transfer during this process is determined from

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(9.584 \text{ kg})(295.17 \text{ kJ/kg}) + (11.946 \text{ kg})(250.02 \text{ kJ/kg}) - (2.362 \text{ kg})(210.49 \text{ kJ/kg}) \\ &= -339 \text{ kJ} \quad \rightarrow \quad Q_{\text{out}} = \mathbf{339 \text{ kJ}} \end{aligned}$$

Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reversed the direction.

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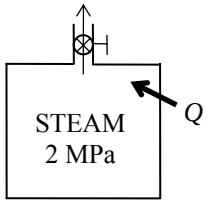
5-122 A rigid tank initially contains superheated steam. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until the temperature rises to 500°C. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the steam leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$\left.\begin{matrix} P_1 = 2\ \text{MPa} \\ T_1 = 300°\text{C} \end{matrix}\right\} \begin{matrix} v_1 = 0.12551\ \text{m}^3/\text{kg} \\ u_1 = 2773.2\ \text{kJ/kg}, \quad h_1 = 3024.2\ \text{kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 2\ \text{MPa} \\ T_2 = 500°\text{C} \end{matrix}\right\} \begin{matrix} v_2 = 0.17568\ \text{m}^3/\text{kg} \\ u_2 = 3116.9\ \text{kJ/kg}, \quad h_2 = 3468.3\ \text{kJ/kg} \end{matrix}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\begin{matrix}\text{Net energy transfer} \\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta E_{\text{system}}}_{\begin{matrix}\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}\end{matrix}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong ke \cong pe \cong 0)$$

The state and thus the enthalpy of the steam leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values. Thus,

$$h_e \cong \frac{h_1 + h_2}{2} = \frac{3024.2 + 3468.3\ \text{kJ/kg}}{2} = 3246.2\ \text{kJ/kg}$$

The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.2\ \text{m}^3}{0.12551\ \text{m}^3/\text{kg}} = 1.594\ \text{kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.2\ \text{m}^3}{0.17568\ \text{m}^3/\text{kg}} = 1.138\text{kg}$$

Then from the mass and energy balance relations,

$$m_e = m_1 - m_2 = 1.594 - 1.138 = 0.456\ \text{kg}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (0.456\ \text{kg})(3246.2\ \text{kJ/kg}) + (1.138\ \text{kg})(3116.9\ \text{kJ/kg}) - (1.594\ \text{kg})(2773.2\ \text{kJ/kg}) \\ &= \mathbf{606.8\ kJ} \end{aligned}$$

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5-123 A cylinder initially contains saturated liquid-vapor mixture of water. The cylinder is connected to a supply line, and the steam is allowed to enter the cylinder until all the liquid is vaporized. The final temperature in the cylinder and the mass of the steam that entered are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The device is insulated and thus heat transfer is negligible.

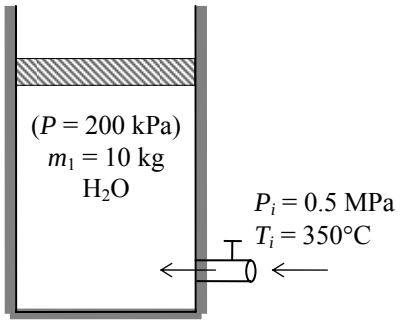
Properties The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{matrix} P_1 = 200 \text{ kPa} \\ x_1 = 0.6 \end{matrix}\right\} h_1 = h_f + x_1 h_{fg}$$

$$= 504.71 + 0.6 \times 2201.6 = 1825.6 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{matrix}\right\} h_2 = h_{g@200 \text{ kPa}} = 2706.3 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_i = 0.5 \text{ MPa} \\ T_i = 350°\text{C} \end{matrix}\right\} h_i = 3168.1 \text{ kJ/kg}$$



Analysis (*a*) The cylinder contains saturated vapor at the final state at a pressure of 200 kPa, thus the final temperature in the cylinder must be

$$T_2 = T_{\text{sat @ 200 kPa}} = \mathbf{120.2°C}$$

(*b*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \text{ (since } Q \cong ke \cong pe \cong 0)$$

Combining the two relations gives

$$0 = W_{\text{b,out}} - (m_2 - m_1) h_i + m_2 u_2 - m_1 u_1$$

or,

$$0 = -(m_2 - m_1) h_i + m_2 h_2 - m_1 h_1$$

since the boundary work and ΔU combine into ΔH for constant pressure expansion and compression processes. Solving for m_2 and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3168.1 - 1825.6) \text{ kJ/kg}}{(3168.1 - 2706.3) \text{ kJ/kg}} (10 \text{ kg}) = 29.07 \text{ kg}$$

Thus,

$$m_i = m_2 - m_1 = 29.07 - 10 = \mathbf{19.07 \text{ kg}}$$

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5-124E A scuba diver's air tank is to be filled with air from a compressed air line. The temperature and mass in the tank at the final state are to be determined.

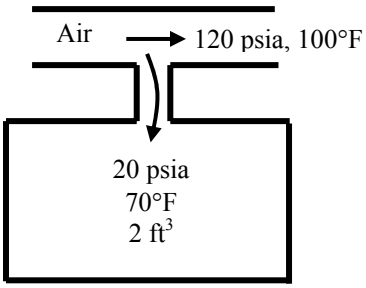
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The gas constant of air is 0.3704 psia·ft^3/lbm·R (Table A-1E). The specific heats of air at room temperature are c_p = 0.240 Btu/lbm·R and c_v = 0.171 Btu/lbm·R (Table A-2E*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$m_i h_i = m_2 u_2 - m_1 u_1$$

$$m_i c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

Combining the two balances:

$$(m_2 - m_1) c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{R T_1} = \frac{(20 \text{ psia})(2 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(70 + 460 \text{ R})} = 0.2038 \text{ lbm}$$

$$m_2 = \frac{P_2 V}{R T_2} = \frac{(120 \text{ psia})(2 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}) T_2} = \frac{647.9}{T_2}$$

Substituting,

$$\left(\frac{647.9}{T_2} - 0.2038 \right)(0.24)(560) = \frac{647.9}{T_2}(0.171) T_2 - (0.2038)(0.171)(530)$$

whose solution is

$$T_2 = \mathbf{727.4 \text{ R}} = \mathbf{267.4°F}$$

The final mass is then

$$m_2 = \frac{647.9}{T_2} = \frac{647.9}{727.4} = \mathbf{0.890 \text{ lbm}}$$

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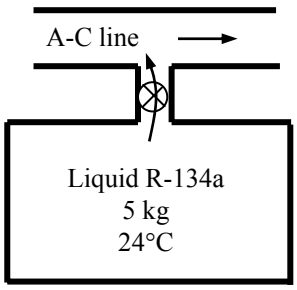
5-125 R-134a from a tank is discharged to an air-conditioning line in an isothermal process. The final quality of the R-134a in the tank and the total heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the exit remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$\begin{aligned} m_{\text{in}} - m_{\text{out}} &= \Delta m_{\text{system}} \\ -m_e &= m_2 - m_1 \\ m_e &= m_1 - m_2 \end{aligned}$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$\begin{aligned} Q_{\text{in}} - m_e h_e &= m_2 u_2 - m_1 u_1 \\ Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + m_e h_e \end{aligned}$$

Combining the two balances:

$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$

The initial state properties of R-134a in the tank are

$$\left.\begin{array}{l} T_1 = 24°\text{C} \\ x = 0 \end{array}\right\} \begin{array}{l} v_1 = 0.0008261\,\text{m}^3/\text{kg} \\ u_1 = 84.44\,\text{kJ/kg} \\ h_e = 84.98\,\text{kJ/kg} \end{array} \qquad \text{(Table A-11)}$$

Note that we assumed that the refrigerant leaving the tank is at saturated liquid state, and found the exiting enthalpy accordingly. The volume of the tank is

$$V = m_1 v_1 = (5\,\text{kg})(0.0008261\,\text{m}^3/\text{kg}) = 0.004131\,\text{m}^3$$

The final specific volume in the container is

$$v_2 = \frac{V}{m_2} = \frac{0.004131\,\text{m}^3}{0.25\,\text{kg}} = 0.01652\,\text{m}^3/\text{kg}$$

The final state is now fixed. The properties at this state are (Table A-11)

$$\left.\begin{array}{l} T_2 = 24°\text{C} \\ v_2 = 0.01652\,\text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{v_2 - v_f}{v_{fg}} = \dfrac{0.01652 - 0.0008261}{0.031834 - 0.0008261} = \mathbf{0.5061} \\ u_2 = u_f + x_2 u_{fg} = 84.44\,\text{kJ/kg} + (0.5061)(158.65\,\text{kJ/kg}) = 164.73\,\text{kJ/kg} \end{array}$$

Substituting into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e \\ &= (0.25\,\text{kg})(164.73\,\text{kJ/kg}) - (5\,\text{kg})(84.44\,\text{kJ/kg}) + (4.75\,\text{kg})(84.98\,\text{kJ/kg}) \\ &= \mathbf{22.64\,kJ} \end{aligned}$$

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5-126E Oxygen is supplied to a medical facility from 10 compressed oxygen tanks in an isothermal process. The mass of oxygen used and the total heat transfer to the tanks are to be determined.

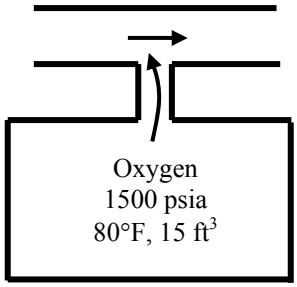
Assumptions **1** This is an unsteady process but it can be analyzed as a uniform-flow process. **2** Oxygen is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved.

Properties The gas constant of oxygen is 0.3353 psia·ft^3/lbm·R (Table A-1E). The specific heats of oxygen at room temperature are $c_p = 0.219$ Btu/lbm·R and $c_v = 0.157$ Btu/lbm·R (Table A-2E*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$
$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + m_e h_e$$
$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$

Combining the two balances:

$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses, and the mass used are

$$m_1 = \frac{P_1 V}{R T_1} = \frac{(1500 \text{ psia})(15 \text{ ft}^3)}{(0.3353 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(80 + 460 \text{ R})} = 124.3 \text{ lbm}$$

$$m_2 = \frac{P_2 V}{R T_2} = \frac{(300 \text{ psia})(15 \text{ ft}^3)}{(0.3353 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(80 + 460 \text{ R})} = 24.85 \text{ lbm}$$

$$m_e = m_1 - m_2 = 124.3 - 24.85 = \mathbf{99.41 lbm}$$

Substituting into the energy balance equation,

$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$
$$= (24.85)(0.157)(540) - (124.3)(0.157)(540) + (99.41)(0.219)(540)$$
$$= \mathbf{3328 Btu}$$

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5-127E Steam is supplied from a line to a weighted piston-cylinder device. The final temperature (and quality if appropriate) of the steam in the piston cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$m_i = m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$
$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$
$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

The boundary work is determined from

$$W_{b,\text{out}} = P(V_2 - V_1) = P(m_2 v_2 - m_1 v_1) = P m_2 v_2$$

Substituting, the energy balance equation simplifies into

$$P m_2 v_2 = m_2 (h_i - u_2)$$
$$P v_2 = h_i - u_2$$

The enthalpy of steam at the inlet is

$$\left.\begin{array}{l} P_i = 300 \text{ psia} \\ T_i = 450°\text{F} \end{array}\right\} h_i = 1226.4 \text{ Btu/lbm} \quad \text{(Table A - 6E)}$$

Substituting this value into the energy balance equation and using an iterative solution of this equation gives (or better yet using EES software)

$$T_2 = \mathbf{425.1°F}$$
$$u_2 = 1135.5 \text{ Btu/lbm}$$
$$v_2 = 2.4575 \text{ ft}^3/\text{lbm}$$

The final mass is

$$m_2 = \frac{V_2}{v_2} = \frac{10 \text{ ft}^3}{2.4575 \text{ ft}^3/\text{lbm}} = 4.069 \text{ lbm}$$

and the work produced is

$$W_{b,\text{out}} = P V_2 = (200 \text{ psia})(10 \text{ ft}^3)\left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3}\right) = \mathbf{370.1 Btu}$$

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5-128E Oxygen is supplied from a line to a weighted piston-cylinder device. The final temperature of the oxygen in the piston cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic. **4** Oxygen is an ideal gas with constant specific heats.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$m_i = m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$
$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$
$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

The boundary work is determined from

$$W_{b,\text{out}} = P(V_2 - V_1) = P(m_2 v_2 - m_1 v_1) = P m_2 v_2$$

Substituting, the energy balance equation simplifies into

$$P m_2 v_2 = m_2 (h_i - u_2)$$
$$P v_2 = h_i - u_2$$
$$R T_2 = c_p T_i - c_v T_2$$

Solving for the final temperature,

$$R T_2 = c_p T_i - c_v T_2 \longrightarrow T_2 = \frac{c_p}{R + c_v} T_i = \frac{c_p}{c_p} T_i = T_i = \mathbf{450°F}$$

The work produced is

$$W_{b,\text{out}} = P V_2 = (200\ \text{psia})(10\ \text{ft}^3)\left(\frac{1\ \text{Btu}}{5.404\ \text{psia} \cdot \text{ft}^3}\right) = \mathbf{370.1\ Btu}$$

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5-129 A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered and the heat transfer are to be determined.

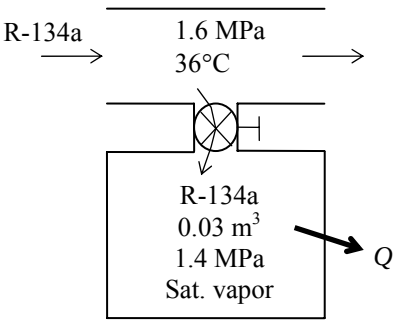
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1.4 \text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@1.4\text{ MPa}} = 0.01411 \text{ m}^3/\text{kg} \\ u_1 = u_{g@1.4\text{ MPa}} = 256.37 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.6 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_2 = v_{f@1.6\text{ MPa}} = 0.0009400 \text{ m}^3/\text{kg} \\ u_2 = u_{f@1.6\text{ MPa}} = 134.43 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.6 \text{ MPa} \\ T_i = 36°\text{C} \end{array}\right\} h_i = h_{f@36°\text{C}} = 102.33 \text{ kJ/kg}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \text{ (since } W \cong ke \cong pe \cong 0)$$

(*a*) The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.03 \text{ m}^3}{0.01411 \text{ m}^3/\text{kg}} = 2.127 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.03 \text{ m}^3}{0.0009400 \text{ m}^3/\text{kg}} = 31.92 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 31.92 - 2.127 = \mathbf{29.79 \text{ kg}}$$

(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(29.79 \text{ kg})(102.33 \text{ kJ/kg}) + (31.92 \text{ kg})(134.43 \text{ kJ/kg}) - (2.127 \text{ kg})(256.37 \text{ kJ/kg}) \\ &= \mathbf{697 \text{ kJ}} \end{aligned}$$

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5-130 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of the mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@200°\text{C}} = 0.001157\ \text{m}^3/\text{kg} \\ u_1 = u_{f@200°\text{C}} = 850.46\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_e = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@200°\text{C}} = 852.26\ \text{kJ/kg}$$

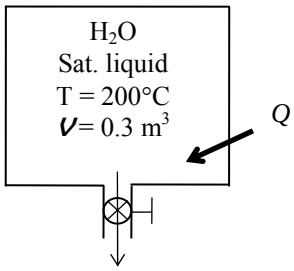
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \ (\text{since } W \cong ke \cong pe \cong 0)$$



The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.3\ \text{m}^3}{0.001157\text{m}^3/\text{kg}} = 259.4\ \text{kg}$$

$$m_2 = \tfrac{1}{2} m_1 = \tfrac{1}{2}(259.4\ \text{kg}) = 129.7\ \text{kg}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 259.4 - 129.7 = 129.7\ \text{kg}$$

Now we determine the final internal energy,

$$v_2 = \frac{V}{m_2} = \frac{0.3\ \text{m}^3}{129.7\ \text{kg}} = 0.002313\ \text{m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002313 - 0.001157}{0.12721 - 0.001157} = 0.009171$$

$$\left.\begin{array}{l} T_2 = 200°\text{C} \\ x_2 = 0.009171 \end{array}\right\} u_2 = u_f + x_2 u_{fg} = 850.46 + (0.009171)(1743.7) = 866.46\ \text{kJ/kg}$$

Then the heat transfer during this process is determined from the energy balance by substitution to be

$$Q = (129.7\ \text{kg})(852.26\ \text{kJ/kg}) + (129.7\ \text{kg})(866.46\ \text{kJ/kg}) - (259.4\ \text{kg})(850.46\ \text{kJ/kg})$$
$$= \mathbf{2308\ kJ}$$

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5-131 A rigid tank initially contains saturated liquid-vapor mixture of refrigerant-134a. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank at constant pressure until no liquid remains inside. The amount of heat transfer is to be determined.

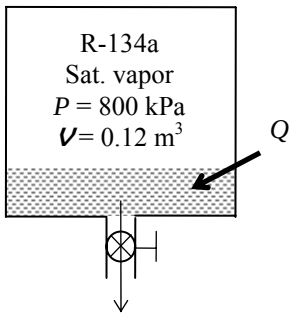
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of R-134a are (Tables A-11 through A-13)

$$P_1 = 800 \text{ kPa} \rightarrow \nu_f = 0.0008458 \text{ m}^3/\text{kg}, \nu_g = 0.025621 \text{ m}^3/\text{kg}$$
$$u_f = 94.79 \text{ kJ/kg}, \quad u_g = 246.79 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \nu_2 = \nu_{g@800\text{ kPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800\text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 800 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@800\text{ kPa}} = 95.47 \text{ kJ/kg}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{\nu_f} + \frac{V_g}{\nu_g} = \frac{0.12 \times 0.25 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} + \frac{0.12 \times 0.75 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 35.47 + 3.513 = 38.98 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (35.47)(94.79) + (3.513)(246.79) = 4229.2 \text{ kJ}$$

$$m_2 = \frac{V}{\nu_2} = \frac{0.12 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 4.684 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 38.98 - 4.684 = 34.30 \text{ kg}$$

$$Q_{\text{in}} = (34.30 \text{ kg})(95.47 \text{ kJ/kg}) + (4.684 \text{ kg})(246.79 \text{ kJ/kg}) - 4229 \text{ kJ} = \mathbf{201.2 \text{ kJ}}$$

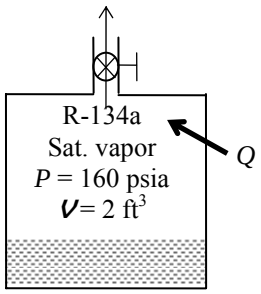
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5-132E A rigid tank initially contains saturated liquid-vapor mixture of R-134a. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until all the liquid in the tank disappears. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Properties The properties of R-134a are (Tables A-11E through A-13E)

$$P_1 = 160 \text{ psia} \rightarrow \nu_f = 0.01413 \text{ ft}^3/\text{lbm},\ \nu_g = 0.29316 \text{ ft}^3/\text{lbm}$$
$$u_f = 48.10 \text{ Btu/lbm},\ u_g = 108.50 \text{ Btu/lbm}$$
$$\left.\begin{matrix} P_2 = 160 \text{ psia} \\ \text{sat. vapor} \end{matrix}\right\} \begin{matrix} \nu_2 = \nu_{g@160\text{ psia}} = 0.29316 \text{ ft}^3/\text{lbm} \\ u_2 = u_{g@160\text{ psia}} = 108.50 \text{ Btu/lbm} \end{matrix}$$
$$\left.\begin{matrix} P_e = 160 \text{ psia} \\ \text{sat. vapor} \end{matrix}\right\} h_e = h_{g@160\text{ psia}} = 117.18 \text{ Btu/lbm}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\begin{matrix}\text{Net energy transfer}\\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta E_{\text{system}}}_{\begin{matrix}\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}\end{matrix}}$$
$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{\nu_f} + \frac{V_g}{\nu_g} = \frac{2 \times 0.2 \text{ ft}^3}{0.01413 \text{ ft}^3/\text{lbm}} + \frac{2 \times 0.8 \text{ ft}^3}{0.29316 \text{ ft}^3/\text{lbm}} = 7.077 + 6.48 = 13.56 \text{ lbm}$$
$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (7.077)(48.10) + (6.48)(108.50) = 1043 \text{ Btu}$$
$$m_2 = \frac{V}{\nu_2} = \frac{2 \text{ ft}^3}{0.29316 \text{ ft}^3/\text{lbm}} = 6.822 \text{ lbm}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 13.56 - 6.822 = 6.736 \text{ lbm}$$

$$\begin{aligned} Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (6.736 \text{ lbm})(117.18 \text{ Btu/lbm}) + (6.822 \text{ lbm})(108.50 \text{ Btu/lbm}) - 1043 \text{ Btu} \\ &= \mathbf{486 \text{ Btu}} \end{aligned}$$

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5-133 A rigid tank initially contains saturated R-134a liquid-vapor mixture. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The final temperature in the tank, the mass of R-134a that entered, and the heat transfer are to be determined.

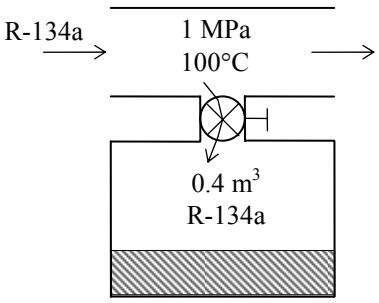
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} T_1 = 14°\text{C} \\ x_1 = 0.7 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.0008020 + 0.7 \times (0.04342 - 0.0008020) = 0.03063\ \text{m}^3/\text{kg} \\ u_1 = u_f + x_1 u_{fg} = 70.57 + 0.7 \times 167.26 = 187.65\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 700\ \text{kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@700\ \text{kPa}} = 0.02936\ \text{m}^3/\text{kg} \\ u_2 = u_{g@700\ \text{kPa}} = 244.48\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.0\ \text{MPa} \\ T_i = 100°C \end{array}\right\} h_i = 335.06\ \text{kJ/kg}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \ (\text{since } W \cong ke \cong pe \cong 0)$$

(*a*) The tank contains saturated vapor at the final state at 800 kPa, and thus the final temperature is the saturation temperature at this pressure,

$$T_2 = T_{\text{sat @ 700 kPa}} = \mathbf{26.7°C}$$

(*b*) The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{0.4\ \text{m}^3}{0.03063\ \text{m}^3/\text{kg}} = 13.06\ \text{kg}$$

$$m_2 = \frac{V}{v_2} = \frac{0.4\ \text{m}^3}{0.02936\ \text{m}^3/\text{kg}} = 13.62\ \text{kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 13.62 - 13.06 = \mathbf{0.5653\ kg}$$

(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(0.5653\ \text{kg})(335.06\ \text{kJ/kg}) + (13.62\ \text{kg})(244.48\ \text{kJ/kg}) - (13.06\ \text{kg})(187.65\ \text{kJ/kg}) \\ &= \mathbf{691\ kJ} \end{aligned}$$

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5-134 A hot-air balloon is considered. The final volume of the balloon and work produced by the air inside the balloon as it expands the balloon skin are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There is no heat transfer.

Properties The gas constant of air is 0.287 kPa·m^3/kg·K (Table A-1).

Analysis The specific volume of the air at the entrance and exit, and in the balloon is

$$\nu = \frac{RT}{P} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(35+273\ \text{K})}{100\ \text{kPa}} = 0.8840\ \text{m}^3/\text{kg}$$

The mass flow rate at the entrance is then

$$\dot{m}_i = \frac{A_i V_i}{\nu} = \frac{(1\ \text{m}^2)(2\ \text{m/s})}{0.8840\ \text{m}^3/\text{kg}} = 2.262\ \text{kg/s}$$

while that at the outlet is

$$\dot{m}_e = \frac{A_e V_e}{\nu} = \frac{(0.5\ \text{m}^2)(1\ \text{m/s})}{0.8840\ \text{m}^3/\text{kg}} = 0.5656\ \text{kg/s}$$

Applying a mass balance to the balloon,

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$m_i - m_e = m_2 - m_1$$
$$m_2 - m_1 = (\dot{m}_i - \dot{m}_e)\Delta t = [(2.262 - 0.5656)\ \text{kg/s}](2\times 60\ \text{s}) = 203.6\ \text{kg}$$

The volume in the balloon then changes by the amount

$$\Delta V = (m_2 - m_1)\nu = (203.6\ \text{kg})(0.8840\ \text{m}^3/\text{kg}) = 180\ \text{m}^3$$

and the final volume of the balloon is

$$V_2 = V_1 + \Delta V = 75 + 180 = \mathbf{255\ m^3}$$

In order to push back the boundary of the balloon against the surrounding atmosphere, the amount of work that must be done is

$$W_{b,\text{out}} = P\Delta V = (100\ \text{kPa})(180\ \text{m}^3)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = \mathbf{18{,}000\ kJ}$$

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5-135 An insulated rigid tank initially contains helium gas at high pressure. A valve is opened, and half of the mass of helium is allowed to escape. The final temperature and pressure in the tank are to be determined.

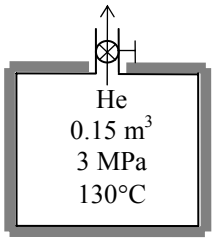
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The tank is insulated and thus heat transfer is negligible. **5** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is k =1.667 (Table A-2).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

$m_2 = \frac{1}{2} m_1$ (given) $\longrightarrow m_e = m_2 = \frac{1}{2} m_1$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-m_e h_e = m_2 u_2 - m_1 u_1 \text{ (since } W \cong Q \cong ke \cong pe \cong 0)$$

Note that the state and thus the enthalpy of helium leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values.

Combining the mass and energy balances: $0 = \frac{1}{2} m_1 h_e + \frac{1}{2} m_1 u_2 - m_1 u_1$

Dividing by $m_1/2$ $\qquad 0 = h_e + u_2 - 2u_1 \;\; or \;\; 0 = c_p \dfrac{T_1 + T_2}{2} + c_v T_2 - 2c_v T_1$

Dividing by c_v: $\qquad 0 = k(T_1 + T_2) + 2T_2 - 4T_1 \qquad \text{since } k = c_p / c_v$

Solving for T_2: $\qquad T_2 = \dfrac{(4-k)}{(2+k)} T_1 = \dfrac{(4-1.667)}{(2+1.667)}(403 \text{ K}) = \mathbf{257\ K}$

The final pressure in the tank is

$$\frac{P_1 \cancel{V}}{P_2 \cancel{V}} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow P_2 = \frac{m_2 T_2}{m_1 T_1} P_1 = \frac{1}{2} \frac{257}{403}(3000 \text{ kPa}) = \mathbf{956\ kPa}$$

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5-136E An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to 30 psia. The amount of electrical work transferred is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

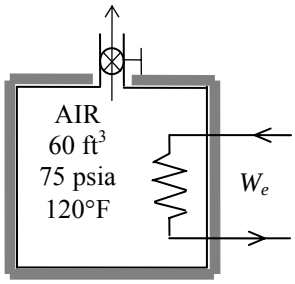
Properties The gas constant of air is R =0.3704 psia.ft^3/lbm.R (Table A-1E). The properties of air are (Table A-17E)

$$T_i = 580 \text{ R} \longrightarrow h_i = 138.66 \text{ Btu/lbm}$$
$$T_1 = 580 \text{ R} \longrightarrow u_1 = 98.90 \text{ Btu/lbm}$$
$$T_2 = 580 \text{ R} \longrightarrow u_2 = 98.90 \text{ Btu/lbm}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \text{ (since } Q \cong ke \cong pe \cong 0)$$

The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(75 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(580 \text{ R})} = 20.95 \text{ lbm}$$
$$m_2 = \frac{P_2 V}{RT_2} = \frac{(30 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(580 \text{ R})} = 8.38 \text{ lbm}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 20.95 - 8.38 = 12.57 \text{ lbm}$$

$$\begin{aligned} W_{\text{e,in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (12.57 \text{ lbm})(138.66 \text{ Btu/lbm}) + (8.38 \text{ lbm})(98.90 \text{ Btu/lbm}) - (20.95 \text{ lbm})(98.90 \text{ Btu/lbm}) \\ &= \mathbf{500 \text{ Btu}} \end{aligned}$$

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5-137 A vertical cylinder initially contains air at room temperature. Now a valve is opened, and air is allowed to escape at constant pressure and temperature until the volume of the cylinder goes down by half. The amount air that left the cylinder and the amount of heat transfer are to be determined.

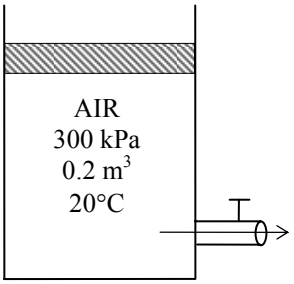
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions other than boundary work. **4** Air is an ideal gas with constant specific heats. **5** The direction of heat transfer is to the cylinder (will be verified).

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1).

Analysis (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{b,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \ \text{(since } ke \cong pe \cong 0)$$

The initial and the final masses of air in the cylinder are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(300 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 0.714 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(300 \text{ kPa})(0.1 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 0.357 \text{ kg} = \tfrac{1}{2} m_1$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 0.714 - 0.357 = \mathbf{0.357 \text{ kg}}$$

(*b*) This is a constant pressure process, and thus the W_b and the ΔU terms can be combined into ΔH to yield

$$Q = m_e h_e + m_2 h_2 - m_1 h_1$$

Noting that the temperature of the air remains constant during this process, we have

$$h_i = h_1 = h_2 = h.$$

Also,

$$m_e = m_2 = \tfrac{1}{2} m_1 .$$

Thus,

$$Q = \left(\tfrac{1}{2} m_1 + \tfrac{1}{2} m_1 - m_1\right) h = \mathbf{0}$$

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5-138 A balloon is initially filled with helium gas at atmospheric conditions. The tank is connected to a supply line, and helium is allowed to enter the balloon until the pressure rises from 100 to 125 kPa. The final temperature in the balloon is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Helium is an ideal gas with constant specific heats. **3** The expansion process is quasi-equilibrium. **4** Kinetic and potential energies are negligible. **5** There are no work interactions involved other than boundary work. **6** Heat transfer is negligible.

Properties The gas constant of helium is $R = 2.0769$ kJ/kg·K (Table A-1). The specific heats of helium are $c_p = 5.1926$ and $c_v = 3.1156$ kJ/kg·K (Table A-2a).

Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

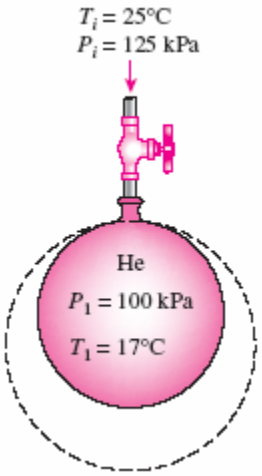
Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$



$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(40\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\text{ K})} = 6.641\text{ kg}$$

$$\frac{P_1}{P_2} = \frac{V_1}{V_2} \longrightarrow V_2 = \frac{P_2}{P_1} V_1 = \frac{125\text{ kPa}}{100\text{ kPa}}(40\text{ m}^3) = 50\text{ m}^3$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(125\text{ kPa})(50\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(T_2\text{ K})} = \frac{3009.3}{T_2}\text{ kg}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = \frac{3009.3}{T_2} - 6.641\text{ kg}$$

Noting that P varies linearly with V, the boundary work done during this process is

$$W_b = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100 + 125)\text{kPa}}{2}(50 - 40)\text{m}^3 = 1125\text{ kJ}$$

Using specific heats, the energy balance relation reduces to

$$W_{\text{b,out}} = m_i c_p T_i - m_2 c_v T_2 + m_1 c_v T_1$$

Substituting,

$$1125 = \left(\frac{3009.3}{T_2} - 6.641\right)(5.1926)(298) - \frac{3009.3}{T_2}(3.1156)T_2 + (6.641)(3.1156)(290)$$

It yields

$$T_2 = \mathbf{315\text{ K}}$$

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5-139 The air in an insulated, rigid compressed-air tank is released until the pressure in the tank reduces to a specified value. The final temperature of the air in the tank is to be determined.

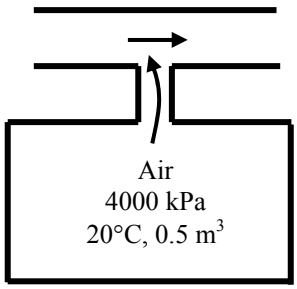
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The gas constant of air is 0.287 kPa·m^3/kg·K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 kJ/kg·K and c_v = 0.718 kJ/kg·K (Table A-2*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$
$$-m_e h_e = m_2 u_2 - m_1 u_1$$
$$0 = m_2 u_2 - m_1 u_1 + m_e h_e$$
$$0 = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$

Combining the two balances:

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{R T_1} = \frac{(4000\ \text{kPa})(0.5\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\ \text{K})} = 23.78\ \text{kg}$$

$$m_2 = \frac{P_2 V}{R T_2} = \frac{(2000\ \text{kPa})(0.5\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}) T_2} = \frac{3484}{T_2}$$

The temperature of air leaving the tank changes from the initial temperature in the tank to the final temperature during the discharging process. We assume that the temperature of the air leaving the tank is the average of initial and final temperatures in the tank. Substituting into the energy balance equation gives

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$
$$0 = \frac{3484}{T_2}(0.718) T_2 - (23.78)(0.718)(293) + \left(23.78 - \frac{3484}{T_2}\right)(1.005)\left(\frac{293 + T_2}{2}\right)$$

whose solution by trial-error or by an equation solver such as EES is

$$T_2 = \mathbf{241\ K} = \mathbf{-32^\circ C}$$

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5-140 An insulated piston-cylinder device with a linear spring is applying force to the piston. A valve at the bottom of the cylinder is opened, and refrigerant is allowed to escape. The amount of refrigerant that escapes and the final temperature of the refrigerant are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process assuming that the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible.

Properties The initial properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{matrix} P_1 = 1.2\ \text{MPa} \\ T_1 = 120°\text{C} \end{matrix}\right\} \begin{matrix} v_1 = 0.02423\ \text{m}^3/\text{kg} \\ u_1 = 325.03\ \text{kJ/kg} \\ h_1 = 354.11\ \text{kJ/kg} \end{matrix}$$

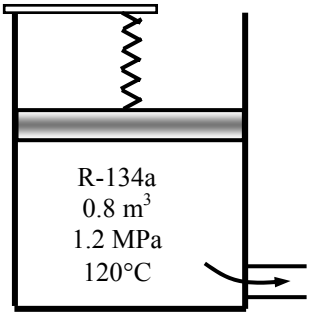
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\begin{matrix}\text{Net energy transfer} \\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta E_{\text{system}}}_{\begin{matrix}\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}\end{matrix}}$$

$$W_{\text{b,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \ \ (\text{since } Q \cong ke \cong pe \cong 0)$$



The initial mass and the relations for the final and exiting masses are

$$m_1 = \frac{V_1}{v_1} = \frac{0.8\ \text{m}^3}{0.02423\ \text{m}^3/\text{kg}} = 33.02\ \text{kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.5\ \text{m}^3}{v_2}$$

$$m_e = m_1 - m_2 = 33.02 - \frac{0.5\ \text{m}^3}{v_2}$$

Noting that the spring is linear, the boundary work can be determined from

$$W_{\text{b,in}} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(1200 + 600)\ \text{kPa}}{2}(0.8 - 0.5)\text{m}^3 = 270\ \text{kJ}$$

Substituting the energy balance,

$$270 - \left(33.02 - \frac{0.5\ \text{m}^3}{v_2}\right)h_e = \left(\frac{0.5\ \text{m}^3}{v_2}\right)u_2 - (33.02\ \text{kg})(325.03\ \text{kJ/kg}) \qquad \text{(Eq. 1)}$$

where the enthalpy of exiting fluid is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder. That is,

$$h_e = \frac{h_1 + h_2}{2} = \frac{(354.11\ \text{kJ/kg}) + h_2}{2}$$

Final state properties of the refrigerant (h_2, u_2, and v_2) are all functions of final pressure (known) and temperature (unknown). The solution may be obtained by a trial-error approach by trying different final state temperatures until Eq. (1) is satisfied. Or solving the above equations simultaneously using an equation solver with built-in thermodynamic functions such as EES, we obtain

$T_2 =$ **96.8°C**, $m_e =$ **22.47 kg,** $h_2 = 336.20$ kJ/kg,

$u_2 = 307.77$ kJ/kg, $v_2 = 0.04739\ \text{m}^3/\text{kg}$, $m_2 = 10.55$ kg

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5-141 Steam at a specified state is allowed to enter a piston-cylinder device in which steam undergoes a constant pressure expansion process. The amount of mass that enters and the amount of heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid entering the device remains constant. **2** Kinetic and potential energies are negligible.

Properties The properties of steam at various states are (Tables A-4 through A-6)

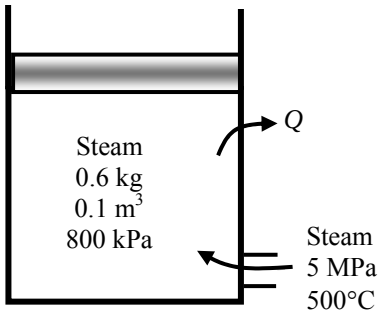
$$v_1 = \frac{V_1}{m_1} = \frac{0.1\ \text{m}^3}{0.6\ \text{kg}} = 0.16667\ \text{m}^3/\text{kg}$$

$$P_2 = P_1$$

$$\left.\begin{matrix} P_1 = 800\ \text{kPa} \\ v_1 = 0.16667\ \text{m}^3/\text{kg} \end{matrix}\right\} u_1 = 2004.4\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_2 = 800\ \text{kPa} \\ T_2 = 250°\text{C} \end{matrix}\right\} \begin{matrix} v_2 = 0.29321\ \text{m}^3/\text{kg} \\ u_2 = 2715.9\ \text{kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_i = 5\ \text{MPa} \\ T_i = 500°\text{C} \end{matrix}\right\} h_i = 3434.7\ \text{kJ/kg}$$



Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\begin{matrix}\text{Net energy transfer} \\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta E_{\text{system}}}_{\begin{matrix}\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}\end{matrix}}$$

$$Q_{\text{in}} - W_{\text{b,out}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

Noting that the pressure remains constant, the boundary work is determined from

$$W_{\text{b,out}} = P(V_2 - V_1) = (800\ \text{kPa})(2 \times 0.1 - 0.1)\text{m}^3 = 80\ \text{kJ}$$

The final mass and the mass that has entered are

$$m_2 = \frac{V_2}{v_2} = \frac{0.2\ \text{m}^3}{0.29321\ \text{m}^3/\text{kg}} = 0.682\ \text{kg}$$

$$m_i = m_2 - m_1 = 0.682 - 0.6 = \mathbf{0.082\ kg}$$

(*b*) Finally, substituting into energy balance equation

$$Q_{\text{in}} - 80\ \text{kJ} + (0.082\ \text{kg})(3434.7\ \text{kJ/kg}) = (0.682\ \text{kg})(2715.9\ \text{kJ/kg}) - (0.6\ \text{kg})(2004.4\ \text{kJ/kg})$$

$$Q_{\text{in}} = \mathbf{447.9\ kJ}$$

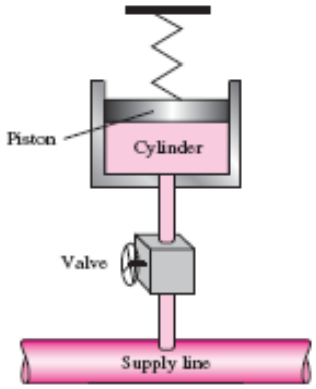
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5-142 Steam is supplied from a line to a piston-cylinder device equipped with a spring. The final temperature (and quality if appropriate) of the steam in the cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \longrightarrow m_i = m_2$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$

$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

Because of the spring, the relation between the pressure and volume is a linear relation. According to the data in the problem statement,

$$P - 300 = \frac{2700}{5} \boldsymbol{V}$$

The final vapor volume is then

$$\boldsymbol{V}_2 = \frac{5}{2700}(1500 - 300) = 2.222 \text{ m}^3$$

The work needed to compress the spring is

$$W_{b,\text{out}} = \int P d\boldsymbol{V} = \int_0^{\boldsymbol{V}_2} \left(\frac{2700}{5} \boldsymbol{V} + 300 \right) d\boldsymbol{V} = \frac{2700}{2 \times 5} \boldsymbol{V}_2^2 + 300 \boldsymbol{V}_2 = 270 \times 2.222^2 + 300 \times 2.222 = \mathbf{2000\ kJ}$$

The enthalpy of steam at the inlet is

$$\left.\begin{array}{l} P_i = 1500 \text{ kPa} \\ T_i = 200°\text{C} \end{array}\right\} h_i = 2796.0 \text{ kJ/kg} \quad \text{(Table A - 6)}$$

Substituting the information found into the energy balance equation gives

$$W_{b,\text{out}} = m_2 (h_i - u_2) \longrightarrow W_{b,\text{out}} = \frac{\boldsymbol{V}_2}{\boldsymbol{v}_2} (h_i - u_2) \longrightarrow 2000 = \frac{2.222}{\boldsymbol{v}_2} (2796.0 - u_2)$$

Using an iterative solution of this equation with steam tables gives (or better yet using EES software)

$$T_2 = \mathbf{233.2°C}$$
$$u_2 = 2664.8 \text{ kJ/kg}$$
$$\boldsymbol{v}_2 = 0.1458 \text{ m}^3/\text{kg}$$

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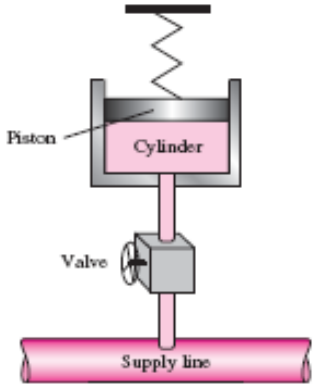
5-143 Air is supplied from a line to a piston-cylinder device equipped with a spring. The final temperature of the steam in the cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic. **4** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is 0.287 kPa·m^3/kg·K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 kJ/kg·K and c_v = 0.718 kJ/kg·K (Table A-2*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \longrightarrow m_i = m_2$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$

$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

Because of the spring, the relation between the pressure and volume is a linear relation. According to the data in the problem statement,

$$P - 300 = \frac{2700}{5} V$$

The final air volume is then

$$V_2 = \frac{5}{2700}(2000 - 300) = 3.148 \text{ m}^3$$

The work needed to compress the spring is

$$W_{b,\text{out}} = \int P dV = \int_0^{V_2} \left(\frac{2700}{5} V + 300 \right) dV = \frac{2700}{2 \times 5} V_2^2 + 300 V_2 = 270 \times 3.148^2 + 300 \times 3.148 = \mathbf{3620\ kJ}$$

Substituting the information found into the energy balance equation gives

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

$$W_{b,\text{out}} = \frac{P_2 V_2}{R T_2}(c_p T_i - c_v T_2)$$

$$3620 = \frac{2000 \times 3.148}{(0.287) T_2}(1.005 \times 600 - 0.718 \times T_2)$$

The final temperature is then

$$T_2 = \mathbf{682.9\ K} = \mathbf{409.9°C}$$

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Review Problems

5-144 Carbon dioxide flows through a throttling valve. The temperature change of CO_2 is to be determined if CO_2 is assumed an ideal gas and a real gas.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

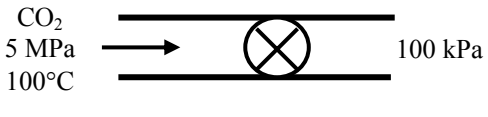
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$.

(*a*) For an ideal gas, $h = h(\text{T})$, and therefore,

$$T_2 = T_1 = 100°C \longrightarrow \Delta T = T_1 - T_2 = \mathbf{0°C}$$

(*b*) We obtain real gas properties of CO_2 from EES software as follows

$$\left.\begin{matrix} P_1 = 5 \text{ MPa} \\ T_1 = 100°\text{C} \end{matrix}\right\} h_1 = 34.77 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 100 \text{ kPa} \\ h_2 = h_1 = 34.77 \text{ kJ/kg} \end{matrix}\right\} T_2 = 66.0°\text{C}$$

Note that EES uses a different reference state from the textbook for CO_2 properties. The temperature difference in this case becomes

$$\Delta T = T_1 - T_2 = 100 - 66.0 = \mathbf{34.0°C}$$

That is, the temperature of CO_2 decreases by 34°C in a throttling process if its real gas properties are used.

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5-145 Helium flows steadily in a pipe and heat is lost from the helium during this process. The heat transfer and the volume flow rate at the exit are to be determined.

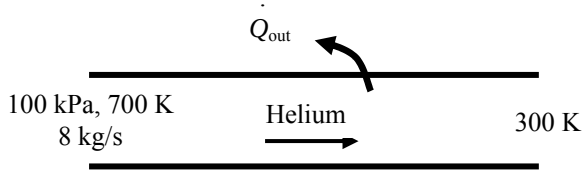
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of helium is 2.0769 kPa.m^3/kg.K. The constant pressure specific heat of air at room temperature is c_p = 5.1926 kJ/kg·°C (Table A-2*a*).

Analysis (a) We take the pipe in which the argon is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 = \dot{m}h_2 + \dot{Q}_{\text{out}}$$
$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2)$$
$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Substituting,

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2) = (8\ \text{kg/s})(5.1926\ \text{kJ/kg}\cdot\text{K})(427 - 27)\text{K} = \mathbf{16{,}620\ kW}$$

(b) The exit specific volume and the volume flow rate are

$$v_2 = \frac{RT_2}{P_2} = \frac{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\ \text{K})}{100\ \text{kPa}} = 6.231\ \text{m}^3/\text{kg}$$
$$\dot{V}_2 = \dot{m}v_2 = (8\ \text{kg/s})(6.231\ \text{m}^3/\text{kg}) = \mathbf{49.85\ m^3/s}$$

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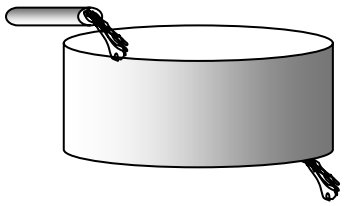
5-146 The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.

Assumptions **1** Water is supplied and discharged steadily. **2** The rate of evaporation of water is negligible. **3** No water is supplied or removed through other means.

Analysis The conservation of mass principle applied to the pool requires that the rate of increase in the amount of water in the pool be equal to the difference between the rate of supply of water and the rate of discharge. That is,

$$\frac{dm_{\text{pool}}}{dt} = \dot{m}_i - \dot{m}_e \quad \rightarrow \quad \dot{m}_i = \frac{dm_{\text{pool}}}{dt} + \dot{m}_e \quad \rightarrow \quad \dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e$$

since the density of water is constant and thus the conservation of mass is equivalent to conservation of volume. The rate of discharge of water is



$$\dot{V}_e = A_e V_e = (\pi D^2/4)V_e = [\pi(0.07\text{ m})^2/4](4\text{ m/s}) = 0.01539\text{ m}^3/\text{s}$$

The rate of accumulation of water in the pool is equal to the cross-section of the pool times the rate at which the water level rises,

$$\frac{dV_{\text{pool}}}{dt} = A_{\text{cross-section}} V_{\text{level}} = (6\text{ m} \times 9\text{ m})(0.025\text{ m/min}) = 1.35\text{ m}^3/\text{min} = 0.0225\text{ m}^3/\text{s}$$

Substituting, the rate at which water is supplied to the pool is determined to be

$$\dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e = 0.0225 + 0.01539 = \mathbf{0.0379\ m^3/s}$$

Therefore, water is supplied at a rate of 0.0379 m^3/s = 37.9 L/s.

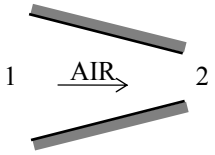
5-147 Air is accelerated in a nozzle. The density of air at the nozzle exit is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 4.18 kg/m^3 at the inlet.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\dot{m}_1 = \dot{m}_2$$
$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$
$$\rho_2 = \frac{A_1}{A_2}\frac{V_1}{V_2}\rho_1 = 2\frac{120\text{ m/s}}{380\text{ m/s}}(4.18\text{ kg/m}^3) = \mathbf{2.64\ kg/m^3}$$



Discussion Note that the density of air decreases considerably despite a decrease in the cross-sectional area of the nozzle.

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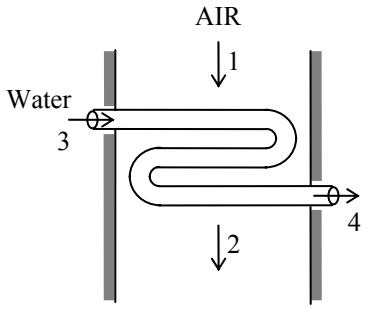
5-148E A heat exchanger that uses hot air to heat cold water is considered. The total flow power and the flow works for both the air and water streams are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm.R =0.06855 Btu/lbm.R (Table A-1E). The specific volumes of water at the inlet and exit are (Table A-4E)

$$\left.\begin{matrix} P_3 = 20 \text{ psia} \\ T_3 = 50°\text{F} \end{matrix}\right\} \nu_3 \cong \nu_{f@50°\text{F}} = 0.01602 \text{ ft}^3/\text{lbm}$$

$$\left.\begin{matrix} P_4 = 17 \text{ psia} \\ T_4 = 90°\text{F} \end{matrix}\right\} \nu_4 \cong \nu_{f@90°\text{F}} = 0.01610 \text{ ft}^3/\text{lbm}$$



Analysis The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(200+460 \text{ R})}{20 \text{ psia}} = 12.22 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{(100/60) \text{ ft}^3/\text{s}}{12.22 \text{ ft}^3/\text{lbm}} = 0.1364 \text{ lbm/s}$$

Combining the flow work expression with the ideal gas equation of state gives

$$w_{\text{flow}} = P_2\nu_2 - P_1\nu_1 = R(T_2 - T_1) = (0.06855 \text{ Btu/lbm}\cdot\text{R})(100-200)\text{R} = \mathbf{-6.855 \text{ Btu/lbm}}$$

The flow work of water is

$$\begin{aligned} w_{\text{flow}} &= P_4\nu_4 - P_3\nu_3 \\ &= \left[(17 \text{ psia})(0.01610 \text{ ft}^3/\text{lbm}) - (20 \text{ psia})(0.01602 \text{ ft}^3/\text{lbm})\right]\left(\frac{1 \text{ Btu}}{5.404 \text{ psia}\cdot\text{ft}^3}\right) \\ &= \mathbf{-0.00864 \text{ Btu/lbm}} \end{aligned}$$

The net flow power for the heat exchanger is

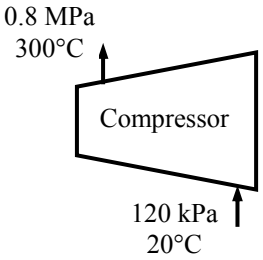
$$\begin{aligned} \dot{W}_{\text{flow}} &= \dot{m}_{\text{air}} w_{\text{flow}} + \dot{m}_{\text{air}} w_{\text{flow}} \\ &= (0.1364 \text{ lbm/s})(\text{-}6.855 \text{ Btu/lbm}) + (0.5 \text{ lbm/s})(\text{-}0.00864 \text{ Btu/lbm}) \\ &= -0.9393 \text{ Btu/s}\left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}}\right) = \mathbf{-1.329 \text{ hp}} \end{aligned}$$

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5-149 An air compressor consumes 6.2 kW of power to compress a specified rate of air. The flow work required by the compressor is to be compared to the power used to increase the pressure of the air.

Assumptions **1** Flow through the compressor is steady. **2** Air is an ideal gas.

Properties The gas constant of air is 0.287 kPa·m^3/kg·K (Table A-1).



Analysis The specific volume of the air at the inlet is

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\ \text{K})}{120\ \text{kPa}} = 0.7008\ \text{m}^3/\text{kg}$$

The mass flow rate of the air is

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.015\ \text{m}^3/\text{s}}{0.7008\ \text{m}^3/\text{kg}} = 0.02140\ \text{kg/s}$$

Combining the flow work expression with the ideal gas equation of state gives the flow work as

$$w_{\text{flow}} = P_2\nu_2 - P_1\nu_1 = R(T_2 - T_1) = (0.287\ \text{kJ/kg}\cdot\text{K})(300-20)\text{K} = 80.36\ \text{kJ/kg}$$

The flow power is

$$\dot{W}_{\text{flow}} = \dot{m}w_{\text{flow}} = (0.02140\ \text{kg/s})(80.36\ \text{kJ/kg}) = \mathbf{1.72\ kW}$$

The remainder of compressor power input is used to increase the pressure of the air:

$$\dot{W} = \dot{W}_{\text{total,in}} - \dot{W}_{\text{flow}} = 6.2 - 1.72 = \mathbf{4.48\ kW}$$

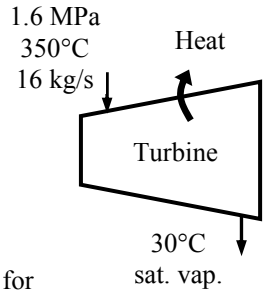
5-150 Steam expands in a turbine whose power production is 9000 kW. The rate of heat lost from the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties From the steam tables (Tables A-6 and A-4)

$$\left.\begin{matrix} P_1 = 1.6\ \text{MPa} \\ T_1 = 350°\text{C} \end{matrix}\right\} h_1 = 3146.0\ \text{kJ/kg}$$

$$\left.\begin{matrix} T_2 = 30°\text{C} \\ x_2 = 1 \end{matrix}\right\} h_2 = 2555.6\ \text{kJ/kg}$$



Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that there is one inlet and one exiti the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{W}_{\text{out}} + \dot{Q}_{\text{out}}$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2) - \dot{W}_{\text{out}}$$

Substituting,

$$\dot{Q}_{\text{out}} = (16\ \text{kg/s})(3146.0 - 2555.6)\ \text{kJ/kg} - 9000\ \text{kW} = \mathbf{446.4\ kW}$$

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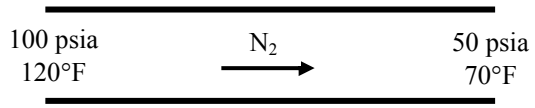
5-151E Nitrogen gas flows through a long, constant-diameter adiabatic pipe. The velocities at the inlet and exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Nitrogen is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The re is no heat transfer from the nitrogen.

Properties The specific heat of nitrogen at the room temperature iss c_p = 0.248 Btu/lbm·R (Table A-2E*a*).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the pipe as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

$$\frac{V_1^2 - V_2^2}{2} = c_p(T_2 - T_1)$$

Combining the mass balance and ideal gas equation of state yields

$$\dot{m}_1 = \dot{m}_2$$

$$\frac{A_1 V_1}{\nu_1} = \frac{A_2 V_2}{\nu_2}$$

$$V_2 = \frac{A_1}{A_2}\frac{\nu_2}{\nu_1}V_1 = \frac{\nu_2}{\nu_1}V_1 = \frac{T_2}{T_1}\frac{P_1}{P_2}V_1$$

Substituting this expression for V_2 into the energy balance equation gives

$$V_1 = \left[\frac{2c_p(T_2 - T_1)}{1 - \left(\frac{T_2}{T_1}\frac{P_1}{P_2}\right)^2}\right]^{0.5} = \left[\frac{2(0.248)(70-120)}{1 - \left(\frac{530}{580}\frac{100}{50}\right)^2}\left(\frac{25{,}037\ \text{ft}^2/\text{s}^2}{1\ \text{Btu/lbm}}\right)\right]^{0.5} = \mathbf{515\ ft/s}$$

The velocity at the exit is

$$V_2 = \frac{T_2}{T_1}\frac{P_1}{P_2}V_1 = \frac{530}{580}\frac{100}{50}515 = \mathbf{941\ ft/s}$$

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5-152 Water at a specified rate is heated by an electrical heater. The current is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The heat losses from the water is negligible.

Properties The specific heat and the density of water are taken to be $c_p = 4.18$ kJ/kg·°C and $\rho = 1$ kg/L (Table A-3).

Analysis We take the pipe in which water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

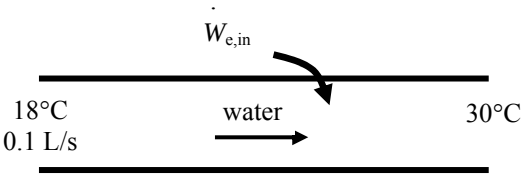
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{e,in}} = \dot{m}h_2$$

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1)$$

$$\mathbf{V}I = \dot{m}c_p(T_2 - T_1)$$



The mass flow rate of the water is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1\,\text{kg/L})(0.1\,\text{L/s}) = 0.1\,\text{kg/s}$$

Substituting into the energy balance equation and solving for the current gives

$$I = \frac{\dot{m}c_p(T_2 - T_1)}{\mathbf{V}} = \frac{(0.1\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot\text{K})(30-18)\text{K}}{110\,\text{V}}\left(\frac{1000\,\text{VI}}{1\,\text{kJ/s}}\right) = \mathbf{45.6\ A}$$

5-153 Water is boiled at $T_{\text{sat}} = 100$°C by an electric heater. The rate of evaporation of water is to be determined.

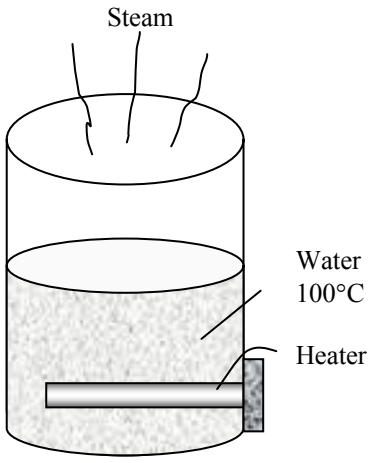
Assumptions **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the water tank are negligible.

Properties The enthalpy of vaporization of water at 100°C is $h_{\text{fg}} = 2256.4$ kJ/kg (Table A-4).

Analysis Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{W}_{\text{e,boiling}}}{h_{fg}} = \frac{3\,\text{kJ/s}}{2256.4\,\text{kJ/kg}}$$

$$= \mathbf{0.00133\ kg/s = 4.79\ kg/h}$$



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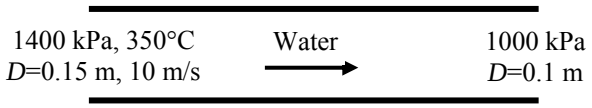
5-154 Steam flows in an insulated pipe. The mass flow rate of the steam and the speed of the steam at the pipe outlet are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions.

Analysis We take the pipe in which steam flows as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 = \dot{m}h_2$$
$$h_1 = h_2$$



The properties of the steam at the inlet and exit are (Table A-6)

$$\left.\begin{array}{l} P = 1400 \text{ kPa} \\ T = 350°\text{C} \end{array}\right\} \begin{array}{l} \nu_1 = 0.20029 \text{ m}^3/\text{kg} \\ h_1 = 3150.1 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1000 \text{ kPa} \\ h_2 = h_1 = 3150.1 \text{ kJ/kg} \end{array}\right\} \nu_2 = 0.28064 \text{ m}^3/\text{kg}$$

The mass flow rate is

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{\pi D_1^2}{4}\frac{V_1}{\nu_1} = \frac{\pi (0.15 \text{ m})^2}{4}\frac{10 \text{ m/s}}{0.20029 \text{ m}^3/\text{kg}} = \mathbf{0.8823 \text{ kg/s}}$$

The outlet velocity will then be

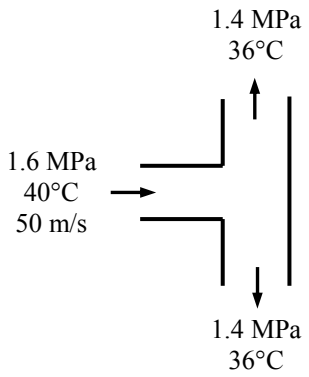
$$V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{4\dot{m}\nu_2}{\pi D_2^2} = \frac{4(0.8823 \text{ kg/s})(0.28064 \text{ m}^3/\text{kg})}{\pi (0.10 \text{ m})^2} = \mathbf{31.53 \text{ m/s}}$$

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5-155 The mass flow rate of a compressed air line is divided into two equal streams by a T-fitting in the line. The velocity of the air at the outlets and the rate of change of flow energy (flow power) across the T-fitting are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** The flow is steady. **3** Since the outlets are identical, it is presumed that the flow divides evenly between the two.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1).



Analysis The specific volumes of air at the inlet and outlets are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(40+273\,\text{K})}{1600\,\text{kPa}} = 0.05614\,\text{m}^3/\text{kg}$$

$$\nu_2 = \nu_3 = \frac{RT_2}{P_2} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(36+273\,\text{K})}{1400\,\text{kPa}} = 0.06335\,\text{m}^3/\text{kg}$$

Assuming an even division of the inlet flow rate, the mass balance can be written as

$$\frac{A_1V_1}{\nu_1} = 2\frac{A_2V_2}{\nu_2} \longrightarrow V_2 = V_3 = \frac{A_1}{A_2}\frac{\nu_2}{\nu_1}\frac{V_1}{2} = \frac{0.06335}{0.05614}\frac{50}{2} = \mathbf{28.21\,m/s}$$

The mass flow rate at the inlet is

$$\dot{m}_1 = \frac{A_1V_1}{\nu_1} = \frac{\pi D_1^2}{4}\frac{V_1}{\nu_1} = \frac{\pi(0.025\,\text{m})^2}{4}\frac{50\,\text{m/s}}{0.05614\,\text{m}^3/\text{kg}} = 0.4372\,\text{kg/s}$$

while that at the outlets is

$$\dot{m}_2 = \dot{m}_3 = \frac{\dot{m}_1}{2} = \frac{0.4372\,\text{kg/s}}{2} = 0.2186\,\text{kg/s}$$

Substituting the above results into the flow power expression produces

$$\begin{aligned}\dot{W}_{\text{flow}} &= 2\dot{m}_2P_2\nu_2 - \dot{m}_1P_1\nu_1 \\ &= 2(0.2186\,\text{kg/s})(1400\,\text{kPa})(0.06335\,\text{m}^3/\text{kg}) - (0.4372\,\text{kg/s})(1600\,\text{kPa})(0.05614\,\text{m}^3/\text{kg}) \\ &= \mathbf{-0.496\,kW}\end{aligned}$$

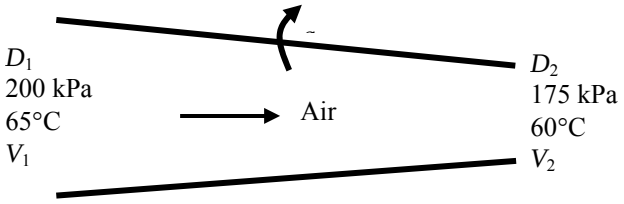
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5-156 Air flows through a non-constant cross-section pipe. The inlet and exit velocities of the air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible. **5** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K. Also, $c_p = 1.005$ kJ/kg.K for air at room temperature (Table A-2)

Analysis We take the pipe as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as



Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \longrightarrow \rho_1 A_1 V_1 = \rho_2 A_2 V_2 \longrightarrow \frac{P_1}{RT_1}\frac{\pi D_1^2}{4}V_1 = \frac{P_2}{RT_2}\frac{\pi D_2^2}{4}V_2 \longrightarrow \frac{P_1}{T_1}D_1^2 V_1 = \frac{P_2}{T_2}D_2^2 V_2$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \text{since } \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

or $$c_p T_1 + \frac{V_1^2}{2} = c_p T_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

Assuming inlet diameter to be 1.4 m and the exit diameter to be 1.0 m, and substituting given values into mass and energy balance equations

$$\left(\frac{200\,\text{kPa}}{338\,\text{K}}\right)(1.4\,\text{m})^2 V_1 = \left(\frac{175\,\text{kPa}}{333\,\text{K}}\right)(1.0\,\text{m})^2 V_2 \qquad (1)$$

$$(1.005\,\text{kJ/kg.K})(338\,\text{K}) + \frac{V_1^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = (1.005\,\text{kJ/kg.K})(333\,\text{K}) + \frac{V_2^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) + 3.3\,\text{kJ/kg} \qquad (2)$$

There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

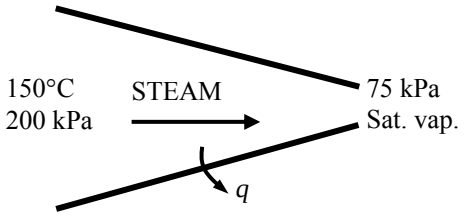
$V_1 =$ **29.9 m/s**

$V_2 =$ **66.1 m/s**

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5-157 Heat is lost from the steam flowing in a nozzle. The exit velocity and the mass flow rate are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions.



Analysis (*a*) We take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{\text{out}} \quad \text{since } \dot{W} \cong \Delta \text{pe} \cong 0)$$

or $$V_2 = \sqrt{2(h_1 - h_2 - q_{\text{out}})}$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} h_1 = 2769.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 75 \text{ kPa} \\ \text{sat. vap.} \end{array}\right\} \begin{array}{l} \nu_2 = 2.2172 \text{ m}^3/\text{kg} \\ h_2 = 2662.4 \text{ kJ/kg} \end{array}$$

Substituting,

$$V_2 = \sqrt{2(h_1 - h_2 - q_{\text{out}})} = \sqrt{2(2769.1 - 2662.4 - 26)\text{kJ/kg}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)} = \mathbf{401.7 \text{ m/s}}$$

(*b*) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 = \frac{1}{2.2172 \text{ m}^3/\text{kg}} (0.001 \text{ m}^2)(401.7 \text{ m/s}) = \mathbf{0.181 kg/s}$$

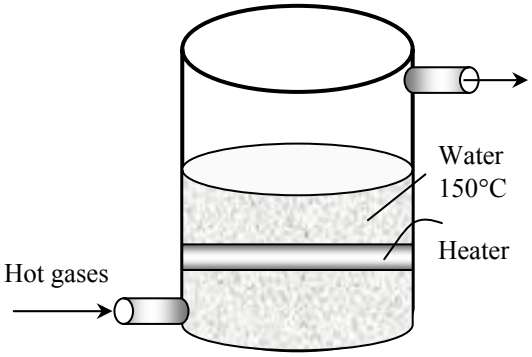
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5-158 Water is boiled at a specified temperature by hot gases flowing through a stainless steel pipe submerged in water. The rate of evaporation of is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the boiler are negligible.

Properties The enthalpy of vaporization of water at 150°C is h_{fg} = 2113.8 kJ/kg (Table A-4).

Analysis The rate of heat transfer to water is given to be 74 kJ/s. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be



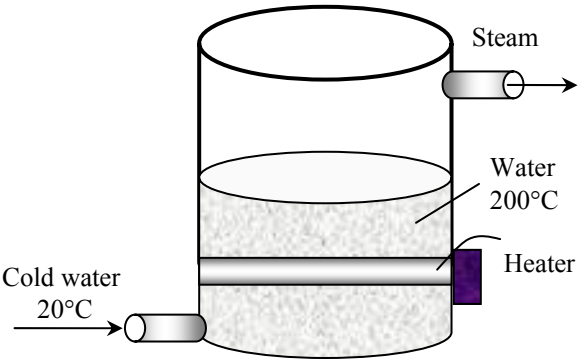
$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{74\ \text{kJ/s}}{2113.8\ \text{kJ/kg}} = \mathbf{0.0350\ kg/s}$$

5-159 Cold water enters a steam generator at 20°C, and leaves as saturated vapor at T_{sat} = 200°C. The fraction of heat used to preheat the liquid water from 20°C to saturation temperature of 200°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the steam generator are negligible. **3** The specific heat of water is constant at the average temperature.

Properties The heat of vaporization of water at 200°C is h_{fg} = 1939.8 kJ/kg (Table A-4), and the specific heat of liquid water is c = 4.18 kJ/kg.°C (Table A-3).

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature. Using the average specific heat, the amount of heat transfer needed to preheat a unit mass of water from 20°C to 200°C is



$$\begin{aligned} q_{\text{preheating}} &= c\Delta T \\ &= (4.18\ \text{kJ/kg}\cdot\text{°C})(200-20)\text{°C} \\ &= 752.4\ \text{kJ/kg} \end{aligned}$$

and

$$\begin{aligned} q_{\text{total}} &= q_{\text{boiling}} + q_{\text{preheating}} \\ &= 1939.8 + 364.1 = 2692.2\ \text{kJ/kg} \end{aligned}$$

Therefore, the fraction of heat used to preheat the water is

$$\text{Fraction to preheat} = \frac{q_{\text{preheating}}}{q_{\text{total}}} = \frac{752.4}{2692.2} = \mathbf{0.2795}\ \text{(or}\ \mathbf{28.0\%}\text{)}$$

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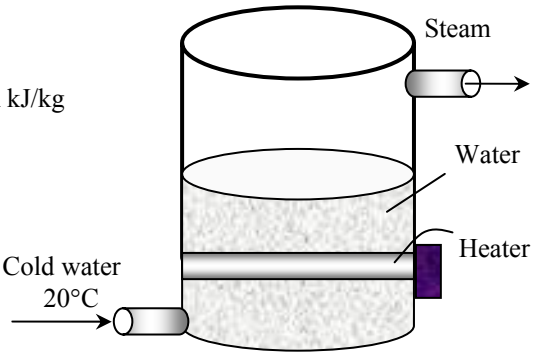
5-160 Cold water enters a steam generator at 20°C and is boiled, and leaves as saturated vapor at boiler pressure. The boiler pressure at which the amount of heat needed to preheat the water to saturation temperature that is equal to the heat of vaporization is to be determined.

Assumptions Heat losses from the steam generator are negligible.

Properties The enthalpy of liquid water at 20°C is 83.91 kJ/kg. Other properties needed to solve this problem are the heat of vaporization h_{fg} and the enthalpy of saturated liquid at the specified temperatures, and they can be obtained from Table A-4.

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature, and Δh represents the amount of heat needed to preheat a unit mass of water from 20°C to the saturation temperature. Therefore,

$$q_{\text{preheating}} = q_{\text{boiling}}$$
$$(h_{f@T_{\text{sat}}} - h_{f@20°C}) = h_{fg@T_{\text{sat}}}$$
$$h_{f@T_{\text{sat}}} - 83.91\,\text{kJ/kg} = h_{fg@T_{\text{sat}}} \;\rightarrow\; h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 83.91\,\text{kJ/kg}$$



The solution of this problem requires choosing a boiling temperature, reading h_f and h_{fg} at that temperature, and substituting the values into the relation above to see if it is satisfied. By trial and error, (Table A-4)

At 310°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1402.0 - 1325.9 = 76.1$ kJ/kg

At 315°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1431.6 - 1283.4 = 148.2$ kJ/kg

The temperature that satisfies this condition is determined from the two values above by interpolation to be 310.6°C. The saturation pressure corresponding to this temperature is **9.94 MPa.**

5-161 An ideal gas expands in a turbine. The volume flow rate at the inlet for a power output of 350 kW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties The properties of the ideal gas are given as $R = 0.30$ kPa.m^3/kg.K, $c_p = 1.13$ kJ/kg·°C, $c_v = 0.83$ kJ/kg·°C.

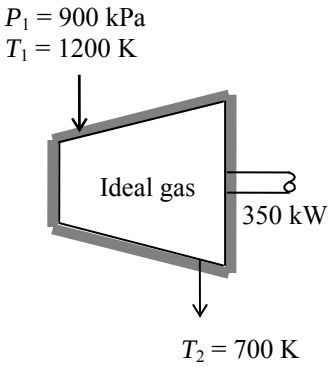
Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow \dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \;\; (\text{since } \dot{Q} \cong \Delta\text{ke} = \Delta\text{pe} \cong 0)$$

which can be rearranged to solve for mass flow rate

$$\dot{m} = \frac{\dot{W}_{\text{out}}}{h_1 - h_2} = \frac{\dot{W}_{\text{out}}}{c_p(T_1 - T_2)} = \frac{350\,\text{kW}}{(1.13\,\text{kJ/kg.K})(1200 - 700)\text{K}} = 0.6195\,\text{kg/s}$$



The inlet specific volume and the volume flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(1200\,\text{K})}{900\,\text{kPa}} = 0.4\,\text{m}^3/\text{kg}$$

Thus, $\dot{V} = \dot{m}v_1 = (0.6195\,\text{kg/s})(0.4\,\text{m}^3/\text{kg}) = \mathbf{0.248\,m^3/s}$

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5-162 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

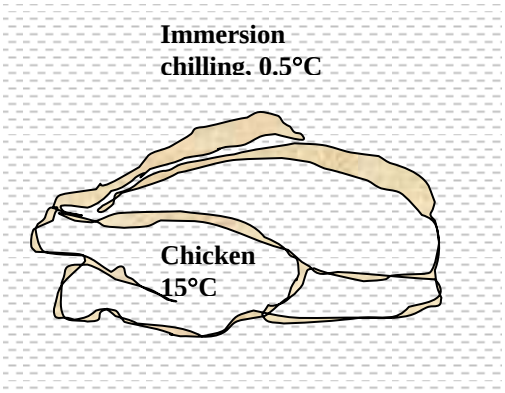
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500 \text{ chicken / h})(2.2 \text{ kg / chicken}) = 1100 \text{ kg / h} = 0.3056 \text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m} c_p \Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.}^\circ\text{C})(15 - 3)^\circ\text{C} = \mathbf{13.0\ kW}$$

The chiller gains heat from the surroundings at a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 0.056 = 13.056 \text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p \Delta T)_{\text{water}}} = \frac{13.056 \text{ kW}}{(4.18 \text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

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5-163 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

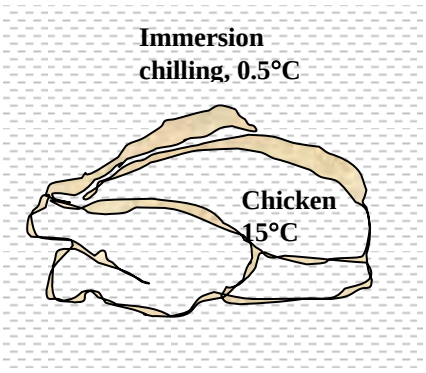
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant. **3** Heat gain of the chiller is negligible.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500 \text{ chicken / h})(2.2 \text{ kg / chicken}) = 1100 \text{ kg / h} = 0.3056 \text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m} c_p \Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.}^{\circ}\text{C})(15 - 3)^{\circ}\text{C} = \mathbf{13.0\ kW}$$

Heat gain of the chiller from the surroundings is negligible. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} = 13.0 \text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p \Delta T)_{\text{water}}} = \frac{13.0 \text{ kW}}{(4.18 \text{ kJ/kg.}^{\circ}\text{C})(2^{\circ}\text{C})} = \mathbf{1.56\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

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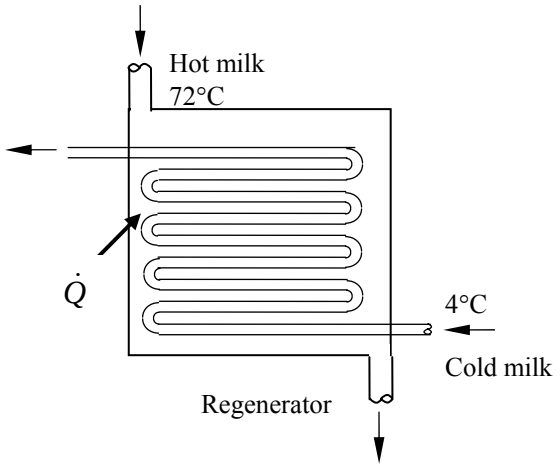
5-164 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year are to be determined.

Assumptions 1 Steady operating conditions exist. **2** The properties of the milk are constant.

Properties The average density and specific heat of milk can be taken to be $\rho_{milk} \cong \rho_{water} = 1$ kg/L and $c_{p,\,milk} = 3.79$ kJ/kg.°C (Table A-3).

Analysis The mass flow rate of the milk is

$$\begin{aligned}\dot{m}_{milk} &= \rho \dot{V}_{milk} \\ &= (1\,\text{kg/L})(20\,\text{L/s}) = 20\,\text{kg/s} \\ &= 72{,}000\,\text{kg/h}\end{aligned}$$



Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\begin{aligned}\dot{E}_{in} &= \dot{E}_{out} \\ \dot{Q}_{in} + \dot{m}h_1 &= \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0) \\ \dot{Q}_{in} &= \dot{m}_{milk} c_p (T_2 - T_1)\end{aligned}$$

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\begin{aligned}\dot{Q}_{current} &= [\dot{m} c_p (T_{pasturization} - T_{refrigeration})]_{milk} \\ &= (20\ \text{kg/s})(3.79\ \text{kJ/kg.ëC})(72 - 4)°\text{C} = 5154\ \text{kJ/s}\end{aligned}$$

The proposed regenerator has an effectiveness of ε = 0.82, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{saved} = \varepsilon \dot{Q}_{current} = (0.82)(5154\,\text{kJ/s}) = 4227\,\text{kJ/s}$$

Noting that the boiler has an efficiency of $\eta_{boiler} = 0.90$, the energy savings above correspond to fuel savings of

$$\text{FuelSaved} = \frac{\dot{Q}_{saved}}{\eta_{boiler}} = \frac{4227\,\text{kJ/s}}{0.90}\frac{1\text{therm}}{105{,}500\,\text{kJ}} = 0.04452\ \text{therm/s}$$

Noting that 1 year = 365 × 24 = 8760 h and unit cost of natural gas is $1.10/therm, the annual fuel and money savings will be

$$\text{Fuel Saved} = (0.04452\ \text{therms/s})(8760 \times 3600\ \text{s}) = \mathbf{1.404 \times 10^6\ therms/yr}$$

$$\begin{aligned}\text{Money saved} &= (\text{Fuel saved})(\text{Unit cost of fuel}) \\ &= (1.404 \times 10^6\ \text{therm/yr})(\$1.10/\text{therm}) \\ &= \mathbf{\$1.544 \times 10^6/yr}\end{aligned}$$

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5-165E A refrigeration system is to cool eggs by chilled air at a rate of 10,000 eggs per hour. The rate of heat removal from the eggs, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

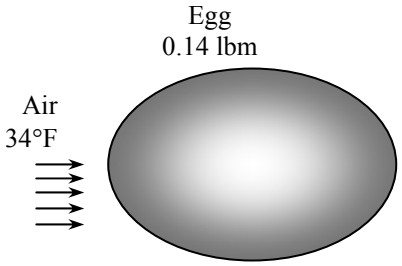
Assumptions **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated. **4** The properties of eggs are constant. **5** The local atmospheric pressure is 1 atm.

Properties The properties of the eggs are given to $\rho = 67.4$ lbm/ft^3 and $c_p = 0.80$ Btu/lbm.°F. The specific heat of air at room temperature $c_p = 0.24$ Btu/lbm. °F (Table A-2E). The gas constant of air is $R = 0.3704$ psia.ft^3/lbm.R (Table A-1E).

Analysis (*a*) Noting that eggs are cooled at a rate of 10,000 eggs per hour, eggs can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{egg}} = (10{,}000 \text{ eggs/h})(0.14 \text{ lbm/egg}) = 1400 \text{ lbm/h} = 0.3889 \text{ lbm/s}$$

Taking the egg flow stream in the cooler as the system, the energy balance for steadily flowing eggs can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{egg}} = \dot{m}_{\text{egg}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the eggs as they are cooled from 90°F to 50ºF at this rate becomes

$$\dot{Q}_{\text{egg}} = (\dot{m} c_p \Delta T)_{\text{egg}} = (1400 \text{ lbm/h})(0.80 \text{ Btu/lbm.°F})(90 - 50)°\text{F} = \mathbf{44{,}800 \text{ Btu/h}}$$

(*b*) All the heat released by the eggs is absorbed by the refrigerated air since heat transfer through he walls of cooler is negligible, and the temperature rise of air is not to exceed 10°F. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p \Delta T)_{\text{air}}} = \frac{44{,}800 \text{ Btu/h}}{(0.24 \text{Btu/lbm.°F})(10°\text{F})} = 18{,}667 \text{ lbm/h}$$

$$\rho_{\text{air}} = \frac{P}{RT} = \frac{14.7 \text{ psia}}{(0.3704 \text{ psia.ft}^3\text{/lbm.R})(34 + 460)\text{R}} = 0.0803 \text{ lbm/ft}^3$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{18{,}667 \text{ lbm/h}}{0.0803 \text{ lbm/ft}^3} = \mathbf{232{,}500 \text{ ft}^3\text{/h}}$$

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5-166 Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined.

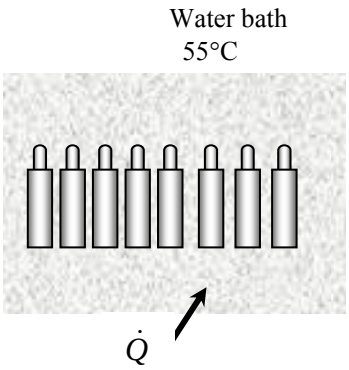
Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 55°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

Properties The specific heat of water at room temperature is $c_p = 4.18$ kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

Analysis (*a*) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{bottle}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$

Then the rate of heat removal by the bottles as they are heated from 20 to 55°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} c_p \Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg.}^\circ\text{C})(55 - 20)^\circ\text{C} = 56{,}000 \text{ W}$$

The amount of water removed by the bottles is

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$

$$= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = \mathbf{2.67 \times 10^{-3} \text{ kg/s}}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg.}^\circ\text{C})(55 - 15)^\circ\text{C} = 446 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 56{,}000 + 446 = \mathbf{56{,}446 \text{ W}}$$

Discussion In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

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5-167 Long aluminum wires are extruded at a velocity of 8 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

Properties The properties of aluminum are given to be $\rho = 2702$ kg/m^3 and $c_p = 0.896$ kJ/kg.°C.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = \rho(\pi r_0^2)\boldsymbol{V} = (2702\,\text{kg/m}^3)\pi(0.0025\,\text{m})^2(8\,\text{m/min}) = 0.4244\,\text{kg/min} = 0.007074\,\text{kg/s}$$

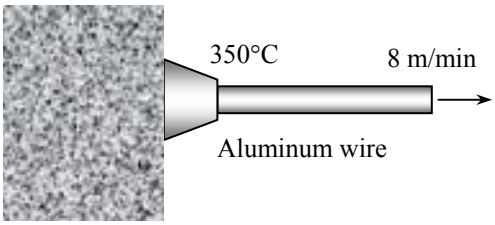
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{wire}} = \dot{m}_{\text{wire}} c_p (T_1 - T_2)$$



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.007074\,\text{kg/s})(0.896\,\text{kJ/kg.°C})(350 - 50)°\text{C} = \mathbf{1.90\,kW}$$

5-168 Long copper wires are extruded at a velocity of 8 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

Properties The properties of copper are given to be ρ = 8950 kg/m^3 and $c_p = 0.383$ kJ/kg.°C.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = \rho(\pi r_0^2)\boldsymbol{V} = (8950\,\text{kg/m}^3)\pi(0.0025\,\text{m})^2(8/60\,\text{m/s}) = 0.02343\,\text{kg/s}$$

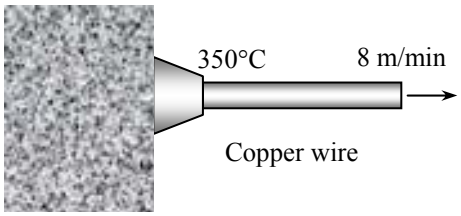
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{wire}} = \dot{m}_{\text{wire}} c_p (T_1 - T_2)$$



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.02343\,\text{kg/s})(0.383\,\text{kJ/kg.°C})(350 - 50)°\text{C} = \mathbf{2.69\,kW}$$

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5-169 Steam at a saturation temperature of T_{sat} = 40°C condenses on the outside of a thin horizontal tube. Heat is transferred to the cooling water that enters the tube at 25°C and exits at 35°C. The rate of condensation of steam is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The changes in kinetic and potential energies are negligible.

Properties The properties of water at room temperature are ρ = 997 kg/m^3 and c_p = 4.18 kJ/kg.°C (Table A-3). The enthalpy of vaporization of water at 40°C is h_{fg} = 2406.0 kJ/kg (Table A-4).

Analysis The mass flow rate of water through the tube is

$$\dot{m}_{water} = \rho V A_c = (997\ \text{kg/m}^3)(2\ \text{m/s})[\pi(0.03\ \text{m})^2/4] = 1.409\ \text{kg/s}$$

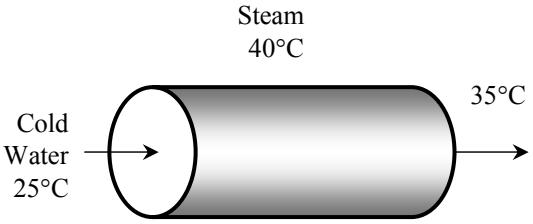
Taking the volume occupied by the cold water in the tube as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}}^{\nearrow 0\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{in} = \dot{Q}_{water} = \dot{m}_{water} c_p (T_2 - T_1)$$



Then the rate of heat transfer to the water and the rate of condensation become

$$\dot{Q} = \dot{m} c_p (T_{out} - T_{in}) = (1.409\ \text{kg/s})(4.18\ \text{kJ/kg} \cdot °\text{C})(35 - 25)°\text{C} = 58.9\ \text{kW}$$

$$\dot{Q} = \dot{m}_{cond} h_{fg} \rightarrow \dot{m}_{cond} = \frac{\dot{Q}}{h_{fg}} = \frac{58.9\ \text{kJ/s}}{2406.0\ \text{kJ/kg}} = \mathbf{0.0245\ kg/s}$$

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5-170E Steam is mixed with water steadily in an adiabatic device. The temperature of the water leaving this device is to be determined.

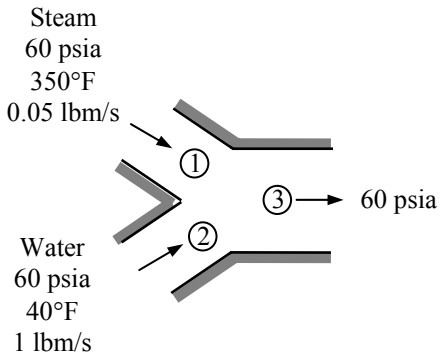
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions. **4** There is no heat transfer between the mixing device and the surroundings.

Analysis We take the mixing device as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3$$



From a mass balance

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.05 + 1 = 1.05 \text{ lbm/s}$$

The enthalpies of steam and water are (Table A-6E and A-4E)

$$\left.\begin{array}{l} P_1 = 60 \text{ psia} \\ T_1 = 350°\text{F} \end{array}\right\} h_1 = 1208.3 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 60 \text{ psia} \\ T_2 = 40°\text{F} \end{array}\right\} h_2 \cong h_{f\,@\,40°\text{F}} = 8.032 \text{ Btu/lbm}$$

Substituting into the energy balance equation solving for the exit enthalpy gives

$$h_3 = \frac{\dot{m}_1 h_1 + \dot{m}_2 h_2}{\dot{m}_3} = \frac{(0.05 \text{ lbm/s})(1208.3 \text{ Btu/lbm}) + (1 \text{ lbm/s})(8.032 \text{ Btu/lbm})}{1.05 \text{ lbm/s}} = 65.19 \text{ Btu/lbm}$$

At the exit state P_3=60 psia and h_3=65.19 kJ/kg. An investigation of Table A-5E reveals that this is compressed liquid state. Approximating this state as saturated liquid at the specified temperature, the temperature may be determined from Table A-4E to be

$$\left.\begin{array}{l} P_3 = 60 \text{ psia} \\ h_3 = 65.19 \text{ Btu/lbm} \end{array}\right\} T_3 \cong T_{f\,@\,h=65.19 \text{ Btu/lbm}} = \mathbf{97.2°F}$$

Discussion The exact answer is determined at the compressed liquid state using EES to be **97.0°F**, indicating that the saturated liquid approximation is a reasonable one.

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5-171 A constant-pressure R-134a vapor separation unit separates the liquid and vapor portions of a saturated mixture into two separate outlet streams. The flow power needed to operate this unit and the mass flow rate of the two outlet streams are to be determined.

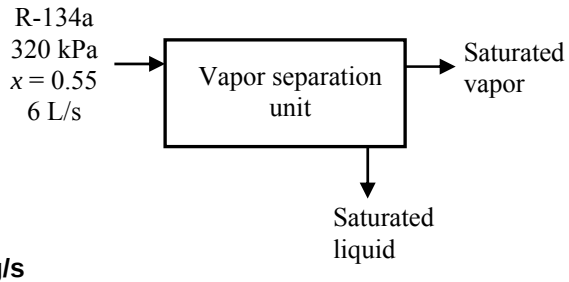
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions.

Analysis The specific volume at the inlet is (Table A-12)

$$\left.\begin{array}{l} P_1 = 320\ \text{kPa} \\ x_1 = 0.55 \end{array}\right\} \nu_1 = \nu_f + x_1(\nu_g - \nu_f) = 0.0007772 + (0.55)(0.06360 - 0.0007772) = 0.03533\ \text{m}^3/\text{kg}$$

The mass flow rate at the inlet is then

$$\dot{m}_1 = \frac{\dot{V}_1}{\nu_1} = \frac{0.006\ \text{m}^3/\text{s}}{0.03533\ \text{m}^3/\text{kg}} = 0.1698\ \text{kg/s}$$



For each kg of mixture processed, 0.55 kg of vapor are processed. Therefore,

$$\dot{m}_2 = 0.7\dot{m}_1 = 0.55 \times 0.1698 = \mathbf{0.09340\ kg/s}$$
$$\dot{m}_3 = \dot{m}_1 - \dot{m}_2 = 0.45\dot{m}_1 = 0.45 \times 0.1698 = \mathbf{0.07642\ kg/s}$$

The flow power for this unit is

$$\begin{aligned} \dot{W}_{\text{flow}} &= \dot{m}_2 P_2 \nu_2 + \dot{m}_3 P_3 \nu_3 - \dot{m}_1 P_1 \nu_1 \\ &= (0.09340\ \text{kg/s})(320\ \text{kPa})(0.06360\ \text{m}^3/\text{kg}) + (0.07642\ \text{kg/s})(320\ \text{kPa})(0.0007772\ \text{m}^3/\text{kg}) \\ &\quad - (0.1698\ \text{kg/s})(320\ \text{kPa})(0.03533\ \text{m}^3/\text{kg}) \\ &= \mathbf{0\ kW} \end{aligned}$$

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5-172E A small positioning control rocket in a satellite is driven by a container filled with R-134a at saturated liquid state. The number of bursts this rocket experience before the quality in the container is 90% or more is to be determined.

Analysis The initial and final specific volumes are

$$\left.\begin{array}{l} T_1 = -10°\text{F} \\ x_1 = 0 \end{array}\right\} \nu_1 = 0.01171\,\text{ft}^3/\text{lbm} \quad \text{(Table A-11E)}$$

$$\left.\begin{array}{l} T_2 = -10°\text{F} \\ x_2 = 0.90 \end{array}\right\} \nu_2 = \nu_f + x_2\nu_{fg} = 0.01171 + (0.90)(2.7091 - 0.01171) = 2.4394\,\text{ft}^3/\text{lbm}$$

The initial and final masses in the container are

$$m_1 = \frac{V}{\nu_1} = \frac{2\,\text{ft}^3}{0.01171\,\text{m}^3/\text{kg}} = 170.8\,\text{lbm}$$

$$m_2 = \frac{V}{\nu_2} = \frac{2\,\text{ft}^3}{2.4394\,\text{ft}^3/\text{lbm}} = 0.8199\,\text{lbm}$$

Then,

$$\Delta m = m_1 - m_2 = 170.8 - 0.8199 = 170.0\,\text{lbm}$$

The amount of mass released during each control burst is

$$\Delta m_b = \dot{m}\Delta t = (0.05\,\text{lbm/s})(5\,\text{s}) = 0.25\,\text{lbm}$$

The number of bursts that can be executed is then

$$N_b = \frac{\Delta m}{\Delta m_b} = \frac{170.0\,\text{lbm}}{0.25\,\text{lbm/burst}} = \mathbf{680\ bursts}$$

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5-173E The relationships between the mass flow rate and the time for the inflation and deflation of an air bag are given. The volume of this bag as a function of time are to be plotted.

Assumptions Uniform flow exists at the inlet and outlet.

Properties The specific volume of air during inflation and deflation are given to be 15 and 13 ft^3/lbm, respectively.

Analysis The volume of the airbag at any time is given by

$$V(t) = \int_{\text{in flow time}} (\dot{m}v)_{\text{in}}\, dt - \int_{\text{out flow time}} (\dot{m}v)_{\text{out}}\, d$$

Applying at different time periods as given in problem statement give

$$V(t) = \int_0^t (15\ \text{ft}^3/\text{lbm})\frac{20\ \text{lbm/s}}{10\ \text{ms}}\left(\frac{1\ \text{s}}{1000\ \text{ms}}\right)t dt \quad 0 \le t \le 10\ \text{ms}$$

$$V(t) = \int_0^t 0.015t^2\ \text{ft}^3/\text{ms}^2 \quad 0 \le t \le 10\ \text{ms}$$

$$V(t) = V(10\ \text{ms}) + \int_{10\ \text{ms}}^t (15\ \text{ft}^3/\text{lbm})(20\ \text{lbm/s})\left(\frac{1\ \text{s}}{1000\ \text{ms}}\right)t dt \quad 10 < t \le 12\ \text{ms}$$

$$V(t) = V(10\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2\,(t - 10\ \text{ms}) \quad 10 < t \le 12\ \text{ms}$$

$$V(t) = V(12\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2\,(t - 12\ \text{ms}) - \int_{12\ \text{ms}}^t (13\ \text{ft}^3/\text{lbm})\frac{16\ \text{lbm/s}}{(30\text{-}12)\ \text{ms}}\left(\frac{1\ \text{s}}{1000\ \text{ms}}\right)(t - 12\ \text{ms})dt \quad 12 < t \le 25\ \text{ms}$$

$$V(t) = V(12\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2\,(t - 12\ \text{ms}) - \int_{12\ \text{ms}}^t 0.011556\ \text{ft}^3/\text{ms}^2\,(t - 12\ \text{ms})dt \quad 12 < t \le 25\ \text{ms}$$

$$V(t) = V(12\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2\,(t - 12\ \text{ms}) - \frac{0.011556}{2}(t^2 - 144\ \text{ms}^2) + 0.13867(t - 12\ \text{ms}) \quad 12 < t \le 25\ \text{ms}$$

$$V(t) = V(25\ \text{ms}) - \frac{0.011556}{2}(t^2 - 625\ \text{ms}^2) + 0.13867(t - 25\ \text{ms}) \quad 25 < t \le 30\ \text{ms}$$

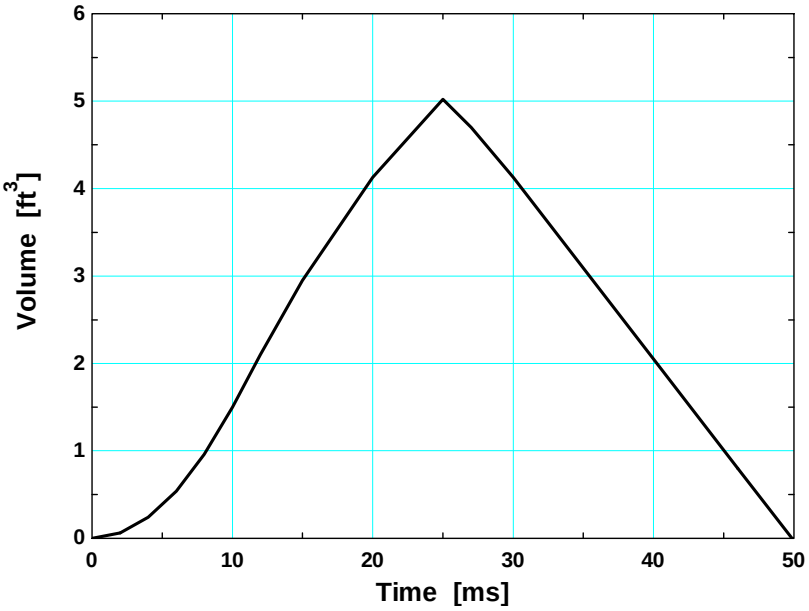
$$V(t) = V(30\ \text{ms}) - \int_{12\ \text{ms}}^t (13\ \text{ft}^3/\text{lbm})(16\ \text{lbm/s})\left(\frac{1\ \text{s}}{1000\ \text{ms}}\right)dt \quad 30 < t \le 50\ \text{ms}$$

$$V(t) = V(30\ \text{ms}) - (0.208\ \text{ft}^3/\text{ms})(t - 30\ \text{ms}) \quad 30 < t \le 50\ \text{ms}$$

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The results with some suitable time intervals are

Time, ms	V, ft^3
0	0
2	0.06
4	0.24
6	0.54
8	0.96
10	1.50
12	2.10
15	2.95
20	4.13
25	5.02
27	4.70
30	4.13
40	2.05
46	0.80
49.85	0



Alternative solution

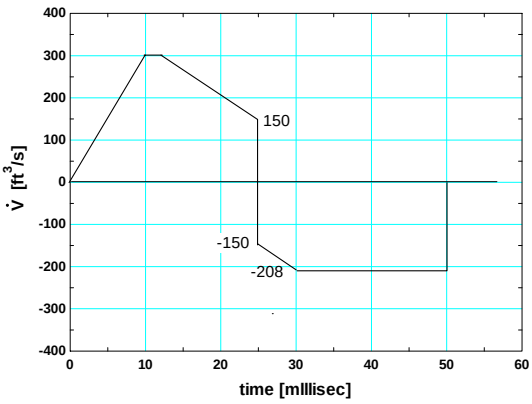
The net volume flow rate is obtained from

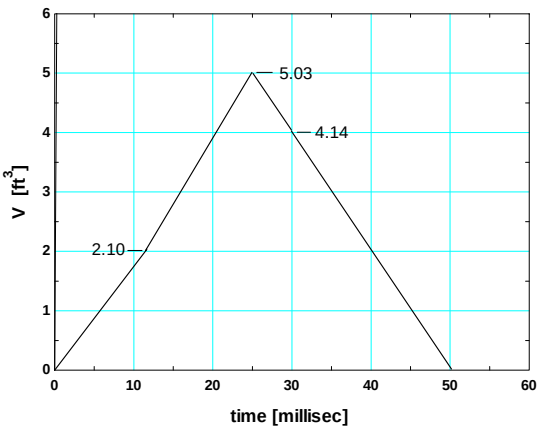
$$\dot{\forall} = (\dot{m}v)_{\text{in}} - (\dot{m}v)_{\text{out}}$$

which is sketched on the figure below. The volume of the airbag is given by

$$\forall = \int \dot{\forall} dt$$

The results of a graphical interpretation of the volume is also given in the figure below. Note that the evaluation of the above integral is simply the area under the process curve.



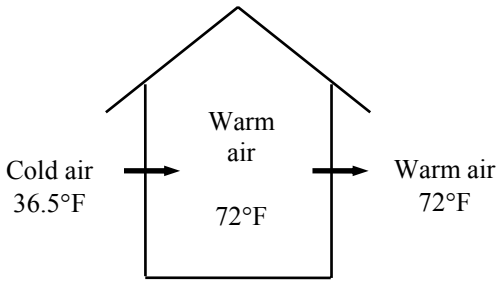


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5-174E A winterizing project is to reduce the infiltration rate of a house from 2.2 ACH to 1.1 ACH. The resulting cost savings are to be determined.

Assumptions **1** The house is maintained at 72°F at all times. **2** The latent heat load during the heating season is negligible. **3** The infiltrating air is heated to 72°F before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm·R (Table A-1E). The specific heat of air at room temperature is 0.24 Btu/lbm·°F (Table A-2E).



Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5 \text{ psia}}{(0.3704 \text{ psia.ft}^3/\text{lbm.R})(496.5 \text{ R})} = 0.0734 \text{ lbm/ft}^3$$

The volume of the house is

$$\boldsymbol{V}_{\text{building}} = (\text{Floor area})(\text{Height}) = (4500 \text{ ft}^2)(9 \text{ ft}) = 40{,}500 \text{ ft}^3$$

We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the house. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = \rho \dot{\boldsymbol{V}} c_p(T_2 - T_1)$$

The reduction in the infiltration rate is 2.2 – 1.1 = 1.1 ACH. The reduction in the sensible infiltration heat load corresponding to it is

$$\dot{Q}_{\text{infiltration, saved}} = \rho_o c_p (ACH_{\text{saved}})(\boldsymbol{V}_{\text{building}})(T_i - T_o)$$

$$= (0.0734 \text{ lbm/ft}^3)(0.24 \text{ Btu/lbm.°F})(1.1/\text{h})(40{,}500 \text{ ft}^3)(72 - 36.5)°\text{F}$$

$$= 27{,}860 \text{ Btu/h} = 0.2786 \text{ therm/h}$$

since 1 therm = 100,000 Btu. The number of hours during a six month period is 6×30×24 = 4320 h. Noting that the furnace efficiency is 0.92 and the unit cost of natural gas is $1.24/therm, the energy and money saved during the 6-month period are

$$\text{Energy savings} = (\dot{Q}_{\text{infiltration, saved}})(\text{No. of hours per year})/\text{Efficiency}$$

$$= (0.2786 \text{ therm/h})(4320 \text{ h/year})/0.92$$

$$= 1308 \text{ therms/year}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of energy})$$

$$= (1308 \text{ therms/year})(\$1.24/\text{therm})$$

$$= \mathbf{\$1622/year}$$

Therefore, reducing the infiltration rate by one-half will reduce the heating costs of this homeowner by $1622 per year.

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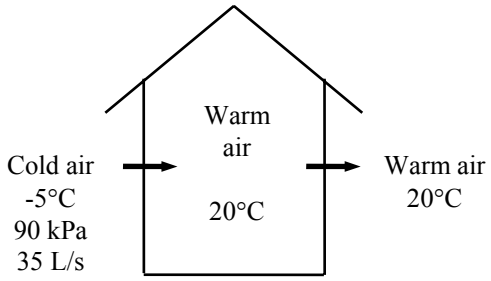
5-175 Outdoors air at -5°C and 90 kPa enters the building at a rate of 35 L/s while the indoors is maintained at 20°C. The rate of sensible heat loss from the building due to infiltration is to be determined.

Assumptions **1** The house is maintained at 20°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 20°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg·K. The specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2).

Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{90\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(-5+273\ \text{K})} = 1.17\ \text{kg/m}^3$$



We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the sensible infiltration heat load corresponding to an infiltration rate of 35 L/s becomes

$$\dot{Q}_{\text{infiltration}} = \rho_o \dot{V}_{air} c_p (T_i - T_o)$$

$$= (1.17\ \text{kg/m}^3)(0.035\ \text{m}^3/\text{s})(1.005\ \text{kJ/kg.°C})[20-(-5)]\text{°C}$$

$$= \mathbf{1.029\ kW}$$

Therefore, sensible heat will be lost at a rate of 1.029 kJ/s due to infiltration.

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5-176 Chilled air is to cool a room by removing the heat generated in a large insulated classroom by lights and students. The required flow rate of air that needs to be supplied to the room is to be determined.

Assumptions **1** The moisture produced by the bodies leave the room as vapor without any condensing, and thus the classroom has no latent heat load. **2** Heat gain through the walls and the roof is negligible. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady operating conditions exist.

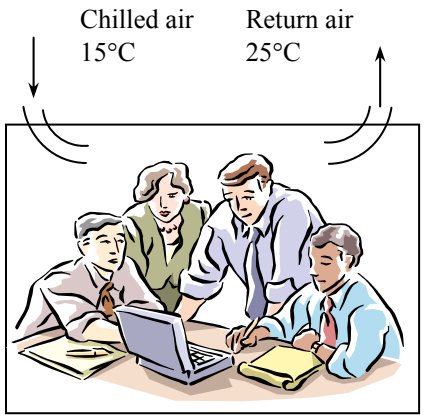
Properties The specific heat of air at room temperature is 1.005 kJ/kg·°C (Table A-2). The average rate of sensible heat generation by a person is given to be 60 W.

Analysis The rate of sensible heat generation by the people in the room and the total rate of sensible internal heat generation are

$$\dot{Q}_{\text{gen, sensible}} = \dot{q}_{\text{gen, sensible}}(\text{No. of people}) = (60\ \text{W/person})(150\ \text{persons}) = 9000\ \text{W}$$

$$\dot{Q}_{\text{total, sensible}} = \dot{Q}_{\text{gen, sensible}} + \dot{Q}_{\text{lighting}} = 9000 + 6000 = 15{,}000\ \text{W}$$

Both of these effects can be viewed as heat gain for the chilled air stream, which can be viewed as a steady stream of cool air that is heated as it flows in an imaginary duct passing through the room. The energy balance for this imaginary steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{total, sensible}} = \dot{m}c_p(T_2 - T_1)$$

Then the required mass flow rate of chilled air becomes

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{total, sensible}}}{c_p \Delta T} = \frac{15\ \text{kJ/s}}{(1.005\ \text{kJ/kg} \cdot °\text{C})(25 - 15)°\text{C}} = \mathbf{1.49\ kg/s}$$

Discussion The latent heat will be removed by the air-conditioning system as the moisture condenses outside the cooling coils.

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5-177 A fan is powered by a 0.5 hp motor, and delivers air at a rate of 85 m^3/min. The highest possible air velocity at the fan exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity and the change in potential energy are negligible, $V_1 \cong 0$ and $\Delta pe \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** The efficiencies of the motor and the fan are 100% since best possible operation is assumed. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be ρ = 1.18 kg/m^3. The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

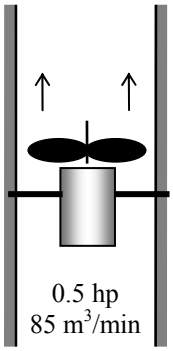
Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

The velocity of air leaving the fan will be highest when all of the entire electrical energy drawn by the motor is converted to kinetic energy, and the friction between the air layers is zero. In this best possible case, no energy will be converted to thermal energy, and thus the temperature change of air will be zero, $T_2 = T_1$. Then the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{e,in}} + \dot{m}h_1 = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } V_1 \cong 0 \text{ and } \Delta\text{pe} \cong 0)$$



Noting that the temperature and thus enthalpy remains constant, the relation above simplifies further to

$$\dot{W}_{e,in} = \dot{m}V_2^2/2$$

where

$$\dot{m} = \rho\dot{V} = (1.18\,\text{kg/m}^3)(85\,\text{m}^3/\text{min}) = 100.3\,\text{kg/min} = 1.67\,\text{kg/s}$$

Solving for V_2 and substituting gives

$$V_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} = \sqrt{\frac{2(0.5\,\text{hp})}{1.67\,\text{kg/s}}\left(\frac{745.7\,\text{W}}{1\,\text{hp}}\right)\left(\frac{1\,\text{m}^2/\text{s}^2}{1\,\text{W}}\right)} = \mathbf{21.1\ m/s}$$

Discussion In reality, the velocity will be less because of the inefficiencies of the motor and the fan.

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5-178 The average air velocity in the circular duct of an air-conditioning system is not to exceed 8 m/s. If the fan converts 80 percent of the electrical energy into kinetic energy, the size of the fan motor needed and the diameter of the main duct are to be determined.

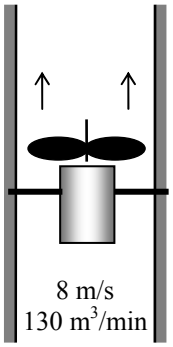
Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity is negligible, $V_1 \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be $\rho = 1.20$ kg/m^3. The constant pressure specific heat of air at room temperature is $c_p = 1.005$ kJ/kg.°C (Table A-2).

Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The change in the kinetic energy of air as it is accelerated from zero to 8 m/s at a rate of 130 m^3/min is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1.20 \text{ kg/m}^3)(130 \text{ m}^3/\text{min}) = 156 \text{ kg/min} = 2.6 \text{ kg/s}$$

$$\Delta \dot{\text{KE}} = \dot{m}\frac{V_2^2 - V_1^2}{2} = (2.6 \text{ kg/s})\frac{(8 \text{ m/s})^2 - 0}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 0.0832 \text{ kW}$$



It is stated that this represents 80% of the electrical energy consumed by the motor. Then the total electrical power consumed by the motor is determined to be

$$0.7\dot{W}_{\text{motor}} = \Delta K\dot{E} \;\rightarrow\; \dot{W}_{\text{motor}} = \frac{\Delta K\dot{E}}{0.8} = \frac{0.0832 \text{ kW}}{0.8} = \mathbf{0.104 \text{ kW}}$$

The diameter of the main duct is

$$\dot{\boldsymbol{V}} = VA = V(\pi D^2/4) \;\rightarrow\; D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(130 \text{ m}^3/\text{min})}{\pi(8 \text{ m/s})}\left(\frac{1 \text{ min}}{60 \text{ s}}\right)} = \mathbf{0.587 \text{ m}}$$

Therefore, the motor should have a rated power of at least 0.104 kW, and the diameter of the duct should be at least 58.7 cm

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5-179 An adiabatic air compressor is powered by a direct-coupled steam turbine, which is also driving a generator. The net power delivered to the generator is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The devices are adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Properties From the steam tables (Tables A-4 through 6)

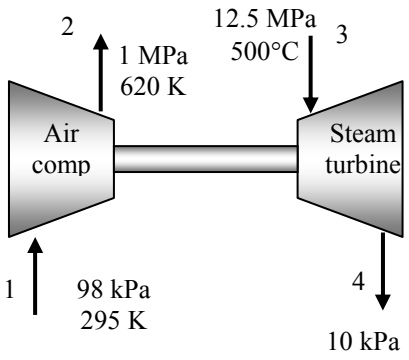
$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} h_3 = 3343.6 \text{ kJ/kg}$$

and

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg}$$

From the air table (Table A-17),

$$T_1 = 295 \text{ K} \longrightarrow h_1 = 295.17 \text{ kJ/kg}$$
$$T_2 = 620 \text{ K} \longrightarrow h_2 = 628.07 \text{ kJ/kg}$$



Analysis There is only one inlet and one exit for either device, and thus $\dot{m}_{\text{in}} = \dot{m}_{\text{out}} = \dot{m}$. We take either the turbine or the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for either steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \nearrow^{0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

For the turbine and the compressor it becomes

Compressor:

$$\dot{W}_{\text{comp, in}} + \dot{m}_{\text{air}} h_1 = \dot{m}_{\text{air}} h_2 \quad \rightarrow \quad \dot{W}_{\text{comp, in}} = \dot{m}_{\text{air}} (h_2 - h_1)$$

Turbine:

$$\dot{m}_{\text{steam}} h_3 = \dot{W}_{\text{turb, out}} + \dot{m}_{\text{steam}} h_4 \quad \rightarrow \quad \dot{W}_{\text{turb, out}} = \dot{m}_{\text{steam}} (h_3 - h_4)$$

Substituting,

$$\dot{W}_{\text{comp,in}} = (10 \text{ kg/s})(628.07 - 295.17) \text{ kJ/kg} = 3329 \text{ kW}$$
$$\dot{W}_{\text{turb,out}} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{ kJ/kg} = 23{,}777 \text{ kW}$$

Therefore,

$$\dot{W}_{\text{net,out}} = \dot{W}_{\text{turb,out}} - \dot{W}_{\text{comp,in}} = 23{,}777 - 3329 = \mathbf{20{,}448 \text{ kW}}$$

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5-180 Helium is compressed by a compressor. The power required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats. **4** The compressor is adiabatic.

Properties The constant pressure specific heat of helium is $c_p = 5.1926$ kJ/kg·K. The gas constant is $R = 2.0769$ kJ/kg·K (Table A-2a).

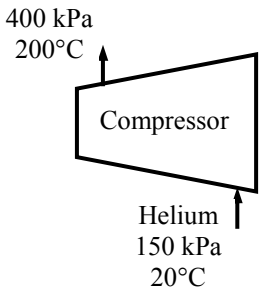
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\,\text{K})}{150\,\text{kPa}} = 4.0569\,\text{m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{(0.1\,\text{m}^2)(15\,\text{m/s})}{4.0569\,\text{m}^3/\text{kg}} = 0.3697\,\text{kg/s}$$

Then the power input is determined from the energy balance equation to be

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = (0.3697\,\text{kg/s})(5.1926\,\text{kJ/kg}\cdot\text{K})(200-20)\text{K} = \mathbf{345.5\ kW}$$

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5-181 Saturated R-134a vapor is compressed to a specified state. The power input is given. The rate of heat transfer is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties From the R-134a tables (Table A-11)

$$\left.\begin{array}{l} T_1 = 10°\text{C} \\ x_1 = 1 \end{array}\right\} h_1 = 256.16\ \text{kJ/kg}$$

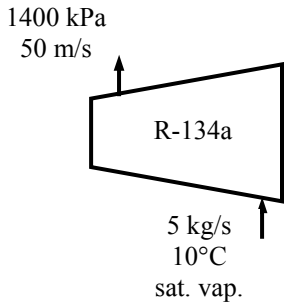
Analysis We take the compressor as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the compressor, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} + \dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) \quad (\text{since } \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} + \dot{m}\left(h_1 - h_2 - \frac{V_2^2}{2}\right)$$



Substituting,

$$\begin{aligned} \dot{Q}_{\text{out}} &= \dot{W}_{\text{in}} + \dot{m}\left(h_1 - h_2 - \frac{V_2^2}{2}\right) \\ &= 132.4\ \text{kW} + (5\ \text{kg/s})\left[256.116 - 281.39)\text{kJ/kg} - \frac{(50\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right] \\ &= \mathbf{0.02\ kW} \end{aligned}$$

which is practically zero and therefore the process is adiabatic.

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5-182 A submarine that has an air-ballast tank originally partially filled with air is considered. Air is pumped into the ballast tank until it is entirely filled with air. The final temperature and mass of the air in the ballast tank are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** The process is adiabatic. **3** There are no work interactions.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1). The specific heat ratio of air at room temperature is $k = 1.4$ (Table A-2*a*). The specific volume of water is taken 0.001 m^3/kg.

Analysis The conservation of mass principle applied to the air gives

$$\frac{dm_a}{dt} = \dot{m}_{\text{in}}$$

and as applied to the water becomes

$$\frac{dm_w}{dt} = -\dot{m}_{\text{out}}$$

The first law for the ballast tank produces

$$0 = \frac{d(mu)_a}{dt} + \frac{d(mu)_w}{dt} + h_w\dot{m}_w - h_a\dot{m}_a$$

Combining this with the conservation of mass expressions, rearranging and canceling the common *dt* term produces

$$d(mu)_a + d(mu)_w = h_a dm_a + h_w dm_w$$

Integrating this result from the beginning to the end of the process gives

$$\left[(mu)_2 - (mu)_1\right]_a + \left[(mu)_2 - (mu)_1\right]_w = h_a(m_2 - m_1)_a + h_w(m_2 - m_1)_w$$

Substituting the ideal gas equation of state and the specific heat models for the enthalpies and internal energies expands this to

$$\frac{P\boldsymbol{V}_2}{RT_2}c_{\boldsymbol{v}}T_2 - \frac{P\boldsymbol{V}_1}{RT_1}c_{\boldsymbol{v}}T_1 - m_{w,1}c_wT_w = c_pT_{\text{in}}\left(\frac{P\boldsymbol{V}_2}{RT_2} - \frac{P\boldsymbol{V}_1}{RT_1}\right) - m_{w,1}c_wT_w$$

When the common terms are cancelled, this result becomes

$$T_2 = \frac{\boldsymbol{V}_2}{\dfrac{\boldsymbol{V}_1}{T_1} + \dfrac{1}{kT_{\text{in}}}(\boldsymbol{V}_2 - \boldsymbol{V}_1)} = \frac{700}{\dfrac{100}{288} + \dfrac{1}{(1.4)(293)}(700 - 100)} = \mathbf{386.8\ K}$$

The final mass from the ideal gas relation is

$$m_2 = \frac{P\boldsymbol{V}_2}{RT_2} = \frac{(1500\text{ kPa})(700\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(386.8\text{ K})} = \mathbf{9460\ kg}$$

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5-183 A submarine that has an air-ballast tank originally partially filled with air is considered. Air is pumped into the ballast tank in an isothermal manner until it is entirely filled with air. The final mass of the air in the ballast tank and the total heat transfer are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** There are no work interactions.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1). The specific heats of air at room temperature are $c_p = 1.005$ kJ/kg·K and $c_v = 0.718$ kJ/kg·K (Table A-2*a*). The specific volume of water is taken 0.001 m^3/kg.

Analysis The initial air mass is

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(1500\ \text{kPa})(100\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288.15\ \text{K})} = 1814\ \text{kg}$$

and the initial water mass is

$$m_w = \frac{V_1}{v_1} = \frac{600\ \text{m}^3}{0.001\ \text{m}^3/\text{kg}} = 600{,}000\ \text{kg}$$

and the final mass of air in the tank is

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(1500\ \text{kPa})(700\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288.15\ \text{K})} = \mathbf{12{,}697\ kg}$$

The first law when adapted to this system gives

$$\begin{aligned} Q_{\text{in}} + m_i h_i - m_e h_e &= m_2 u_2 - m_1 u_1 \\ Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + m_e h_e - m_i h_i \\ Q_{\text{in}} &= m_2 c_v T - (m_{a,1} c_v T + m_w u_w) + m_w h_w - (m_2 - m_1) c_p T \end{aligned}$$

Noting that

$$u_w \cong h_w = 62.98\ \text{kJ/kg}$$

Substituting,

$$\begin{aligned} Q_{\text{in}} &= 12{,}697 \times 0.718 \times 288 - (1814 \times 0.718 \times 288 + 600{,}000 \times 62.98) \\ &\quad + 600{,}000 \times 62.98 - (12{,}697 - 1814) \times 1.005 \times 288 \\ &= \mathbf{0\ kJ} \end{aligned}$$

The process is adiabatic.

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5-184 A cylindrical tank is charged with nitrogen from a supply line. The final mass of nitrogen in the tank and final temperature are to be determined for two cases.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved.

Properties The gas constant of nitrogen is 0.2968 kPa·m^3/kg·K (Table A-1). The specific heats of nitrogen at room temperature are c_p = 1.039 kJ/kg·K and c_v = 0.743 kJ/kg·K (Table A-2*a*).

Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

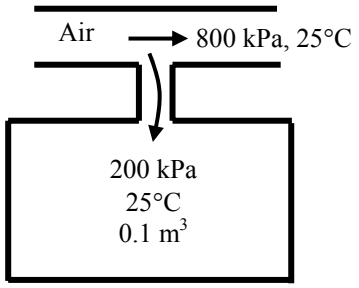
$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = m_2 u_2 - m_1 u_1$$

$$m_i c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$



Combining the two balances:

$$(m_2 - m_1) c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{R T_1} = \frac{(200\,\text{kPa})(0.1\,\text{m}^3)}{(0.2968\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\,\text{K})} = 0.2261\,\text{kg}$$

$$m_2 = \frac{P_2 V}{R T_2} = \frac{(800\,\text{kPa})(0.1\,\text{m}^3)}{(0.2968\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}) T_2} = \frac{269.5}{T_2}$$

Substituting,

$$\left(\frac{269.5}{T_2} - 0.2261\right)(1.039)(298) = \frac{269.5}{T_2}(0.743)T_2 - (0.2261)(0.743)(298)$$

whose solution is

$$T_2 = \mathbf{379.0\,K}$$

The final mass is then

$$m_2 = \frac{269.5}{T_2} = \frac{269.5}{379.0} = \mathbf{0.7112\,kg}$$

(*b*) When there is rapid heat transfer between the nitrogen and tank such that the cylinder and nitrogen remain in thermal equilibrium during the process, the energy balance equation may be written as

$$(m_2 - m_1) c_p T_i = (m_{nit,2} c_v T_2 + m_t c_t T_2) - (m_{nit,1} c_v T_1 + m_t c_t T_1)$$

Substituting,

$$\left(\frac{269.5}{T_2} - 0.2261\right)(1.039)(298) = \left[\frac{269.5}{T_2}(0.743)T_2 + (50)(0.43)T_2\right] - \left[(0.2261)(0.743)(298) + (50)(0.43)(298)\right]$$

whose solution is

$$T_2 = \mathbf{300.7\,K}$$

The final mass is then

$$m_2 = \frac{269.5}{300.7} = \frac{269.5}{300.7} = \mathbf{0.8962\,kg}$$

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5-185 The air in a tank is released until the pressure in the tank reduces to a specified value. The mass withdrawn from the tank is to be determined for three methods of analysis.

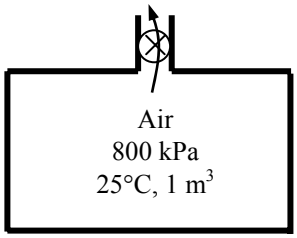
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work or heat interactions involved.

Properties The gas constant of air is 0.287 kPa·m^3/kg·K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 kJ/kg·K and c_v = 0.718 kJ/kg·K. Also k =1.4 (Table A-2*a*).

Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$
$$-m_e h_e = m_2 u_2 - m_1 u_1$$
$$0 = m_2 u_2 - m_1 u_1 + m_e h_e$$
$$0 = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$

Combining the two balances:

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{R T_1} = \frac{(800\ \text{kPa})(1\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(25+273\ \text{K})} = 9.354\ \text{kg}$$

$$m_2 = \frac{P_2 V}{R T_2} = \frac{(150\ \text{kPa})(1\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}) T_2} = \frac{522.6}{T_2}$$

The temperature of air leaving the tank changes from the initial temperature in the tank to the final temperature during the discharging process. We assume that the temperature of the air leaving the tank is the average of initial and final temperatures in the tank. Substituting into the energy balance equation gives

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$
$$0 = \frac{522.6}{T_2}(0.718) T_2 - (9.354)(0.718)(298) + \left(9.354 - \frac{522.6}{T_2}\right)(1.005)\left(\frac{298 + T_2}{2}\right)$$

whose solution is

$$T_2 = 191.0\ \text{K}$$

Substituting, the final mass is

$$m_2 = \frac{522.6}{191} = 2.736\ \text{kg}$$

and the mass withdrawn is

$$m_e = m_1 - m_2 = 9.354 - 2.736 = \mathbf{6.618\ kg}$$

(*b*) Considering the process in two parts, first from 800 kPa to 400 kPa and from 400 kPa to 150 kPa, the solution will be as follows:

From 800 kPa to 400 kPa:

$$m_2 = \frac{P_2 V}{R T_2} = \frac{(400\ \text{kPa})(1\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}) T_2} = \frac{1394}{T_2}$$

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$$0 = \frac{1394}{T_2}(0.718)T_2 - (9.354)(0.718)(298) + \left(9.354 - \frac{1394}{T_2}\right)(1.005)\left(\frac{298 + T_2}{2}\right)$$

$$T_2 = 245.1\,\text{K}$$

$$m_2 = \frac{1394}{245.1} = 5.687\,\text{kg}$$

$$m_{e,1} = m_1 - m_2 = 9.354 - 5.687 = 3.667\,\text{kg}$$

From 400 kPa to 150 kPa:

$$0 = \frac{522.6}{T_2}(0.718)T_2 - (5.687)(0.718)(245.1) + \left(5.687 - \frac{522.6}{T_2}\right)(1.005)\left(\frac{245.1 + T_2}{2}\right)$$

$$T_2 = 186.5\,\text{K}$$

$$m_2 = \frac{522.6}{186.5} = 2.803\,\text{kg}$$

$$m_{e,2} = m_1 - m_2 = 5.687 - 2.803 = 2.884\,\text{kg}$$

The total mass withdrawn is

$$m_e = m_{e,1} + m_{e,2} = 3.667 + 2.884 = \mathbf{6.551\,kg}$$

(*c*) The mass balance may be written as

$$\frac{dm}{dt} = -\dot{m}_e$$

When this is combined with the ideal gas equation of state, it becomes

$$\frac{\boldsymbol{V}}{R}\frac{d(P/T)}{dt} = -\dot{m}_e$$

since the tank volume remains constant during the process. An energy balance on the tank gives

$$\frac{d(mu)}{dt} = -h_e\dot{m}_e$$

$$c_v\frac{d(mT)}{dt} = c_pT\frac{dm}{dt}$$

$$c_v\frac{\boldsymbol{V}}{R}\frac{dP}{dt} = c_pT\frac{\boldsymbol{V}}{R}\frac{d(P/T)}{dt}$$

$$c_v\frac{dP}{dt} = c_p\left(\frac{dP}{dt} - \frac{P}{T}\frac{dT}{dt}\right)$$

$$(c_p - c_v)\frac{dP}{P} = c_p\frac{dT}{dt}$$

When this result is integrated, it gives

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (298\,\text{K})\left(\frac{150\,\text{kPa}}{800\,\text{kPa}}\right)^{0.4/1.4} = 184.7\,\text{K}$$

The final mass is

$$m_2 = \frac{P_2\boldsymbol{V}}{RT_2} = \frac{(150\,\text{kPa})(1\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.7\,\text{K})} = 2.830\,\text{kg}$$

and the mass withdrawn is

$$m_e = m_1 - m_2 = 9.354 - 2.830 = \mathbf{6.524\,kg}$$

Discussion The result in first method is in error by 1.4% while that in the second method is in error by 0.4%.

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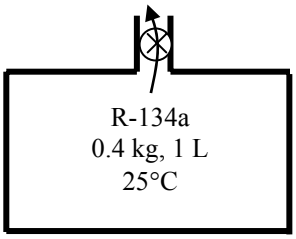
5-186 A tank initially contains saturated mixture of R-134a. A valve is opened and R-134a vapor only is allowed to escape slowly such that temperature remains constant. The heat transfer necessary with the surroundings to maintain the temperature and pressure of the R-134a constant is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the exit remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$\begin{aligned} m_{\text{in}} - m_{\text{out}} &= \Delta m_{\text{system}} \\ -m_e &= m_2 - m_1 \\ m_e &= m_1 - m_2 \end{aligned}$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$\begin{aligned} Q_{\text{in}} - m_e h_e &= m_2 u_2 - m_1 u_1 \\ Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + m_e h_e \end{aligned}$$

Combining the two balances:

$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$

The specific volume at the initial state is

$$v_1 = \frac{V}{m_1} = \frac{0.001\,\text{m}^3}{0.4\,\text{kg}} = 0.0025\,\text{m}^3/\text{kg}$$

The initial state properties of R-134a in the tank are

$$\left.\begin{aligned} T_1 &= 26°\text{C} \\ v_1 &= 0.0025\,\text{m}^3/\text{kg} \end{aligned}\right\} \begin{aligned} x_1 &= \frac{v_1 - v_f}{v_{fg}} = \frac{0.0025 - 0.0008313}{0.029976 - 0.0008313} = 0.05726 \\ u_1 &= u_f + x_1 u_{fg} = 87.26 + (0.05726)(156.87) = 96.24\,\text{kJ/kg} \end{aligned} \quad \text{(Table A-11)}$$

The enthalpy of saturated vapor refrigerant leaving the bottle is

$$h_e = h_{g@26°\text{C}} = 264.68\,\text{kJ/kg}$$

The specific volume at the final state is

$$v_2 = \frac{V}{m_2} = \frac{0.001\,\text{m}^3}{0.1\,\text{kg}} = 0.01\,\text{m}^3/\text{kg}$$

The internal energy at the final state is

$$\left.\begin{aligned} T_2 &= 26°\text{C} \\ v_2 &= 0.01\,\text{m}^3/\text{kg} \end{aligned}\right\} \begin{aligned} x_2 &= \frac{v_2 - v_f}{v_{fg}} = \frac{0.01 - 0.0008313}{0.029976 - 0.0008313} = 0.3146 \\ u_2 &= u_f + x_1 u_{fg} = 87.26 + (0.3146)(156.87) = 136.61\,\text{kJ/kg} \end{aligned} \quad \text{(Table A-11)}$$

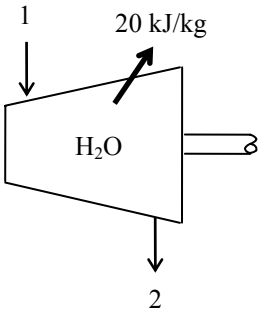
Substituting into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e \\ &= (0.1\,\text{kg})(136.61\,\text{kJ/kg}) - (0.4\,\text{kg})(96.24\,\text{kJ/kg}) + (0.4 - 0.1\,\text{kg})(264.68\,\text{kJ/kg}) \\ &= \mathbf{54.6\,kJ} \end{aligned}$$

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5-187 Steam expands in a turbine steadily. The mass flow rate of the steam, the exit velocity, and the power output are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible.



Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 7\ \text{MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} \nu_1 = 0.05567\ \text{m}^3/\text{kg} \\ h_1 = 3650.6\ \text{kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 25\ \text{kPa} \\ x_2 = 0.95 \end{array}\right\} \begin{array}{l} \nu_2 = \nu_f + x_2\nu_{fg} = 0.00102 + (0.95)(6.2034 - 0.00102) = 5.8933\ \text{m}^3/\text{kg} \\ h_2 = h_f + x_2 h_{fg} = 271.96 + (0.95)(2345.5) = 2500.2\ \text{kJ/kg} \end{array}$$

Analysis (*a*) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\nu_1} V_1 A_1 = \frac{1}{0.05567\ \text{m}^3/\text{kg}} (60\ \text{m/s})(0.015\ \text{m}^2) = \mathbf{16.17\ kg/s}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity is determined from

$$\dot{m} = \frac{1}{\nu_2} V_2 A_2 \longrightarrow V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{(16.17\ \text{kg/s})(5.8933\ \text{m}^3/\text{kg})}{0.14\ \text{m}^2} = \mathbf{680.6\ m/s}$$

(*c*) We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \nearrow 0\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{Q}_{\text{out}} - \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Then the power output of the turbine is determined by substituting to be

$$\dot{W}_{\text{out}} = -(16.17 \times 20)\ \text{kJ/s} - (16.17\ \text{kg/s})\left(2500.2 - 3650.6 + \frac{(680.6\ \text{m/s})^2 - (60\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right)$$

$$= \mathbf{14{,}560\ kW}$$

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5-188 Problem 5-187 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 cm^2 to 3000 cm^2 is to be investigated. The exit velocity and the power output are to be plotted against the exit pressure for the exit areas of 1000, 2000, and 3000 cm^2.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

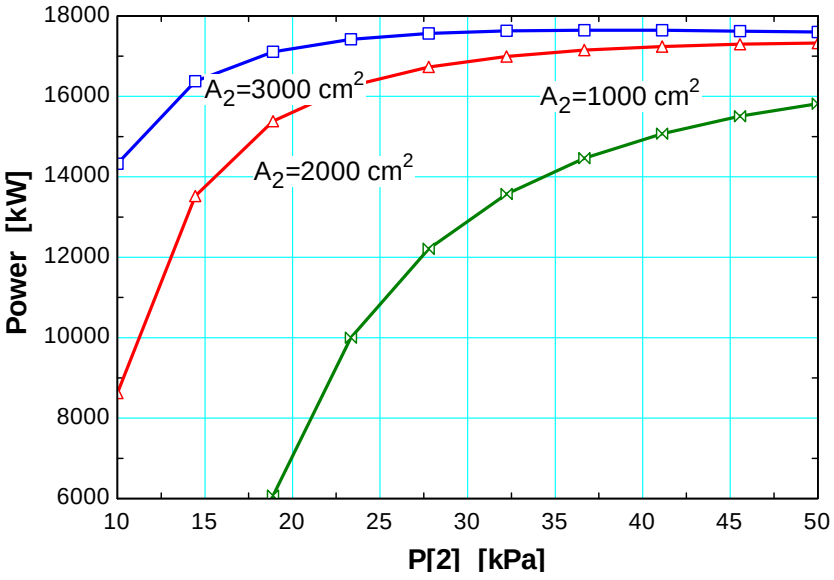
```
Fluid$='Steam_IAPWS'
A[1]=150 [cm^2]
T[1]=600 [C]
P[1]=7000 [kPa]
Vel[1]= 60 [m/s]
A[2]=1400 [cm^2]
P[2]=25 [kPa]
q_out = 20 [kJ/kg]
m_dot = A[1]*Vel[1]/v[1]*convert(cm^2,m^2)
v[1]=volume(Fluid$, T=T[1], P=P[1]) "specific volume of steam at state 1"
Vel[2]=m_dot*v[2]/(A[2]*convert(cm^2,m^2))
v[2]=volume(Fluid$, x=0.95, P=P[2]) "specific volume of steam at state 2"
T[2]=temperature(Fluid$, P=P[2], v=v[2]) "[C]" "not required, but good to know"

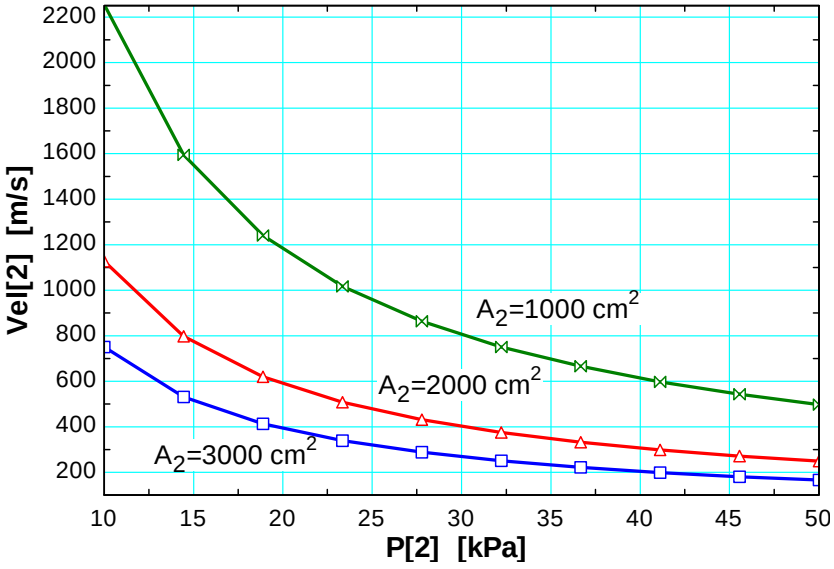
"[conservation of Energy for steady-flow:"
"Ein_dot - Eout_dot = DeltaE_dot"  "For steady-flow, DeltaE_dot = 0"
DELTAE_dot=0
"For the turbine as the control volume, neglecting the PE of each flow steam:"
E_dot_in=E_dot_out
h[1]=enthalpy(Fluid$,T=T[1], P=P[1])
E_dot_in=m_dot*(h[1]+ Vel[1]^2/2*Convert(m^2/s^2, kJ/kg))
h[2]=enthalpy(Fluid$,x=0.95, P=P[2])
E_dot_out=m_dot*(h[2]+ Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+ m_dot *q_out+ W_dot_out
Power=W_dot_out
Q_dot_out=m_dot*q_out
```

P_2 [kPa]	Power [kW]	Vel_2 [m/s]
10	-22158	2253
14.44	-1895	1595
18.89	6071	1239
23.33	9998	1017
27.78	12212	863.2
32.22	13573	751.1
36.67	14464	665.4
41.11	15075	597.8
45.56	15507	543
50	15821	497.7

Table values are for A[2]=1000 [cm^2]

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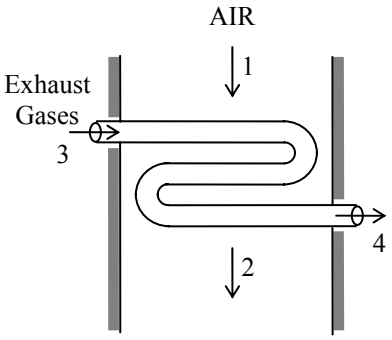
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5-189 Air is preheated by the exhaust gases of a gas turbine in a regenerator. For a specified heat transfer rate, the exit temperature of air and the mass flow rate of exhaust gases are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the regenerator to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Exhaust gases can be treated as air. **6** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpies of air are (Table A-17)

$$T_1 = 550\ \text{K} \ \rightarrow\ h_1 = 555.74\ \text{kJ/kg}$$
$$T_3 = 800\ \text{K} \ \rightarrow\ h_3 = 821.95\ \text{kJ/kg}$$
$$T_4 = 600\ \text{K} \ \rightarrow\ h_4 = 607.02\ \text{kJ/kg}$$



Analysis (*a*) We take the *air side* of the heat exchanger as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{Q}_{\text{in}} + \dot{m}_{\text{air}} h_1 = \dot{m}_{\text{air}} h_2 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_2 - h_1)$$

Substituting,

$$3200\ \text{kJ/s} = (800/60\ \text{kg/s})(h_2 - 554.71\ \text{kJ/kg}) \ \rightarrow\ h_2 = 794.71\ \text{kJ/kg}$$

Then from Table A-17 we read

$$T_2 = \mathbf{775.1\ K}$$

(*b*) Treating the exhaust gases as an ideal gas, the mass flow rate of the exhaust gases is determined from the steady-flow energy relation applied only to the exhaust gases,

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_{\text{exhaust}} h_3 = \dot{Q}_{\text{out}} + \dot{m}_{\text{exhaust}} h_4 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{Q}_{\text{out}} = \dot{m}_{\text{exhaust}} (h_3 - h_4)$$
$$3200\ \text{kJ/s} = \dot{m}_{\text{exhaust}} (821.95 - 607.02)\ \text{kJ/kg}$$

It yields

$$\dot{m}_{\text{exhaust}} = \mathbf{14.9\ kg/s}$$

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5-190 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** The pipe is insulated and thus the heat losses are negligible.

Properties The density and specific heat of water at room temperature are $\rho = 1000$ kg/m^3 and $c = 4.18$ kJ/kg·°C (Table A-3).

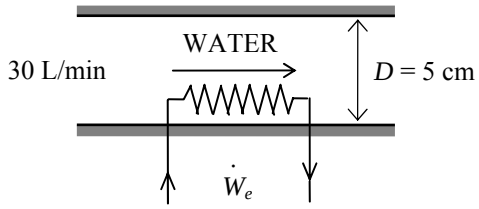
Analysis (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. Also, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{out} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + \nu\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$



The mass flow rate of water through the pipe is

$$\dot{m} = \rho \dot{V}_1 = (1000 \text{ kg/m}^3)(0.030 \text{ m}^3/\text{min}) = 30 \text{ kg/min}$$

Therefore,

$$\dot{W}_{e,in} = \dot{m}c(T_2 - T_1) = (30/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(55 - 20)^\circ\text{C} = \mathbf{73.2\ kW}$$

(*b*) The average velocity of water through the pipe is determined from

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\pi r^2} = \frac{0.030 \text{ m}^3/\text{min}}{\pi(0.025 \text{ m})^2} = \mathbf{15.3\ m/min}$$

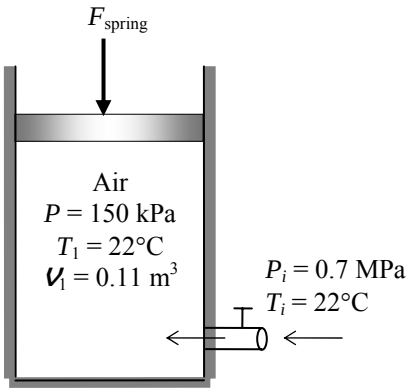
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5-191 An insulated cylinder equipped with an external spring initially contains air. The tank is connected to a supply line, and air is allowed to enter the cylinder until its volume doubles. The mass of the air that entered and the final temperature in the cylinder are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **4** The spring is a linear spring. **5** The device is insulated and thus heat transfer is negligible. **6** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-1). The specific heats of air at room temperature are $c_v = 0.718$ and $c_p = 1.005$ kJ/kg·K (Table A-2a). Also, $u = c_v T$ and $h = c_p T$.

Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as



Mass balance:

$$m_{in} - m_{out} = \Delta m_{system} \quad \rightarrow \quad m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$m_i h_i = W_{b,out} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

Combining the two relations, $(m_2 - m_1)h_i = W_{b,out} + m_2 u_2 - m_1 u_1$

or, $(m_2 - m_1)c_p T_i = W_{b,out} + m_2 c_v T_2 - m_1 c_v T_1$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 V_1}{R T_1} = \frac{(150 \text{ kPa})(0.11 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(295 \text{ K})} = 0.1949 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{R T_2} = \frac{(600 \text{ kPa})(0.22 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}) T_2} = \frac{459.9}{T_2}$$

Then from the mass balance becomes $m_i = m_2 - m_1 = \dfrac{459.9}{T_2} - 0.1949$

The spring is a linear spring, and thus the boundary work for this process can be determined from

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(150 + 600)\text{kPa}}{2}(0.22 - 0.11)\text{m}^3 = 41.25 \text{ kJ}$$

Substituting into the energy balance, the final temperature of air T_2 is determined to be

$$-41.25 = -\left(\frac{459.9}{T_2} - 0.1949\right)(1.005)(295) + \left(\frac{459.9}{T_2}\right)(0.718)(T_2) - (0.1949)(0.718)(295)$$

It yields $T_2 =$ **351 K**

Thus, $m_2 = \dfrac{459.9}{T_2} = \dfrac{459.9}{351.4} = 1.309 \text{ kg}$

and $m_i = m_2 - m_1 = 1.309 - 0.1949 =$ **1.11 kg**

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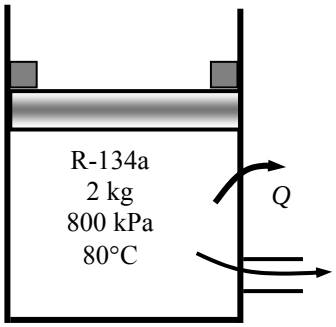
5-192 R-134a is allowed to leave a piston-cylinder device with a pair of stops. The work done and the heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device is assumed to be constant. **2** Kinetic and potential energies are negligible.

Properties The properties of R-134a at various states are (Tables A-11 through A-13)

$$\left.\begin{matrix} P_1 = 800\ \text{kPa} \\ T_1 = 80°\text{C} \end{matrix}\right\} \begin{matrix} v_1 = 0.032659\ \text{m}^3/\text{kg} \\ u_1 = 290.84\ \text{kJ/kg} \\ h_1 = 316.97\ \text{kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 500\ \text{kPa} \\ T_2 = 20°\text{C} \end{matrix}\right\} \begin{matrix} v_2 = 0.042115\ \text{m}^3/\text{kg} \\ u_2 = 242.40\ \text{kJ/kg} \\ h_2 = 263.46\ \text{kJ/kg} \end{matrix}$$



Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\begin{matrix}\text{Net energy transfer} \\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta E_{\text{system}}}_{\begin{matrix}\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}\end{matrix}}$$

$$W_{\text{b,in}} - Q_{\text{out}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since ke} \cong \text{pe} \cong 0)$$

The volumes at the initial and final states and the mass that has left the cylinder are

$$V_1 = m_1 v_1 = (2\ \text{kg})(0.032659\ \text{m}^3/\text{kg}) = 0.06532\ \text{m}^3$$

$$V_2 = m_2 v_2 = (1/2) m_1 v_2 = (1/2)(2\ \text{kg})(0.042115\ \text{m}^3/\text{kg}) = 0.04212\ \text{m}^3$$

$$m_e = m_1 - m_2 = 2 - 1 = 1\ \text{kg}$$

The enthalpy of the refrigerant withdrawn from the cylinder is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder

$$h_e = (1/2)(h_1 + h_2) = (1/2)(316.97 + 263.46) = 290.21\ \text{kJ/kg}$$

Noting that the pressure remains constant after the piston starts moving, the boundary work is determined from

$$W_{\text{b,in}} = P_2 (V_1 - V_2) = (500\ \text{kPa})(0.06532 - 0.04212)\text{m}^3 = \mathbf{11.6\ kJ}$$

(*b*) Substituting,

$$11.6\ \text{kJ} - Q_{\text{out}} - (1\ \text{kg})(290.21\ \text{kJ/kg}) = (1\ \text{kg})(242.40\ \text{kJ/kg}) - (2\ \text{kg})(290.84\ \text{kJ/kg})$$

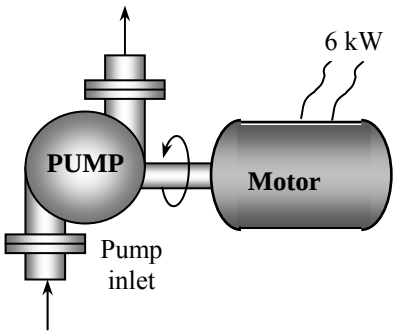
$$Q_{\text{out}} = \mathbf{60.7\ kJ}$$

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5-193 The pressures across a pump are measured. The mechanical efficiency of the pump and the temperature rise of water are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The pump is driven by an external motor so that the heat generated by the motor is dissipated to the atmosphere. **3** The elevation difference between the inlet and outlet of the pump is negligible, $z_1 = z_2$. **4** The inlet and outlet diameters are the same and thus the inlet and exit velocities are equal, $V_1 = V_2$.

Properties We take the density of water to be 1 kg/L = 1000 kg/m^3 and its specific heat to be 4.18 kJ/kg · °C (Table A-3).



Analysis (*a*) The mass flow rate of water through the pump is

$$\dot{m} = \rho \dot{V} = (1\,\text{kg/L})(18\,\text{L/s}) = 18\,\text{kg/s}$$

The motor draws 6 kW of power and is 95 percent efficient. Thus the mechanical (shaft) power it delivers to the pump is

$$\dot{W}_{\text{pump,shaft}} = \eta_{\text{motor}} \dot{W}_{\text{electric}} = (0.95)(6\,\text{kW}) = 5.7\,\text{kW}$$

To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) - \dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right)$$

Simplifying it for this case and substituting the given values,

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}\left(\frac{P_2 - P_1}{\rho}\right) = (18\,\text{kg/s})\left(\frac{(300-100)\text{kPa}}{1000\,\text{kg/m}^3}\right)\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) = 3.6\,\text{kW}$$

Then the mechanical efficiency of the pump becomes

$$\eta_{pump} = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump,shaft}}} = \frac{3.6\,\text{kW}}{5.7\,\text{kW}} = 0.632 = \mathbf{63.2\%}$$

(*b*) Of the 5.7-kW mechanical power supplied by the pump, only 3.6 kW is imparted to the fluid as mechanical energy. The remaining 2.1 kW is converted to thermal energy due to frictional effects, and this "lost" mechanical energy manifests itself as a heating effect in the fluid,

$$\dot{E}_{\text{mech,loss}} = \dot{W}_{\text{pump,shaft}} - \Delta \dot{E}_{\text{mech,fluid}} = 5.7 - 3.6 = 2.1\,\text{kW}$$

The temperature rise of water due to this mechanical inefficiency is determined from the thermal energy balance,

$$\dot{E}_{\text{mech,loss}} = \dot{m}(u_2 - u_1) = \dot{m}c\Delta T$$

Solving for ΔT,

$$\Delta T = \frac{\dot{E}_{\text{mech,loss}}}{\dot{m}c} = \frac{2.1\,\text{kW}}{(18\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot\text{K})} = \mathbf{0.028°C}$$

Therefore, the water will experience a temperature rise of 0.028°C, which is very small, as it flows through the pump.

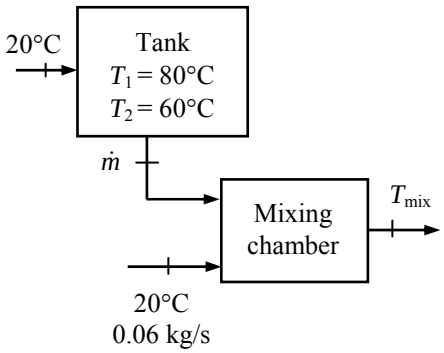
Discussion In an actual application, the temperature rise of water will probably be less since part of the heat generated will be transferred to the casing of the pump and from the casing to the surrounding air. If the entire pump motor were submerged in water, then the 2.1 kW dissipated to the air due to motor inefficiency would also be transferred to the surrounding water as heat. This would cause the water temperature to rise more.

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5-194 A water tank is heated by electricity. The water withdrawn from the tank is mixed with cold water in a chamber. The mass flow rate of hot water withdrawn from the tank and the average temperature of mixed water are to be determined.

Assumptions **1** The process in the mixing chamber is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties The specific heat and density of water are taken to be $c_p = 4.18$ kJ/kg.K, $\rho = 1000$ kg/m^3 (Table A-3).



Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$

or
$$\dot{m}_{hot} c_p T_{tank,ave} + \dot{m}_{cold} c_p T_{cold} = (\dot{m}_{hot} + \dot{m}_{cold}) c_p T_{mixture} \qquad (1)$$

Similarly, an energy balance may be written on the water tank as

$$\left[\dot{W}_{e,in} + \dot{m}_{hot} c_p (T_{cold} - T_{tank,ave})\right]\Delta t = m_{tank} c_p (T_{tank,2} - T_{tank,1}) \qquad (2)$$

where

$$T_{tank,ave} = \frac{T_{tank,1} + T_{tank,2}}{2} = \frac{80 + 60}{2} = 70°C$$

and

$$m_{tank} = \rho V = (1000 \text{ kg/m}^3)(0.060 \text{ m}^3) = 60 \text{ kg}$$

Substituting into Eq. (2),

$$[1.6 \text{ kJ/s} + \dot{m}_{hot}(4.18 \text{ kJ/kg.°C})(20 - 70)°C](8 \times 60 \text{ s}) = (60 \text{ kg})(4.18 \text{ kJ/kg.°C})(60 - 80)°C$$

$$\longrightarrow \dot{m}_{hot} = \mathbf{0.0577 \text{ kg/s}}$$

Substituting into Eq. (1),

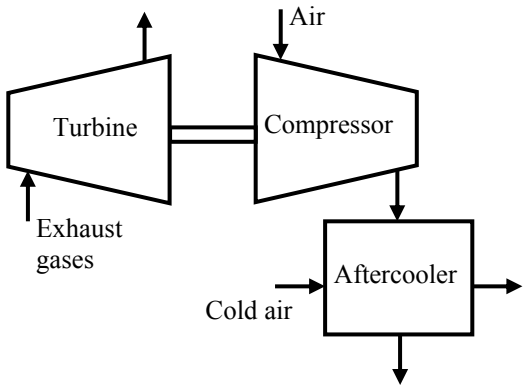
$$(0.0577 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70°C) + (0.06 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(20°C)$$
$$= [(0.0577 + 0.06)\text{kg/s}](4.18 \text{ kJ/kg.°C}) T_{mixture}$$

$$\longrightarrow T_{mixture} = \mathbf{44.5°C}$$

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5-195 The turbocharger of an internal combustion engine consisting of a turbine, a compressor, and an aftercooler is considered. The temperature of the air at the compressor outlet and the minimum flow rate of ambient air are to be determined.

Assumptions **1** All processes are steady since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air properties are used for exhaust gases. **4** Air is an ideal gas with constant specific heats. **5** The mechanical efficiency between the turbine and the compressor is 100%. **6** All devices are adiabatic. **7** The local atmospheric pressure is 100 kPa.



Properties The constant pressure specific heats of exhaust gases, warm air, and cold ambient air are taken to be c_p = 1.063, 1.008, and 1.005 kJ/kg·K, respectively (Table A-2b).

Analysis (*a*) An energy balance on turbine gives

$$\dot{W}_{\rm T} = \dot{m}_{\rm exh}\, c_{p,\rm exh}\left(T_{\rm exh,1} - T_{\rm exh,2}\right) = (0.02\ \text{kg/s})(1.063\ \text{kJ/kg}\cdot\text{K})(400-350)\text{K} = 1.063\ \text{kW}$$

This is also the power input to the compressor since the mechanical efficiency between the turbine and the compressor is assumed to be 100%. An energy balance on the compressor gives the air temperature at the compressor outlet

$$\dot{W}_{\rm C} = \dot{m}_{\rm a}\, c_{p,\rm a}\,(T_{\rm a,2} - T_{\rm a,1})$$

$$1.063\ \text{kW} = (0.018\ \text{kg/s})(1.008\ \text{kJ/kg}\cdot\text{K})(T_{\rm a,2} - 50)\text{K} \longrightarrow T_{\rm a,2} = \mathbf{108.6\ °C}$$

(*b*) An energy balance on the aftercooler gives the mass flow rate of cold ambient air

$$\dot{m}_{\rm a} c_{p,\rm a}(T_{\rm a,2} - T_{\rm a,3}) = \dot{m}_{\rm ca} c_{p,\rm ca}(T_{\rm ca,2} - T_{\rm ca,1})$$

$$(0.018\ \text{kg/s})(1.008\ \text{kJ/kg}\cdot\text{°C})(108.6-80)\text{°C} = \dot{m}_{\rm ca}(1.005\ \text{kJ/kg}\cdot\text{°C})(40-30)\text{°C}$$

$$\dot{m}_{\rm ca} = 0.05161\ \text{kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of aftercooler. That is,

$$\nu_{\rm ca} = \frac{RT}{P} = \frac{(0.287\ \text{kJ/kg}\cdot\text{K})(30+273\ \text{K})}{100\ \text{kPa}} = 0.8696\ \text{m}^3/\text{kg}$$

$$\dot{V}_{\rm ca} = \dot{m}\nu_{\rm ca} = (0.05161\ \text{kg/s})(0.8696\ \text{m}^3/\text{kg}) = \mathbf{0.0449\ m^3/s} = \mathbf{44.9\ L/s}$$

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5-196 Heat is transferred to a pressure cooker at a specified rate for a specified time period. The cooking temperature and the water remaining in the cooker are to be determined.

Assumptions **1** This process can be analyzed as a *uniform-flow process* since the properties of the steam leaving the control volume remain constant during the entire cooking process. **2** The kinetic and potential energies of the streams

are negligible, ke ≅ pe ≅ 0. **3** The pressure cooker is stationary and thus its kinetic and potential energy changes are zero; that is, $\Delta KE = \Delta PE = 0$ and $\Delta E_{system} = \Delta U_{system}$. **4** The pressure (and thus temperature) in the pressure

cooker remains constant. **5** Steam leaves as a saturated vapor at the cooker pressure. **6** There are no boundary, electrical, or shaft work interactions involved. **7** Heat is transferred to the cooker at a constant rate.

Analysis We take the *pressure cooker* as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that this is an unsteady-flow process since changes occur within the control volume. Also, there is one exit and no inlets for mass flow.

(*a*) The absolute pressure within the cooker is

$$P_{abs} = P_{gage} + P_{atm} = 75 + 100 = 175 \text{ kPa}$$

Since saturation conditions exist in the cooker at all times, the cooking temperature must be the saturation temperature corresponding to this pressure. From Table A–5, it is

$$T_2 = T_{sat\ @\ 175\ kPa} = \mathbf{116.04°C}$$

which is about 16°C higher than the ordinary cooking temperature.

(*b*) Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

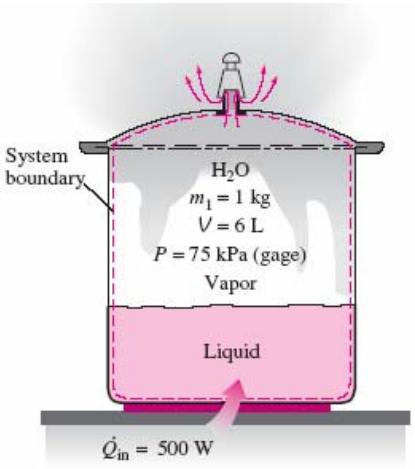
Mass balance:

$$m_{in} - m_{out} = \Delta m_{system} \quad \rightarrow \quad -m_e = (m_2 - m_1)_{CV} \quad \text{or} \quad m_e = (m_1 - m_2)_{CV}$$

Energy balance:

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong ke \cong pe \cong 0)$$



Combining the mass and energy balances gives

$$Q_{in} = (m_1 - m_2)h_e + m_2 u_2 - m_1 u_1$$

The amount of heat transfer during this process is found from

$$Q_{in} = \dot{Q}_{in} \Delta T = (0.5 \text{ kJ/s})(30 \times 60 \text{ s}) = 900 \text{ kJ}$$

Steam leaves the pressure cooker as saturated vapor at 175 kPa at all times. Thus,

$$h_e = h_{g@175\ kPa} = 2700.2 \text{ kJ/kg}$$

The initial internal energy is found after the quality is determined:

$$v_1 = \frac{V}{m_1} = \frac{0.006 \text{ m}^3}{1 \text{ kg}} = 0.006 \text{ m}^3/\text{kg}$$

$$x_1 = \frac{v_1 - v_f}{v_{fg}} = \frac{0.006 - 0.001}{1.004 - 0.001} = 0.00499$$

Thus,

$$u_1 = u_f + x_1 u_{fg} = 486.82 + (0.00499)(2037.7) \text{ kJ/kg} = 497 \text{ kJ/kg}$$

and

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$$U_1 = m_1 u_1 = (1\,\text{kg})(497\,\text{kJ/kg}) = 497\,\text{kJ}$$

The mass of the system at the final state is $m_2 = V/v_2$. Substituting this into the energy equation yields

$$Q_{\text{in}} = \left(m_1 - \frac{V}{v_2} \right) h_e + \left(\frac{V}{v_2} u_2 - m_1 u_1 \right)$$

There are two unknowns in this equation, u_2 and v_2. Thus we need to relate them to a single unknown before we can determine these unknowns. Assuming there is still some liquid water left in the cooker at the final state (i.e., saturation conditions exist), v_2 and u_2 can be expressed as

$$v_2 = v_f + x_2 v_{fg} = 0.001 + x_2 (1.004 - 0.001)\,\text{m}^3/\text{kg}$$
$$u_2 = u_f + x_2 u_{fg} = 486.82 + x_2 (2037.7)\,\text{kJ/kg}$$

Recall that during a boiling process at constant pressure, the properties of each phase remain constant (only the amounts change). When these expressions are substituted into the above energy equation, x_2 becomes the only unknown, and it is determined to be

$$x_2 = 0.009$$

Thus,

$$v_2 = 0.001 + (0.009)(1.004 - 0.001)\,\text{m}^3/\text{kg} = 0.010\,\text{m}^3/\text{kg}$$

and

$$m_2 = \frac{V}{v_2} = \frac{0.006\,\text{m}^3}{0.01\,\text{m}^3/\text{kg}} = \mathbf{0.6\,kg}$$

Therefore, after 30 min there is 0.6 kg water (liquid + vapor) left in the pressure cooker.

Discussion Note that almost half of the water in the pressure cooker has evaporated during cooking.

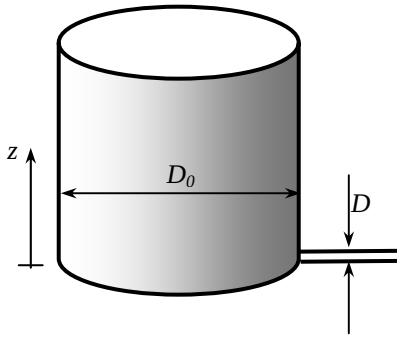
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5-197 A water tank open to the atmosphere is initially filled with water. The tank discharges to the atmosphere through a long pipe connected to a valve. The initial discharge velocity from the tank and the time required to empty the tank are to be determined.

Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The tank is considered to be empty when the water level drops to the center of the valve.

Analysis (*a*) Substituting the known quantities, the discharge velocity can be expressed as

$$V = \sqrt{\frac{2gz}{1.5 + fL/D}} = \sqrt{\frac{2gz}{1.5 + 0.015(100\,\text{m})/(0.10\,\text{m})}} = \sqrt{0.1212gz}$$



Then the initial discharge velocity becomes

$$V_1 = \sqrt{0.1212gz_1} = \sqrt{0.1212(9.81\,\text{m/s}^2)(2\,\text{m})} = \mathbf{1.54\ m/s}$$

where *z* is the water height relative to the center of the orifice at that time.

(*b*) The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{V} = A_{\text{pipe}} V_2 = \frac{\pi D^2}{4}\sqrt{0.1212gz}$$

Then the amount of water that flows through the pipe during a differential time interval *dt* is

$$dV = \dot{V}dt = \frac{\pi D^2}{4}\sqrt{0.1212gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV = A_{\text{tank}}(-dz) = -\frac{\pi D_0^2}{4}dz \qquad (2)$$

where *dz* is the change in the water level in the tank during *dt*. (Note that *dz* is a negative quantity since the positive direction of *z* is upwards. Therefore, we used –*dz* to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{0.1212gz}\,dt = -\frac{\pi D_0^2}{4}dz \;\rightarrow\; dt = -\frac{D_0^2}{D^2}\frac{dz}{\sqrt{0.1212gz}} = -\frac{D_0^2}{D^2\sqrt{0.1212g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2\sqrt{0.1212g}}\int_{z=z_1}^{0} z^{-1/2}dz \;\rightarrow\; t_f = -\frac{D_0^2}{D^2\sqrt{0.1212g}}\left.\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2\sqrt{0.1212g}}z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{2D_0^2}{D^2}\sqrt{\frac{z_1}{0.1212g}} = \frac{2(10\,\text{m})^2}{(0.1\,\text{m})^2}\sqrt{\frac{2\,\text{m}}{0.1212(9.81\,\text{m/s}^2)}} = 25{,}940\,\text{s} = \mathbf{7.21\,h}$$

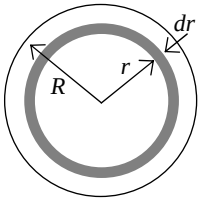
Discussion The draining time can be shortened considerably by installing a pump in the pipe.

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5-198 A fluid is flowing in a circular pipe. A relation is to be obtained for the average fluid velocity in therms of $V(r)$, R, and r.

Analysis Choosing a circular ring of area $dA = 2\pi r dr$ as our differential area, the mass flow rate through a cross-sectional area can be expressed as

$$\dot{m} = \int_A \rho V(r) dA = \int_0^R \rho V(r) 2\pi r\, dr$$



Solving for V_{avg},

$$V_{avg} = \frac{2}{R^2}\int_0^R V(r) r\, dr$$

5-199 Two streams of same ideal gas at different states are mixed in a mixing chamber. The simplest expression for the mixture temperature in a specified format is to be obtained.

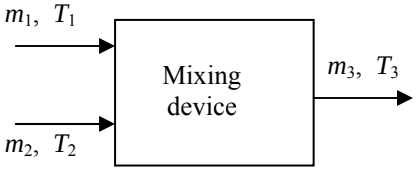
Analysis The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}^{\nearrow 0 \text{ (steady)}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$\dot{m}_1 c_p T_1 + \dot{m}_2 c_p T_2 = \dot{m}_3 c_p T_3$$



and,

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

Solving for final temperature, we find

$$T_3 = \frac{\dot{m}_1}{\dot{m}_3} T_1 + \frac{\dot{m}_2}{\dot{m}_3} T_2$$

5-200 A rigid container filled with an ideal gas is heated while gas is released so that the temperature of the gas remaining in the container stays constant. An expression for the mass flow rate at the outlet as a function of the rate of pressure change in the container is to be derived.

Analysis At any instant, the mass in the control volume may be expressed as

$$m_{CV} = \frac{\boldsymbol{V}}{\boldsymbol{v}} = \frac{\boldsymbol{V}}{RT} P$$

Since there are no inlets to this control volume, the conservation of mass principle becomes

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dm_{CV}}{dt}$$

$$\dot{m}_{out} = -\frac{dm_{CV}}{dt} = -\frac{\boldsymbol{V}}{RT}\frac{dP}{dt}$$

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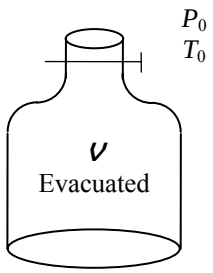
5-201 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

Analysis We take the bottle as the system. It is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = \text{ke} \cong \text{pe} \cong 0)$$

Combining the two balances:

$$Q_{\text{in}} = m_2(u_2 - h_i) = m_2(c_v T_2 - c_p T_i)$$

but

$$T_{\text{i}} = T_2 = T_0$$

and

$$c_p - c_v = R.$$

Substituting,

$$Q_{\text{in}} = m_2\left(c_v - c_p\right)T_0 = -m_2 R T_0 = -\frac{P_0 V}{R T_0} R T_0 = -P_0 V$$

Therefore,

$$Q_{\text{out}} = \boldsymbol{P_0 V} \quad \text{(Heat is lost from the tank)}$$

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5-202 Steam is accelerated by a nozzle steadily from a low velocity to a velocity of 280 m/s at a rate of 2.5 kg/s. If the temperature and pressure of the steam at the nozzle exit are 400°C and 2 MPa, the exit area of the nozzle is

(a) 8.4 cm^2 (b) 10.7 cm^2 (c) 13.5 cm^2 (d) 19.6 cm^2 (e) 23.0 cm^2

Answer (c) 13.5 cm^2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel_1=0 "m/s"
Vel_2=280 "m/s"
m=2.5 "kg/s"
T2=400 "C"
P2=2000 "kPa"
"The rate form of energy balance is E_dot_in - E_dot_out = DELTAE_dot_cv"
v2=VOLUME(Steam_IAPWS,T=T2,P=P2)
m=(1/v2)*A2*Vel_2  "A2 in m^2"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
P2*v2ideal=R*(T2+273)
m=(1/v2ideal)*W1_A2*Vel_2 "assuming ideal gas"
P1*v2ideal=R*T2
m=(1/v2ideal)*W2_A2*Vel_2 "assuming ideal gas and using C"
m=W3_A2*Vel_2 "not using specific volume"
```

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5-203 Steam enters a diffuser steadily at 0.5 MPa, 300°C, and 122 m/s at a rate of 3.5 kg/s. The inlet area of the diffuser is

(a) 15 cm^2 (b) 50 cm^2 (c) 105 cm^2 (d) 150 cm^2 (e) 190 cm^2

Answer (d) 150 cm^2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel_1=122 "m/s"
m=3.5 "kg/s"
T1=300 "C"
P1=500 "kPa"
"The rate form of energy balance is E_dot_in - E_dot_out = DELTAE_dot_cv"
v1=VOLUME(Steam_IAPWS,T=T1,P=P1)
m=(1/v1)*A*Vel_1  "A in m^2"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
P1*v1ideal=R*(T1+273)
m=(1/v1ideal)*W1_A*Vel_1 "assuming ideal gas"
P1*v2ideal=R*T1
m=(1/v2ideal)*W2_A*Vel_1 "assuming ideal gas and using C"
m=W3_A*Vel_1 "not using specific volume"
```

5-204 An adiabatic heat exchanger is used to heat cold water at 15°C entering at a rate of 5 kg/s by hot air at 90°C entering also at rate of 5 kg/s. If the exit temperature of hot air is 20°C, the exit temperature of cold water is

(a) 27°C (b) 32°C (c) 52°C (d) 85°C (e) 90°C

Answer (b) 32°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Cp_air=1.005 "kJ/kg-C"
Tw1=15 "C"
m_dot_w=5 "kg/s"
Tair1=90 "C"
Tair2=20 "C"
m_dot_air=5 "kg/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(Tw2-Tw1)

"Some Wrong Solutions with Common Mistakes:"
(Tair1-Tair2)=(W1_Tw2-Tw1) "Equating temperature changes of fluids"
Cv_air=0.718 "kJ/kg.K"
m_dot_air*Cv_air*(Tair1-Tair2)=m_dot_w*C_w*(W2_Tw2-Tw1) "Using Cv for air"
W3_Tw2=Tair1 "Setting inlet temperature of hot fluid = exit temperature of cold fluid"
W4_Tw2=Tair2 "Setting exit temperature of hot fluid = exit temperature of cold fluid"
```

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5-205 A heat exchanger is used to heat cold water at 15°C entering at a rate of 2 kg/s by hot air at 85°C entering at rate of 3 kg/s. The heat exchanger is not insulated, and is loosing heat at a rate of 25 kJ/s. If the exit temperature of hot air is 20°C, the exit temperature of cold water is

(a) 28°C (b) 35°C (c) 38°C (d) 41°C (e) 80°C

Answer (b) 35°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Cp_air=1.005 "kJ/kg-C"
Tw1=15 "C"
m_dot_w=2 "kg/s"
Tair1=85 "C"
Tair2=20 "C"
m_dot_air=3 "kg/s"
Q_loss=25 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(Tw2-Tw1)+Q_loss

"Some Wrong Solutions with Common Mistakes:"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(W1_Tw2-Tw1) "Not considering Q_loss"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(W2_Tw2-Tw1)-Q_loss  "Taking heat loss as heat gain"
(Tair1-Tair2)=(W3_Tw2-Tw1) "Equating temperature changes of fluids"
Cv_air=0.718 "kJ/kg.K"
m_dot_air*Cv_air*(Tair1-Tair2)=m_dot_w*C_w*(W4_Tw2-Tw1)+Q_loss  "Using Cv for air"
```

5-206 An adiabatic heat exchanger is used to heat cold water at 15°C entering at a rate of 5 kg/s by hot water at 90°C entering at rate of 4 kg/s. If the exit temperature of hot water is 50°C, the exit temperature of cold water is

(a) 42°C (b) 47°C (c) 55°C (d) 78°C (e) 90°C

Answer (b) 47°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Tcold_1=15 "C"
m_dot_cold=5 "kg/s"
Thot_1=90 "C"
Thot_2=50 "C"
m_dot_hot=4 "kg/s"
Q_loss=0 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_w*(Thot_1-Thot_2)=m_dot_cold*C_w*(Tcold_2-Tcold_1)+Q_loss

"Some Wrong Solutions with Common Mistakes:"
Thot_1-Thot_2=W1_Tcold_2-Tcold_1 "Equating temperature changes of fluids"
W2_Tcold_2=90 "Taking exit temp of cold fluid=inlet temp of hot fluid"
```

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5-207 In a shower, cold water at 10°C flowing at a rate of 5 kg/min is mixed with hot water at 60°C flowing at a rate of 2 kg/min. The exit temperature of the mixture will be

(a) 24.3°C (b) 35.0°C (c) 40.0°C (d) 44.3°C (e) 55.2°C

Answer (a) 24.3°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Tcold_1=10 "C"
m_dot_cold=5 "kg/min"
Thot_1=60 "C"
m_dot_hot=2 "kg/min"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_w*Thot_1+m_dot_cold*C_w*Tcold_1=(m_dot_hot+m_dot_cold)*C_w*Tmix
"Some Wrong Solutions with Common Mistakes:"
W1_Tmix=(Tcold_1+Thot_1)/2 "Taking the average temperature of inlet fluids"
```

5-208 In a heating system, cold outdoor air at 7°C flowing at a rate of 4 kg/min is mixed adiabatically with heated air at 70°C flowing at a rate of 3 kg/min. The exit temperature of the mixture is

(a) 34°C (b) 39°C (c) 45°C (d) 63°C (e) 77°C

Answer (a) 34°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_air=1.005 "kJ/kg-C"
Tcold_1=7 "C"
m_dot_cold=4 "kg/min"
Thot_1=70 "C"
m_dot_hot=3 "kg/min"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_air*Thot_1+m_dot_cold*C_air*Tcold_1=(m_dot_hot+m_dot_cold)*C_air*Tmix
"Some Wrong Solutions with Common Mistakes:"
W1_Tmix=(Tcold_1+Thot_1)/2 "Taking the average temperature of inlet fluids"
```

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5-209 Hot combustion gases (assumed to have the properties of air at room temperature) enter a gas turbine at 1 MPa and 1500 K at a rate of 0.1 kg/s, and exit at 0.2 MPa and 900 K. If heat is lost from the turbine to the surroundings at a rate of 15 kJ/s, the power output of the gas turbine is

(a) 15 kW (b) 30 kW (c) 45 kW (d) 60 kW (e) 75 kW

Answer (c) 45 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp_air=1.005 "kJ/kg-C"
T1=1500 "K"
T2=900 "K"
m_dot=0.1 "kg/s"
Q_dot_loss=15 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_out+Q_dot_loss=m_dot*Cp_air*(T1-T2)
"Alternative: Variable specific heats - using EES data"
W_dot_outvariable+Q_dot_loss=m_dot*(ENTHALPY(Air,T=T1)-ENTHALPY(Air,T=T2))

"Some Wrong Solutions with Common Mistakes:"
W1_Wout=m_dot*Cp_air*(T1-T2) "Disregarding heat loss"
W2_Wout-Q_dot_loss=m_dot*Cp_air*(T1-T2) "Assuming heat gain instead of loss"
```

5-210 Steam expands in a turbine from 4 MPa and 500°C to 0.5 MPa and 250°C at a rate of 1350 kg/h. Heat is lost from the turbine at a rate of 25 kJ/s during the process. The power output of the turbine is

(a) 157 kW (b) 207 kW (c) 182 kW (d) 287 kW (e) 246 kW

Answer (a) 157 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=500 "C"
P1=4000 "kPa"
T2=250 "C"
P2=500 "kPa"
m_dot=1350/3600 "kg/s"
Q_dot_loss=25 "kJ/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_out+Q_dot_loss=m_dot*(h1-h2)

"Some Wrong Solutions with Common Mistakes:"
W1_Wout=m_dot*(h1-h2)  "Disregarding heat loss"
W2_Wout-Q_dot_loss=m_dot*(h1-h2)  "Assuming heat gain instead of loss"
u1=INTENERGY(Steam_IAPWS,T=T1,P=P1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
W3_Wout+Q_dot_loss=m_dot*(u1-u2) "Using internal energy instead of enthalpy"
W4_Wout-Q_dot_loss=m_dot*(u1-u2) "Using internal energy and wrong direction for heat"
```

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5-211 Steam is compressed by an adiabatic compressor from 0.2 MPa and 150°C to 0.8 MPa and 350°C at a rate of 1.30 kg/s. The power input to the compressor is

(a) 511 kW (b) 393 kW (c) 302 kW (d) 717 kW (e) 901 kW

Answer (a) 511 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=200 "kPa"
T1=150 "C"
P2=800 "kPa"
T2=350 "C"
m_dot=1.30 "kg/s"
Q_dot_loss=0 "kJ/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_in-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
W1_Win-Q_dot_loss=(h2-h1)/m_dot  "Dividing by mass flow rate instead of multiplying"
W2_Win-Q_dot_loss=h2-h1  "Not considering mass flow rate"
u1=INTENERGY(Steam_IAPWS,T=T1,P=P1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
W3_Win-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
W4_Win-Q_dot_loss=u2-u1 "Using internal energy and ignoring mass flow rate"
```

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5-212 Refrigerant-134a is compressed by a compressor from the saturated vapor state at 0.14 MPa to 0.9 MPa and 60°C at a rate of 0.108 kg/s. The refrigerant is cooled at a rate of 1.10 kJ/s during compression. The power input to the compressor is

(a) 4.94 kW (b) 6.04 kW (c) 7.14 kW (d) 7.50 kW (e) 8.13 kW

Answer (c) 7.14 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=140 "kPa"
x1=1
P2=900 "kPa"
T2=60 "C"
m_dot=0.108 "kg/s"
Q_dot_loss=1.10 "kJ/s"
h1=ENTHALPY(R134a,x=x1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_in-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
W1_Win+Q_dot_loss=m_dot*(h2-h1)   "Wrong direction for heat transfer"
W2_Win =m_dot*(h2-h1)  "Not considering heat loss"
u1=INTENERGY(R134a,x=x1,P=P1)
u2=INTENERGY(R134a,T=T2,P=P2)
W3_Win-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
W4_Win+Q_dot_loss=u2-u1 "Using internal energy and wrong direction for heat transfer"
```

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5-213 Refrigerant-134a expands in an adiabatic turbine from 1.2 MPa and 100°C to 0.18 MPa and 50°C at a rate of 1.25 kg/s. The power output of the turbine is

(a) 46.3 kW (b) 66.4 kW (c) 72.7 kW (d) 89.2 kW (e) 112.0 kW

Answer (a) 46.3 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1200 "kPa"
T1=100 "C"
P2=180 "kPa"
T2=50 "C"
m_dot=1.25 "kg/s"
Q_dot_loss=0 "kJ/s"
h1=ENTHALPY(R134a,T=T1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
-W_dot_out-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
-W1_Wout-Q_dot_loss=(h2-h1)/m_dot  "Dividing by mass flow rate instead of multiplying"
-W2_Wout-Q_dot_loss=h2-h1  "Not considering mass flow rate"
u1=INTENERGY(R134a,T=T1,P=P1)
u2=INTENERGY(R134a,T=T2,P=P2)
-W3_Wout-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
-W4_Wout-Q_dot_loss=u2-u1 "Using internal energy and ignoring mass flow rate"
```

5-214 Refrigerant-134a at 1.4 MPa and 90°C is throttled to a pressure of 0.6 MPa. The temperature of the refrigerant after throttling is

(a) 22°C (b) 56°C (c) 82°C (d) 80°C (e) 90.0°C

Answer (d) 80°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1400 "kPa"
T1=90 "C"
P2=600 "kPa"
h1=ENTHALPY(R134a,T=T1,P=P1)
T2=TEMPERATURE(R134a,h=h1,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming the temperature to remain constant"
W2_T2=TEMPERATURE(R134a,x=0,P=P2) "Taking the temperature to be the saturation temperature at P2"
u1=INTENERGY(R134a,T=T1,P=P1)
W3_T2=TEMPERATURE(R134a,u=u1,P=P2) "Assuming u=constant"
v1=VOLUME(R134a,T=T1,P=P1)
W4_T2=TEMPERATURE(R134a,v=v1,P=P2) "Assuming v=constant"
```

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5-215 Air at 27°C and 5 atm is throttled by a valve to 1 atm. If the valve is adiabatic and the change in kinetic energy is negligible, the exit temperature of air will be

(a) 10°C (b) 15°C (c) 20°C (d) 23°C (e) 27°C

Answer (e) 27°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"The temperature of an ideal gas remains constant during throttling, and thus T2=T1"
T1=27 "C"
P1=5 "atm"
P2=1 "atm"
T2=T1 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1*P1/P2 "Assuming v=constant and using C"
W2_T2=(T1+273)*P1/P2-273 "Assuming v=constant and using K"
W3_T2=T1*P2/P1 "Assuming v=constant and pressures backwards and using C"
W4_T2=(T1+273)*P2/P1 "Assuming v=constant and pressures backwards and using K"
```

5-216 Steam at 1 MPa and 300°C is throttled adiabatically to a pressure of 0.4 MPa. If the change in kinetic energy is negligible, the specific volume of the steam after throttling will be

(a) 0.358 m^3/kg (b) 0.233 m^3/kg (c) 0.375 m^3/kg (d) 0.646 m^3/kg (e) 0.655 m^3/kg

Answer (d) 0.646 m^3/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1000 "kPa"
T1=300 "C"
P2=400 "kPa"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
v2=VOLUME(Steam_IAPWS,h=h1,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_v2=VOLUME(Steam_IAPWS,T=T1,P=P2)  "Assuming the volume to remain constant"
u1=INTENERGY(Steam,T=T1,P=P1)
W2_v2=VOLUME(Steam_IAPWS,u=u1,P=P2) "Assuming u=constant"
W3_v2=VOLUME(Steam_IAPWS,T=T1,P=P2)  "Assuming T=constant"
```

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5-217 Air is to be heated steadily by an 8-kW electric resistance heater as it flows through an insulated duct. If the air enters at 50°C at a rate of 2 kg/s, the exit temperature of air will be

(a) 46.0°C (b) 50.0°C (c) 54.0°C (d) 55.4°C (e) 58.0°C

Answer (c) 54.0°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=1.005 "kJ/kg-C"
T1=50 "C"
 m_dot=2 "kg/s"
W_dot_e=8 "kJ/s"
W_dot_e=m_dot*Cp*(T2-T1)
"Checking using data from EES table"
W_dot_e=m_dot*(ENTHALPY(Air,T=T_2table)-ENTHALPY(Air,T=T1))

"Some Wrong Solutions with Common Mistakes:"
Cv=0.718 "kJ/kg.K"
W_dot_e=Cp*(W1_T2-T1) "Not using mass flow rate"
W_dot_e=m_dot*Cv*(W2_T2-T1) "Using Cv"
W_dot_e=m_dot*Cp*W3_T2 "Ignoring T1"
```

5-218 Saturated water vapor at 40°C is to be condensed as it flows through a tube at a rate of 0.20 kg/s. The condensate leaves the tube as a saturated liquid at 40°C. The rate of heat transfer from the tube is

(a) 34 kJ/s (b) 481 kJ/s (c) 2406 kJ/s (d) 514 kJ/s (e) 548 kJ/s

Answer (b) 481 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=40 "C"
 m_dot=0.20 "kg/s"
h_f=ENTHALPY(Steam_IAPWS,T=T1,x=0)
h_g=ENTHALPY(Steam_IAPWS,T=T1,x=1)
h_fg=h_g-h_f
Q_dot=m_dot*h_fg

"Some Wrong Solutions with Common Mistakes:"
W1_Q=m_dot*h_f  "Using hf"
W2_Q=m_dot*h_g "Using hg"
W3_Q=h_fg "not using mass flow rate"
W4_Q=m_dot*(h_f+h_g) "Adding hf and hg"
```

5-219 … 5-223 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 6
THE SECOND LAW OF THERMODYNAMICS

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The Second Law of Thermodynamics and Thermal Energy Reservoirs

6-1C Transferring 5 kWh of heat to an electric resistance wire and producing 5 kWh of electricity.

6-2C An electric resistance heater which consumes 5 kWh of electricity and supplies 6 kWh of heat to a room.

6-3C Transferring 5 kWh of heat to an electric resistance wire and producing 6 kWh of electricity.

6-4C No. Heat cannot flow from a low-temperature medium to a higher temperature medium.

6-5C A thermal-energy reservoir is a body that can supply or absorb finite quantities of heat isothermally. Some examples are the oceans, the lakes, and the atmosphere.

6-6C Yes. Because the temperature of the oven remains constant no matter how much heat is transferred to the potatoes.

6-7C The surrounding air in the room that houses the TV set.

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Heat Engines and Thermal Efficiency

6-8C No. Such an engine violates the Kelvin-Planck statement of the second law of thermodynamics.

6-9C Heat engines are cyclic devices that receive heat from a source, convert some of it to work, and reject the rest to a sink.

6-10C No. Because 100% of the work can be converted to heat.

6-11C It is expressed as "No heat engine can exchange heat with a single reservoir, and produce an equivalent amount of work".

6-12C (a) No, (b) Yes. According to the second law, no heat engine can have and efficiency of 100%.

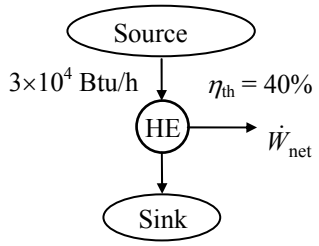
6-13C No. Such an engine violates the Kelvin-Planck statement of the second law of thermodynamics.

6-14C No. The Kelvin-Plank limitation applies only to heat engines; engines that receive heat and convert some of it to work.

6-15C Method (b). With the heating element in the water, heat losses to the surrounding air are minimized, and thus the desired heating can be achieved with less electrical energy input.

6-16E The rate of heat input and thermal efficiency of a heat engine are given. The power output of the heat engine is to be determined.

Assumptions **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.



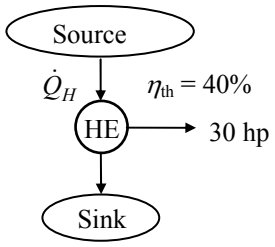
Analysis Applying the definition of the thermal efficiency to the heat engine,

$$\dot{W}_{net} = \eta_{th}\dot{Q}_H$$
$$= (0.4)(3\times10^4 \text{ Btu/h})\left(\frac{1\text{ hp}}{2544.5\text{ Btu/h}}\right)$$
$$= \mathbf{4.72\text{ hp}}$$

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6-17 The power output and thermal efficiency of a heat engine are given. The rate of heat input is to be determined.

Assumptions **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.



Analysis Applying the definition of the thermal efficiency to the heat engine,

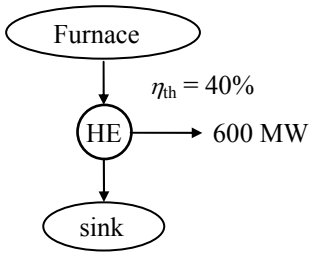
$$\dot{Q}_H = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th}}} = \frac{30\ \text{hp}}{0.4}\left(\frac{0.7457\ \text{kJ/s}}{1\ \text{hp}}\right) = \mathbf{55.9\ kJ/s}$$

6-18 The power output and thermal efficiency of a power plant are given. The rate of heat rejection is to be determined, and the result is to be compared to the actual case in practice.

Assumptions **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.

Analysis The rate of heat supply to the power plant is determined from the thermal efficiency relation,

$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{600\ \text{MW}}{0.4} = 1500\ \text{MW}$$



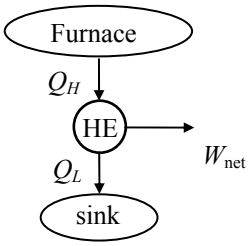
The rate of heat transfer to the river water is determined from the first law relation for a heat engine,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,out}} = 1500 - 600 = \mathbf{900\ \ MW}$$

In reality the amount of heat rejected to the river will be **lower** since part of the heat will be lost to the surrounding air from the working fluid as it passes through the pipes and other components.

6-19 The work output and heat input of a heat engine are given. The heat rejection is to be determined.

Assumptions **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.



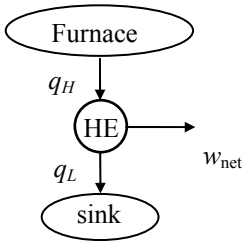
Analysis Applying the first law to the heat engine gives

$$Q_L = Q_H - W_{\text{net}} = 700\ \text{kJ} - 250\ \text{kJ} = \mathbf{450\ kJ}$$

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6-20 The heat rejection and thermal efficiency of a heat engine are given. The heat input to the engine is to be determined.

Assumptions **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.



Analysis According to the definition of the thermal efficiency as applied to the heat engine,

$$w_{net} = \eta_{th} q_H$$
$$q_H - q_L = \eta_{th} q_H$$

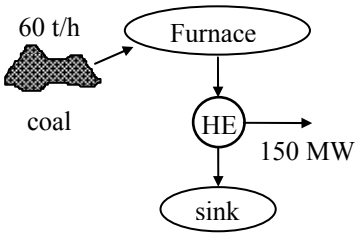
which when rearranged gives

$$q_H = \frac{q_L}{1-\eta_{th}} = \frac{1000 \text{ kJ/kg}}{1-0.4} = \mathbf{1667 \text{ kJ/kg}}$$

6-21 The power output and fuel consumption rate of a power plant are given. The thermal efficiency is to be determined.

Assumptions The plant operates steadily.

Properties The heating value of coal is given to be 30,000 kJ/kg.



Analysis The rate of heat supply to this power plant is

$$\dot{Q}_H = \dot{m}_{coal} q_{HV,coal}$$
$$= (60{,}000 \text{ kg/h})(30{,}000 \text{ kJ/kg}) = 1.8 \times 10^9 \text{ kJ/h}$$
$$= 500 \text{ MW}$$

Then the thermal efficiency of the plant becomes

$$\eta_{th} = \frac{\dot{W}_{net,out}}{\dot{Q}_H} = \frac{150 \text{ MW}}{500 \text{ MW}} = 0.300 = \mathbf{30.0\%}$$

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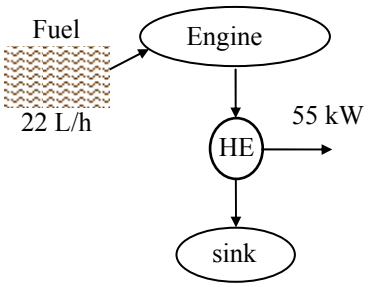
6-22 The power output and fuel consumption rate of a car engine are given. The thermal efficiency of the engine is to be determined.

Assumptions The car operates steadily.

Properties The heating value of the fuel is given to be 44,000 kJ/kg.

Analysis The mass consumption rate of the fuel is

$$\dot{m}_{\text{fuel}} = (\rho \dot{V})_{\text{fuel}} = (0.8 \text{ kg/L})(22 \text{ L/h}) = 17.6 \text{ kg/h}$$



The rate of heat supply to the car is

$$\begin{aligned}\dot{Q}_H &= \dot{m}_{\text{coal}} q_{\text{HV,coal}} \\ &= (17.6 \text{ kg/h})(44{,}000 \text{ kJ/kg}) \\ &= 774{,}400 \text{ kJ/h} = 215.1 \text{ kW}\end{aligned}$$

Then the thermal efficiency of the car becomes

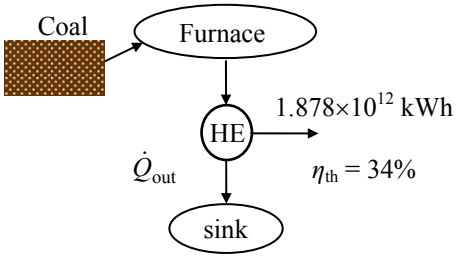
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{55 \text{ kW}}{215.1 \text{ kW}} = 0.256 = \mathbf{25.6\%}$$

6-23 The United States produces about 51 percent of its electricity from coal at a conversion efficiency of about 34 percent. The amount of heat rejected by the coal-fired power plants per year is to be determined.

Analysis Noting that the conversion efficiency is 34%, the amount of heat rejected by the coal plants per year is

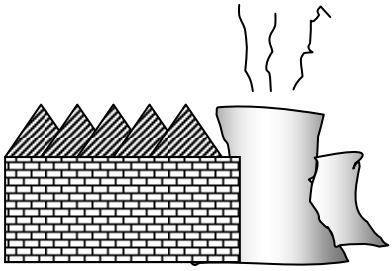
$$\eta_{\text{th}} = \frac{W_{\text{coal}}}{Q_{\text{in}}} = \frac{W_{\text{coal}}}{Q_{\text{out}} + W_{\text{coal}}}$$

$$\begin{aligned}Q_{\text{out}} &= \frac{W_{\text{coal}}}{\eta_{\text{th}}} - W_{\text{coal}} \\ &= \frac{1.878 \times 10^{12} \text{ kWh}}{0.34} - 1.878 \times 10^{12} \text{ kWh} \\ &= \mathbf{3.646 \times 10^{12} \text{ kWh}}\end{aligned}$$



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6-24 The projected power needs of the United States is to be met by building inexpensive but inefficient coal plants or by building expensive but efficient IGCC plants. The price of coal that will enable the IGCC plants to recover their cost difference from fuel savings in 5 years is to be determined.



Assumptions **1** Power is generated continuously by either plant at full capacity. **2** The time value of money (interest, inflation, etc.) is not considered.

Properties The heating value of the coal is given to be 28×10^6 kJ/ton.

Analysis For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are

$$\text{Construction cost}_{\text{coal}} = (150{,}000{,}000\text{ kW})(\$1300/\text{kW}) = \$195\times10^9$$

$$\text{Construction cost}_{\text{IGCC}} = (150{,}000{,}000\text{ kW})(\$1500/\text{kW}) = \$225\times10^9$$

$$\text{Construction cost difference} = \$225\times10^9 - \$195\times10^9 = \$30\times10^9$$

The amount of electricity produced by either plant in 5 years is

$$W_e = \dot{W}\Delta t = (150{,}000{,}000\text{ kW})(5\times365\times24\text{ h}) = 6.570\times10^{12}\text{ kWh}$$

The amount of fuel needed to generate a specified amount of power can be determined from

$$\eta = \frac{W_e}{Q_{\text{in}}} \rightarrow Q_{\text{in}} = \frac{W_e}{\eta} \quad \text{or} \quad m_{\text{fuel}} = \frac{Q_{\text{in}}}{\text{Heating value}} = \frac{W_e}{\eta(\text{Heating value})}$$

Then the amount of coal needed to generate this much electricity by each plant and their difference are

$$m_{\text{coal, coal plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570\times10^{12}\text{ kWh}}{(0.40)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 2.112\times10^9\text{ tons}$$

$$m_{\text{coal, IGCC plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570\times10^{12}\text{ kWh}}{(0.48)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.760\times10^9\text{ tons}$$

$$\Delta m_{\text{coal}} = m_{\text{coal, coal plant}} - m_{\text{coal, IGCC plant}} = 2.112\times10^9 - 1.760\times10^9 = 0.352\times10^9\text{ tons}$$

For Δm_{coal} to pay for the construction cost difference of \$30 billion, the price of coal should be

$$\text{Unit cost of coal} = \frac{\text{Construction cost difference}}{\Delta m_{\text{coal}}} = \frac{\$30\times10^9}{0.352\times10^9\text{ tons}} = \mathbf{\$85.2/ton}$$

Therefore, the IGCC plant becomes attractive when the price of coal is above \$85.2 per ton.

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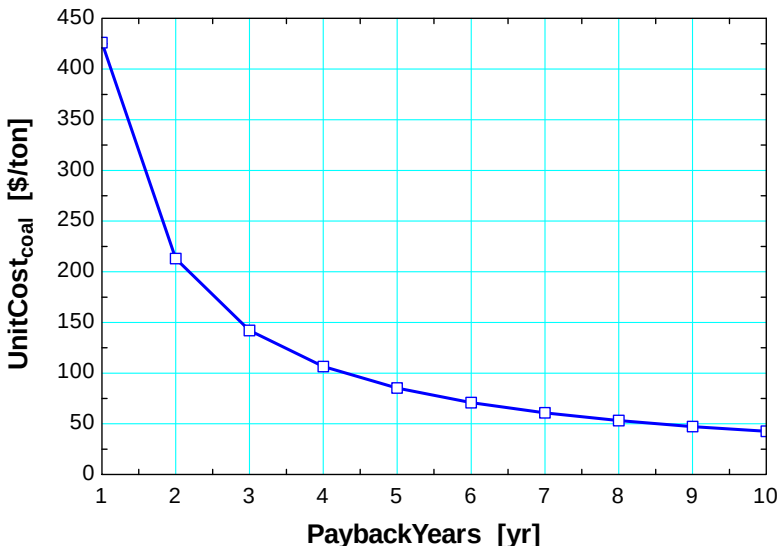
6-25 Problem 6-24 is reconsidered. The price of coal is to be investigated for varying simple payback periods, plant construction costs, and operating efficiency.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
W_dot=15E7 [kW]
Cost_coal=1300 [$/kW]
eta_coal=0.40
Cost_IGCC=1500 [$/kW]
eta_IGCC=0.48
HV_coal=28000 [kJ/kg]
PaybackYears=5 [yr]

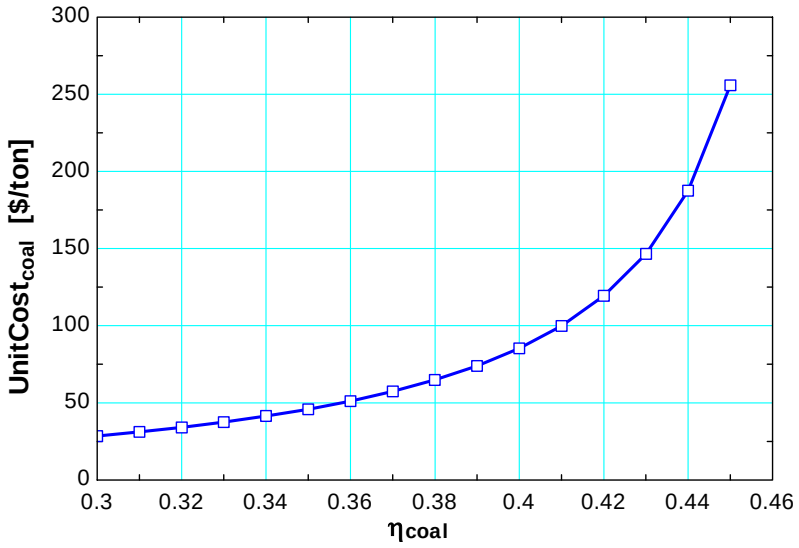
"Analysis"
time=PaybackYears*Convert(yr, h)
ConstCost_coal=W_dot*Cost_coal
ConstCost_IGCC=W_dot*Cost_IGCC
ConstCostDif=ConstCost_IGCC-ConstCost_coal
W_e=W_dot*time
m_coal_coal=W_e/(eta_coal*HV_coal)*Convert(kWh, kJ)
m_coal_IGCC=W_e/(eta_IGCC*HV_coal)*Convert(kWh, kJ)
DELTAm_coal=m_coal_coal-m_coal_IGCC
UnitCost_coal=ConstCostDif/DELTAm_coal*1000
```

PaybackYears [yr]	$UnitCost_{coal}$ [$/ton]
1	426.2
2	213.1
3	142.1
4	106.5
5	85.24
6	71.03
7	60.88
8	53.27
9	47.35
10	42.62

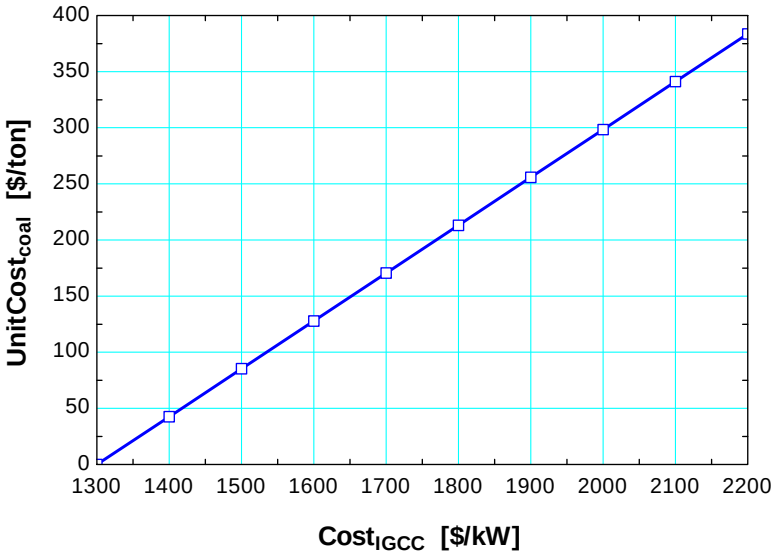


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η_{coal}	$UnitCost_{coal}$ [$/ton]
0.3	28.41
0.31	31.09
0.32	34.09
0.33	37.5
0.34	41.4
0.35	45.9
0.36	51.14
0.37	57.34
0.38	64.78
0.39	73.87
0.4	85.24
0.41	99.85
0.42	119.3
0.43	146.6
0.44	187.5
0.45	255.7



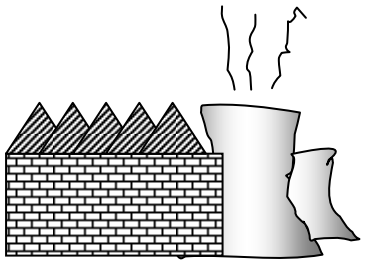
$Cost_{IGCC}$ [$/kW]	$UnitCost_{coal}$ [$/ton]
1300	0
1400	42.62
1500	85.24
1600	127.9
1700	170.5
1800	213.1
1900	255.7
2000	298.3
2100	340.9
2200	383.6



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6-26 The projected power needs of the United States is to be met by building inexpensive but inefficient coal plants or by building expensive but efficient IGCC plants. The price of coal that will enable the IGCC plants to recover their cost difference from fuel savings in 3 years is to be determined.

Assumptions **1** Power is generated continuously by either plant at full capacity. **2** The time value of money (interest, inflation, etc.) is not considered.



Properties The heating value of the coal is given to be 28×10^6 kJ/ton.

Analysis For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are

$$\text{Construction cost}_{\text{coal}} = (150{,}000{,}000\text{ kW})(\$1300/\text{kW}) = \$195\times10^9$$

$$\text{Construction cost}_{\text{IGCC}} = (150{,}000{,}000\text{ kW})(\$1500/\text{kW}) = \$225\times10^9$$

$$\text{Construction cost difference} = \$225\times10^9 - \$195\times10^9 = \$30\times10^9$$

The amount of electricity produced by either plant in 3 years is

$$W_e = \dot{W}\Delta t = (150{,}000{,}000\text{ kW})(3\times365\times24\text{ h}) = 3.942\times10^{12}\text{ kWh}$$

The amount of fuel needed to generate a specified amount of power can be determined from

$$\eta = \frac{W_e}{Q_{\text{in}}} \rightarrow Q_{\text{in}} = \frac{W_e}{\eta} \quad \text{or} \quad m_{\text{fuel}} = \frac{Q_{\text{in}}}{\text{Heating value}} = \frac{W_e}{\eta(\text{Heating value})}$$

Then the amount of coal needed to generate this much electricity by each plant and their difference are

$$m_{\text{coal, coal plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942\times10^{12}\text{ kWh}}{(0.40(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.267\times10^9\text{ tons}$$

$$m_{\text{coal, IGCC plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942\times10^{12}\text{ kWh}}{(0.48)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.055\times10^9\text{ tons}$$

$$\Delta m_{\text{coal}} = m_{\text{coal, coal plant}} - m_{\text{coal, IGCC plant}} = 1.267\times10^9 - 1.055\times10^9 = 0.211\times10^9\text{ tons}$$

For Δm_{coal} to pay for the construction cost difference of \$30 billion, the price of coal should be

$$\text{Unit cost of coal} = \frac{\text{Construction cost difference}}{\Delta m_{\text{coal}}} = \frac{\$30\times10^9}{0.211\times10^9\text{ tons}} = \mathbf{\$142/ton}$$

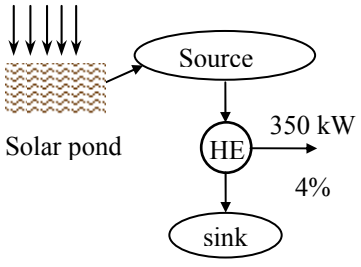
Therefore, the IGCC plant becomes attractive when the price of coal is above \$142 per ton.

6-27E The power output and thermal efficiency of a solar pond power plant are given. The rate of solar energy collection is to be determined.

Assumptions The plant operates steadily.

Analysis The rate of solar energy collection or the rate of heat supply to the power plant is determined from the thermal efficiency relation to be

$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{350\text{ kW}}{0.04}\left(\frac{1\text{ Btu}}{1.055\text{ kJ}}\right)\left(\frac{3600\text{ s}}{1\text{ h}}\right) = \mathbf{2.986\times10^7\,Btu/h}$$



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6-28 A coal-burning power plant produces 300 MW of power. The amount of coal consumed during a one-day period and the rate of air flowing through the furnace are to be determined.

Assumptions **1** The power plant operates steadily. **2** The kinetic and potential energy changes are zero.

Properties The heating value of the coal is given to be 28,000 kJ/kg.

Analysis (*a*) The rate and the amount of heat inputs to the power plant are

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{300\ \text{MW}}{0.32} = 937.5\ \text{MW}$$

$$Q_{\text{in}} = \dot{Q}_{\text{in}}\Delta t = (937.5\ \text{MJ/s})(24 \times 3600\ \text{s}) = 8.1 \times 10^7\ \text{MJ}$$

The amount and rate of coal consumed during this period are

$$m_{\text{coal}} = \frac{Q_{\text{in}}}{q_{\text{HV}}} = \frac{8.1 \times 10^7\ \text{MJ}}{28\ \text{MJ/kg}} = \mathbf{2.893 \times 10^6\ kg}$$

$$\dot{m}_{\text{coal}} = \frac{m_{\text{coal}}}{\Delta t} = \frac{2.893 \times 10^6\ \text{kg}}{24 \times 3600\ \text{s}} = 33.48\ \text{kg/s}$$

(*b*) Noting that the air-fuel ratio is 12, the rate of air flowing through the furnace is

$$\dot{m}_{\text{air}} = (\text{AF})\dot{m}_{\text{coal}} = (12\ \text{kg air/kg fuel})(33.48\ \text{kg/s}) = \mathbf{401.8\ kg/s}$$

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Refrigerators and Heat Pumps

6-29C The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a cold medium whereas the purpose of a heat pump is to supply heat to a warm medium.

6-30C The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a refrigerated space whereas the purpose of an air-conditioner is remove heat from a living space.

6-31C No. Because the refrigerator consumes work to accomplish this task.

6-32C No. Because the heat pump consumes work to accomplish this task.

6-33C The coefficient of performance of a refrigerator represents the amount of heat removed from the refrigerated space for each unit of work supplied. It can be greater than unity.

6-34C The coefficient of performance of a heat pump represents the amount of heat supplied to the heated space for each unit of work supplied. It can be greater than unity.

6-35C No. The heat pump captures energy from a cold medium and carries it to a warm medium. It does not create it.

6-36C No. The refrigerator captures energy from a cold medium and carries it to a warm medium. It does not create it.

6-37C No device can transfer heat from a cold medium to a warm medium without requiring a heat or work input from the surroundings.

6-38C The violation of one statement leads to the violation of the other one, as shown in Sec. 6-4, and thus we conclude that the two statements are equivalent.

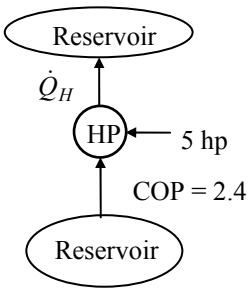
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6-39E The COP and the power input of a residential heat pump are given. The rate of heating effect is to be determined.

Assumptions The heat pump operates steadily.

Analysis Applying the definition of the heat pump coefficient of performance to this heat pump gives

$$\dot{Q}_H = \text{COP}_{\text{HP}}\dot{W}_{\text{net,in}} = (2.4)(5\text{ hp})\left(\frac{2544.5\text{ Btu/h}}{1\text{ hp}}\right) = \mathbf{30{,}530\text{ Btu/h}}$$



6-40 The cooling effect and the rate of heat rejection of an air conditioner are given. The COP is to be determined.

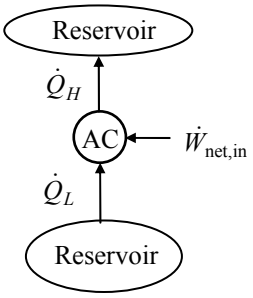
Assumptions The air conditioner operates steadily.

Analysis Applying the first law to the air conditioner gives

$$\dot{W}_{\text{net,in}} = \dot{Q}_H - \dot{Q}_L = 2.5 - 2 = 0.5\text{ kW}$$

Applying the definition of the coefficient of performance,

$$\text{COP}_{\text{R}} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{2.0\text{ kW}}{0.5\text{ kW}} = \mathbf{4}$$

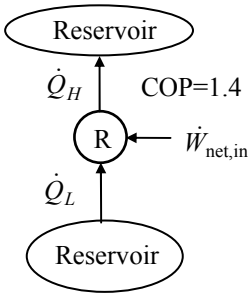


6-41 The power input and the COP of a refrigerator are given. The cooling effect of the refrigerator is to be determined.

Assumptions The refrigerator operates steadily.

Analysis Rearranging the definition of the refrigerator coefficient of performance and applying the result to this refrigerator gives

$$\dot{Q}_L = \text{COP}_{\text{R}}\dot{W}_{\text{net,in}} = (1.4)(3\text{ kW}) = \mathbf{4.2\text{ kW}}$$



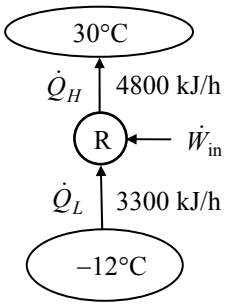
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6-42 A refrigerator is used to keep a food department at a specified temperature. The heat gain to the food department and the heat rejection in the condenser are given. The power input and the COP are to be determined.

Assumptions The refrigerator operates steadily.

Analysis The power input is determined from

$$\begin{aligned}\dot{W}_{\text{in}} &= \dot{Q}_H - \dot{Q}_L \\ &= 4800 - 3300 = 1500 \text{ kJ/h} \\ &= (1500 \text{ kJ/h})\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{0.417 \text{ kW}}\end{aligned}$$



The COP is

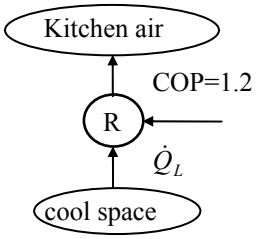
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{3300 \text{ kJ/h}}{1500 \text{ kJ/h}} = \mathbf{2.2}$$

6-43 The COP and the refrigeration rate of a refrigerator are given. The power consumption and the rate of heat rejection are to be determined.

Assumptions The refrigerator operates steadily.

Analysis (*a*) Using the definition of the coefficient of performance, the power input to the refrigerator is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{60 \text{ kJ/min}}{1.2} = 50 \text{ kJ/min} = \mathbf{0.83 \text{ kW}}$$



(*b*) The heat transfer rate to the kitchen air is determined from the energy balance,

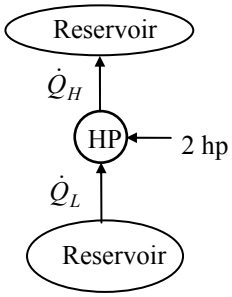
$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{net,in}} = 60 + 50 = \mathbf{110 \text{ kJ/min}}$$

6-44E The heat absorption, the heat rejection, and the power input of a commercial heat pump are given. The COP of the heat pump is to be determined.

Assumptions The heat pump operates steadily.

Analysis Applying the definition of the heat pump coefficient of performance to this heat pump gives

$$\text{COP}_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_{\text{net,in}}} = \frac{15{,}090 \text{ Btu/h}}{2 \text{ hp}}\left(\frac{1 \text{ hp}}{2544.5 \text{ Btu/h}}\right) = \mathbf{2.97}$$



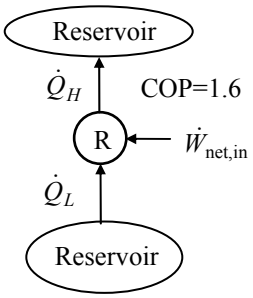
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6-45 The cooling effect and the COP of a refrigerator are given. The power input to the refrigerator is to be determined.

Assumptions The refrigerator operates steadily.

Analysis Rearranging the definition of the refrigerator coefficient of performance and applying the result to this refrigerator gives

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{25{,}000\,\text{kJ/h}}{1.60}\left(\frac{1\,\text{h}}{3600\,\text{s}}\right) = \mathbf{4.34\,kW}$$



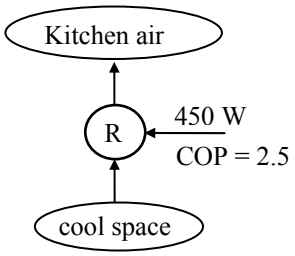
6-46 The COP and the power consumption of a refrigerator are given. The time it will take to cool 5 watermelons is to be determined.

Assumptions **1** The refrigerator operates steadily. **2** The heat gain of the refrigerator through its walls, door, etc. is negligible. **3** The watermelons are the only items in the refrigerator to be cooled.

Properties The specific heat of watermelons is given to be c = 4.2 kJ/kg.°C.

Analysis The total amount of heat that needs to be removed from the watermelons is

$$Q_L = (mc\Delta T)_{\text{watermelons}} = 5 \times (10\ \text{kg})(4.2\ \text{kJ/kg}\cdot{}^\circ\text{C})(20-8)^\circ\text{C} = 2520\ \text{kJ}$$



The rate at which this refrigerator removes heat is

$$\dot{Q}_L = (\text{COP}_\text{R})(\dot{W}_{\text{net,in}}) = (2.5)(0.45\ \text{kW}) = 1.125\ \text{kW}$$

That is, this refrigerator can remove 1.125 kJ of heat per second. Thus the time required to remove 2520 kJ of heat is

$$\Delta t = \frac{Q_L}{\dot{Q}_L} = \frac{2520\ \text{kJ}}{1.125\ \text{kJ/s}} = \mathbf{2240\ s} = \mathbf{37.3\ min}$$

This answer is optimistic since the refrigerated space will gain some heat during this process from the surrounding air, which will increase the work load. Thus, in reality, it will take longer to cool the watermelons.

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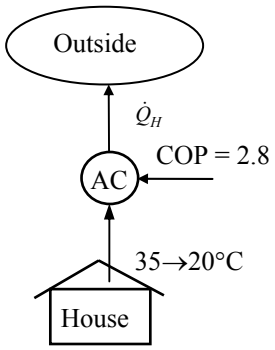
6-47 An air conditioner with a known COP cools a house to desired temperature in 15 min. The power consumption of the air conditioner is to be determined.

Assumptions **1** The air conditioner operates steadily. **2** The house is well-sealed so that no air leaks in or out during cooling. **3** Air is an ideal gas with constant specific heats at room temperature.

Properties The constant volume specific heat of air is given to be $c_v = 0.72$ kJ/kg.°C.

Analysis Since the house is well-sealed (constant volume), the total amount of heat that needs to be removed from the house is

$$Q_L = (mc_v \Delta T)_{\text{House}} = (800\ \text{kg})(0.72\ \text{kJ/kg} \cdot °\text{C})(35 - 20)°\text{C} = 8640\ \text{kJ}$$



This heat is removed in 30 minutes. Thus the average rate of heat removal from the house is

$$\dot{Q}_L = \frac{Q_L}{\Delta t} = \frac{8640\ \text{kJ}}{30 \times 60\ \text{s}} = 4.8\ \text{kW}$$

Using the definition of the coefficient of performance, the power input to the air-conditioner is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{4.8\ \text{kW}}{2.8} = \mathbf{1.71\ kW}$$

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6-48 Problem 6-47 is reconsidered. The rate of power drawn by the air conditioner required to cool the house as a function for air conditioner EER ratings in the range 5 to 15 is to be investigated. Representative costs of air conditioning units in the EER rating range are to be included.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

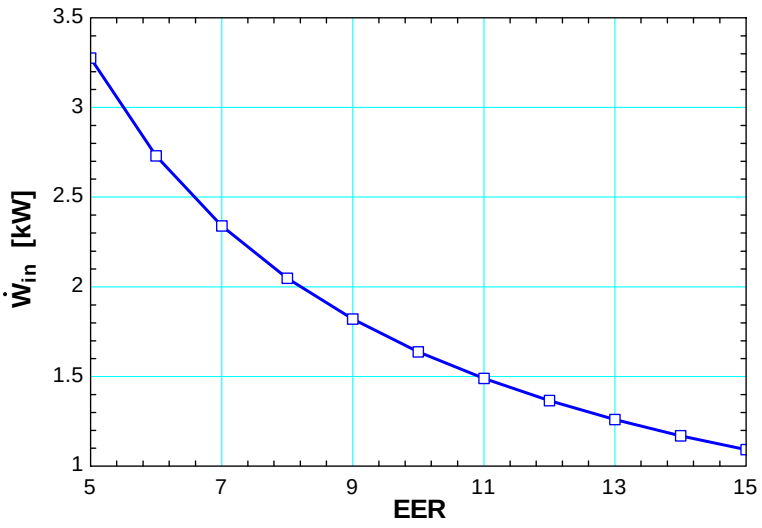
```
"Since it is well sealed, we treat the house as a closed system (constant volume)
to determine the rate of heat transfer required to cool the house.  Apply the first law,
closed system on a rate basis to the house."

"Input Data"
T_1=35 [C]
T_2=20 [C]
c_v = 0.72 [kJ/kg-C]
m_house=800 [kg]
DELTAtime=30 [min]
"EER=5"
COP=EER/3.412

"Assuming no work done on the house and no heat energy added to the house
in the time period with no change in KE and PE, the first law applied to the house is:"
E_dot_in - E_dot_out = DELTAE_dot
E_dot_in = 0
E_dot_out = Q_dot_L
DELTAE_dot = m_house*DELTAu_house/DELTAtime
DELTAu_house = c_v*(T_2-T_1)

"Using the definition of the coefficient of performance of the A/C:"
W_dot_in = Q_dot_L/COP "kJ/min"*convert('kJ/min','kW')  "kW"
Q_dot_H= W_dot_in*convert('KW','kJ/min')  + Q_dot_L  "kJ/min"
```

EER	W_{in} [kW]
5	3.276
6	2.73
7	2.34
8	2.047
9	1.82
10	1.638
11	1.489
12	1.365
13	1.26
14	1.17
15	1.092



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6-49 A refrigerator is used to cool bananas to a specified temperature. The power input is given. The rate of cooling and the COP are to be determined.

Assumptions The refrigerator operates steadily.

Properties The specific heat of banana is 3.35 kJ/kg·°C.

Analysis The rate of cooling is determined from

$$\dot{Q}_L = \dot{m}c_p(T_1 - T_2) = (215/60 \text{ kg/min})(3.35 \text{ kJ/kg} \cdot \text{°C})(24 - 13) \text{°C} = \mathbf{132 \text{ kJ/min}}$$

The COP is

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{(132/60) \text{ kW}}{1.4 \text{ kW}} = \mathbf{1.57}$$

6-50 A refrigerator is used to cool water to a specified temperature. The power input is given. The flow rate of water and the COP of the refrigerator are to be determined.

Assumptions The refrigerator operates steadily.

Properties The specific heat of water is 4.18 kJ/kg·°C and its density is 1 kg/L.

Analysis The rate of cooling is determined from

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = (570/60) \text{ kW} - 2.65 \text{ kW} = 6.85 \text{ kW}$$

The mass flow rate of water is

$$\dot{Q}_L = \dot{m}c_p(T_1 - T_2) \longrightarrow \dot{m} = \frac{\dot{Q}_L}{c_p(T_1 - T_2)} = \frac{6.85 \text{ kW}}{(4.18 \text{ kJ/kg} \cdot \text{°C})(23 - 5) \text{°C}} = 0.09104 \text{ kg/s}$$

The volume flow rate is

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.09104 \text{ kg/s}}{1 \text{ kg/L}} \left(\frac{60 \text{ s}}{1 \text{ min}} \right) = \mathbf{5.46 \text{ L/min}}$$

The COP is

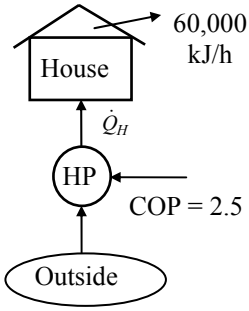
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{6.85 \text{ kW}}{2.65 \text{ kW}} = \mathbf{2.58}$$

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6-51 The rate of heat loss, the rate of internal heat gain, and the COP of a heat pump are given. The power input to the heat pump is to be determined.

Assumptions The heat pump operates steadily.

Analysis The heating load of this heat pump system is the difference between the heat lost to the outdoors and the heat generated in the house from the people, lights, and appliances,



$$\dot{Q}_H = 60{,}000 - 4{,}000 = 56{,}000 \text{ kJ/h}$$

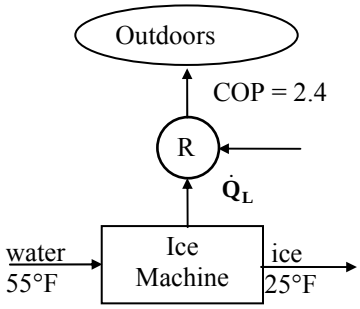
Using the definition of COP, the power input to the heat pump is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{56{,}000 \text{ kJ/h}}{2.5}\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{6.22\ kW}$$

6-52E The COP and the refrigeration rate of an ice machine are given. The power consumption is to be determined.

Assumptions The ice machine operates steadily.

Analysis The cooling load of this ice machine is



$$\dot{Q}_L = \dot{m}q_L = (28 \text{ lbm/h})(169 \text{ Btu/lbm}) = 4732 \text{ Btu/h}$$

Using the definition of the coefficient of performance, the power input to the ice machine system is determined to be

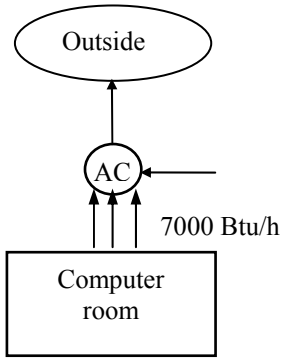
$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R}}} = \frac{4732 \text{ Btu/h}}{2.4}\left(\frac{1 \text{ hp}}{2545 \text{ Btu/h}}\right) = \mathbf{0.775\ hp}$$

6-53E An office that is being cooled adequately by a 12,000 Btu/h window air-conditioner is converted to a computer room. The number of additional air-conditioners that need to be installed is to be determined.

Assumptions **1** The computers are operated by 7 adult men. **2** The computers consume 40 percent of their rated power at any given time.

Properties The average rate of heat generation from a person seated in a room/office is 100 W (given).

Analysis The amount of heat dissipated by the computers is equal to the amount of electrical energy they consume. Therefore,



$$\dot{Q}_{\text{computers}} = (\text{Rated power}) \times (\text{Usage factor}) = (8.4 \text{ kW})(0.4) = 3.36 \text{ kW}$$

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 7 \times (100 \text{ W}) = 700 \text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{computers}} + \dot{Q}_{\text{people}} = 3360 + 700 = 4060 \text{ W} = 13{,}853 \text{ Btu/h}$$

since 1 W = 3.412 Btu/h. Then noting that each available air conditioner provides 7000 Btu/h cooling, the number of air-conditioners needed becomes

$$\text{No. of air conditioners} = \frac{\text{Cooling load}}{\text{Cooling capacity of A/C}} = \frac{13{,}853 \text{ Btu/h}}{7000 \text{ Btu/h}}$$

$$= 1.98 \approx \mathbf{2\ Air\ conditioners}$$

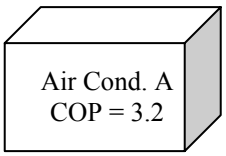
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6-54 A decision is to be made between a cheaper but inefficient air-conditioner and an expensive but efficient air-conditioner for a building. The better buy is to be determined.

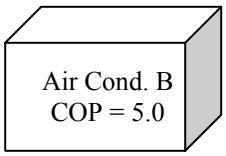
Assumptions The two air conditioners are comparable in all aspects other than the initial cost and the efficiency.

Analysis The unit that will cost less during its lifetime is a better buy. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period. The energy and cost savings of the more efficient air conditioner in this case is

$$\begin{aligned}\text{Energy savings} &= (\text{Annual energy usage of A}) - (\text{Annual energy usage of B})\\ &= (\text{Annual cooling load})(1/\text{COP}_A - 1/\text{COP}_B)\\ &= (120{,}000\ \text{kWh/year})(1/3.2 - 1/5.0)\\ &= 13{,}500\ \text{kWh/year}\end{aligned}$$



$$\begin{aligned}\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy})\\ &= (13{,}500\ \text{kWh/year})(\$0.10/\text{kWh}) = \mathbf{\$1350/year}\end{aligned}$$



The installation cost difference between the two air-conditioners is

$$\text{Cost difference} = \text{Cost of B} - \text{cost of A} = 7000 - 5500 = \$1500$$

Therefore, the more efficient air-conditioner B will pay for the $1500 cost differential in this case in about 1 year.

Discussion A cost conscious consumer will have no difficulty in deciding that the more expensive but more efficient air-conditioner B is clearly the better buy in this case since air conditioners last at least 15 years. But the decision would not be so easy if the unit cost of electricity at that location was much less than $0.10/kWh, or if the annual air-conditioning load of the house was much less than 120,000 kWh.

6-55 A house is heated by resistance heaters, and the amount of electricity consumed during a winter month is given. The amount of money that would be saved if this house were heated by a heat pump with a known COP is to be determined.

Assumptions The heat pump operates steadily.

Analysis The amount of heat the resistance heaters supply to the house is equal to he amount of electricity they consume. Therefore, to achieve the same heating effect, the house must be supplied with 1200 kWh of energy. A heat pump that supplied this much heat will consume electrical power in the amount of

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{1200\ \text{kWh}}{2.4} = 500\ \text{kWh}$$

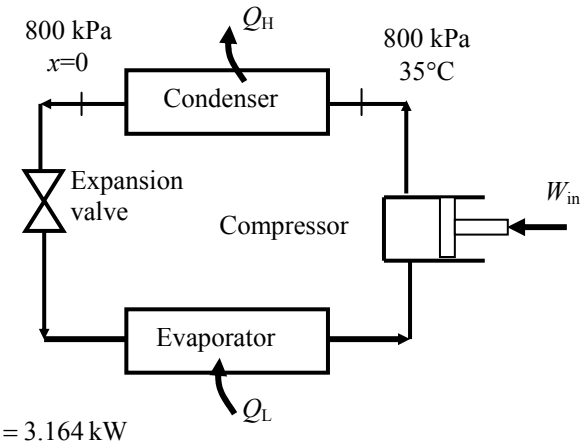
which represent a savings of 1200 – 500 = 700 kWh. Thus the homeowner would have saved

$$(700\ \text{kWh})(0.085\ \$/\text{kWh}) = \mathbf{\$59.50}$$

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6-56 Refrigerant-134a flows through the condenser of a residential heat pump unit. For a given compressor power consumption the COP of the heat pump and the rate of heat absorbed from the outside air are to be determined.

Assumptions **1** The heat pump operates steadily. **2** The kinetic and potential energy changes are zero.



Properties The enthalpies of R-134a at the condenser inlet and exit are

$$\left.\begin{matrix} P_1 = 800\ \text{kPa} \\ T_1 = 35°\text{C} \end{matrix}\right\} h_1 = 271.22\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_2 = 800\ \text{kPa} \\ x_2 = 0 \end{matrix}\right\} h_2 = 95.47\ \text{kJ/kg}$$

Analysis (*a*) An energy balance on the condenser gives the heat rejected in the condenser

$$\dot{Q}_H = \dot{m}(h_1 - h_2) = (0.018\ \text{kg/s})(271.22 - 95.47)\ \text{kJ/kg} = 3.164\ \text{kW}$$

The COP of the heat pump is

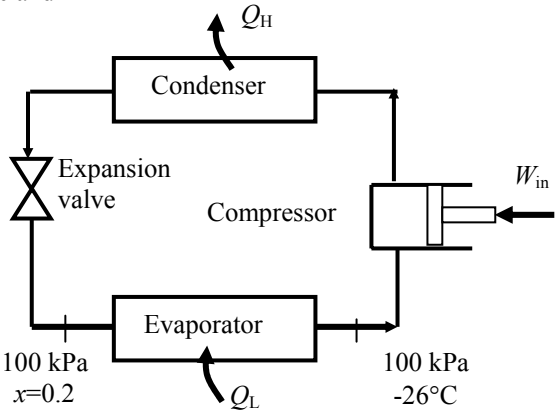
$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.164\ \text{kW}}{1.2\ \text{kW}} = \mathbf{2.64}$$

(*b*) The rate of heat absorbed from the outside air

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = 3.164 - 1.2 = \mathbf{1.96\ kW}$$

6-57 A commercial refrigerator with R-134a as the working fluid is considered. The evaporator inlet and exit states are specified. The mass flow rate of the refrigerant and the rate of heat rejected are to be determined.

Assumptions **1** The refrigerator operates steadily. **2** The kinetic and potential energy changes are zero.



Properties The properties of R-134a at the evaporator inlet and exit states are (Tables A-11 through A-13)

$$\left.\begin{matrix} P_1 = 100\ \text{kPa} \\ x_1 = 0.2 \end{matrix}\right\} h_1 = 60.71\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_2 = 100\ \text{kPa} \\ T_2 = -26°\text{C} \end{matrix}\right\} h_2 = 234.74\ \text{kJ/kg}$$

Analysis (*a*) The refrigeration load is

$$\dot{Q}_L = (\text{COP})\dot{W}_{\text{in}} = (1.2)(0.600\ \text{kW}) = 0.72\ \text{kW}$$

The mass flow rate of the refrigerant is determined from

$$\dot{m}_R = \frac{\dot{Q}_L}{h_2 - h_1} = \frac{0.72\ \text{kW}}{(234.74 - 60.71)\ \text{kJ/kg}} = \mathbf{0.00414\ kg/s}$$

(*b*) The rate of heat rejected from the refrigerator is

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 0.72 + 0.60 = \mathbf{1.32\ kW}$$

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Perpetual-Motion Machines

6-58C This device creates energy, and thus it is a PMM1.

6-59C This device creates energy, and thus it is a PMM1.

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Reversible and Irreversible Processes

6-60C Adiabatic stirring processes are irreversible because the energy stored within the system can not be spontaneously released in a manor to cause the mass of the system to turn the **paddle wheel** in the opposite direction to do work on the surroundings.

6-61C The chemical reactions of combustion processes of a natural gas and air mixture will generate carbon dioxide, water, and other compounds and will release heat energy to a lower temperature surroundings. It is unlikely that the surroundings will return this energy to the reacting system and the products of combustion react spontaneously to reproduce the natural gas and air mixture.

6-62C No. Because it involves heat transfer through a finite temperature difference.

6-63C This process is irreversible. As the block slides down the plane, two things happen, (a) the potential energy of the block decreases, and (b) the block and plane warm up because of the friction between them. The potential energy that has been released can be stored in some form in the surroundings (e.g., perhaps in a spring). When we restore the system to its original condition, we must (a) restore the potential energy by lifting the block back to its original elevation, and (b) cool the block and plane back to their original temperatures.

The potential energy may be restored by returning the energy that was stored during the original process as the block decreased its elevation and released potential energy. The portion of the surroundings in which this energy had been stored would then return to its original condition as the elevation of the block is restored to its original condition.

In order to cool the block and plane to their original temperatures, we have to remove heat from the block and plane. When this heat is transferred to the surroundings, something in the surroundings has to change its state (e.g., perhaps we warm up some water in the surroundings). This change in the surroundings is permanent and cannot be undone. Hence, the original process is irreversible.

6-64C Because reversible processes can be approached in reality, and they form the limiting cases. Work producing devices that operate on reversible processes deliver the most work, and work consuming devices that operate on reversible processes consume the least work.

6-65C When the compression process is non-quasi equilibrium, the molecules before the piston face cannot escape fast enough, forming a high pressure region in front of the piston. It takes more work to move the piston against this high pressure region.

6-66C When an expansion process is non-quasiequilibrium, the molecules before the piston face cannot follow the piston fast enough, forming a low pressure region behind the piston. The lower pressure that pushes the piston produces less work.

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6-67C The irreversibilities that occur within the system boundaries are **internal** irreversibilities; those which occur outside the system boundaries are **external** irreversibilities.

6-68C A reversible expansion or compression process cannot involve unrestrained expansion or sudden compression, and thus it is quasi-equilibrium. A quasi-equilibrium expansion or compression process, on the other hand, may involve external irreversibilities (such as heat transfer through a finite temperature difference), and thus is not necessarily reversible.

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The Carnot Cycle and Carnot's Principle

6-69C The four processes that make up the Carnot cycle are isothermal expansion, reversible adiabatic expansion, isothermal compression, and reversible adiabatic compression.

6-70C They are (1) the thermal efficiency of an irreversible heat engine is lower than the efficiency of a reversible heat engine operating between the same two reservoirs, and (2) the thermal efficiency of all the reversible heat engines operating between the same two reservoirs are equal.

6-71C False. The second Carnot principle states that no heat engine cycle can have a higher thermal efficiency than the Carnot cycle operating between the same temperature limits.

6-72C Yes. The second Carnot principle states that all reversible heat engine cycles operating between the same temperature limits have the same thermal efficiency.

6-73C (a) No, (b) No. They would violate the Carnot principle.

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Carnot Heat Engines

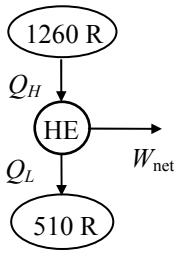
6-74C No.

6-75C The one that has a source temperature of 600°C. This is true because the higher the temperature at which heat is supplied to the working fluid of a heat engine, the higher the thermal efficiency.

6-76E The source and sink temperatures of a heat engine are given. The maximum work per unit heat input to the engine is to be determined.

Assumptions The heat engine operates steadily.

Analysis The maximum work per unit of heat that the engine can remove from the source is the Carnot efficiency, which is determined from

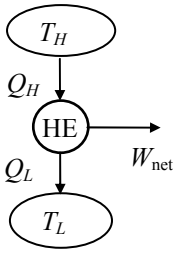


$$\frac{W_{net}}{Q_H} = \eta_{th,max} = 1 - \frac{T_L}{T_H} = 1 - \frac{510\ \text{R}}{1260\ \text{R}} = \mathbf{0.595}$$

6-77 Two pairs of thermal energy reservoirs are to be compared from a work-production perspective.

Assumptions The heat engine operates steadily.

Analysis For the maximum production of work, a heat engine operating between the energy reservoirs would have to be completely reversible. Then, for the first pair of reservoirs



$$\eta_{th,max} = 1 - \frac{T_L}{T_H} = 1 - \frac{325\ \text{K}}{675\ \text{K}} = \mathbf{0.519}$$

For the second pair of reservoirs,

$$\eta_{th,max} = 1 - \frac{T_L}{T_H} = 1 - \frac{275\ \text{K}}{625\ \text{K}} = \mathbf{0.560}$$

The second pair is then capable of producing more work for each unit of heat extracted from the hot reservoir.

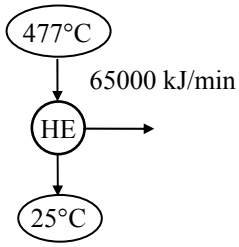
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6-78 The source and sink temperatures of a heat engine and the rate of heat supply are given. The maximum possible power output of this engine is to be determined.

Assumptions The heat engine operates steadily.

Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{298 \text{ K}}{(477 + 273) \text{ K}} = 0.600 \text{ or } 60.0\%$$



Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.600)(65{,}000 \text{ kJ/min}) = 39{,}000 \text{ kJ/min} = \mathbf{653 \text{ kW}}$$

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6-79 Problem 6-78 is reconsidered. The effects of the temperatures of the heat source and the heat sink on the power produced and the cycle thermal efficiency as the source temperature varies from 300°C to 1000°C and the sink temperature varies from 0°C to 50°C are to be studied. The power produced and the cycle efficiency against the source temperature for sink temperatures of 0°C, 25°C, and 50°C are to be plotted.

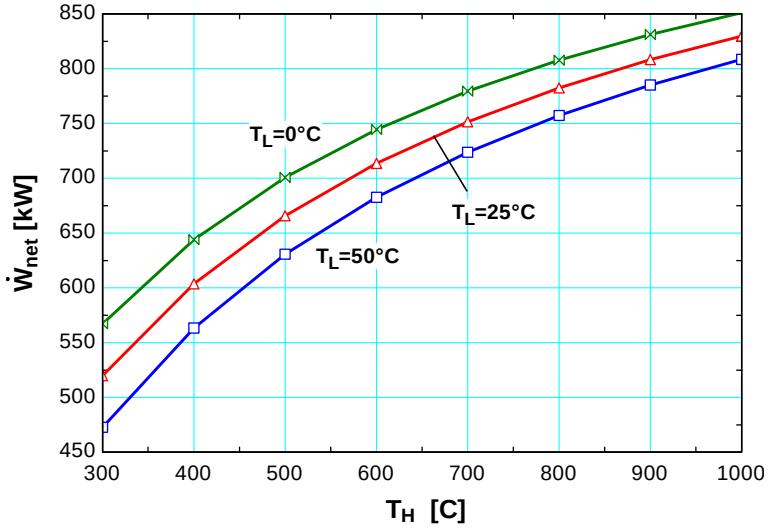
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

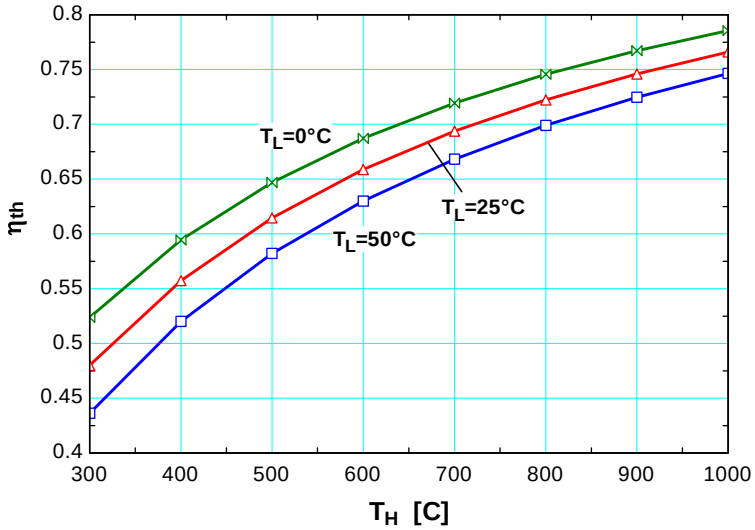
```
T_H = 477 [C]
T_L =25 [C]
Q_dot_H = 65000 [kJ/min]

"First Law applied to the heat engine"
Q_dot_H - Q_dot_L- W_dot_net = 0
W_dot_net_KW=W_dot_net*convert(kJ/min,kW)
"Cycle Thermal Efficiency - Temperatures must be absolute"
eta_th = 1 - (T_L + 273)/(T_H + 273)
"Definition of cycle efficiency"
eta_th=W_dot_net / Q_dot_H
```

T_H [C]	W_{netkW} [kW]	η_{th}
300	567.2	0.5236
400	643.9	0.5944
500	700.7	0.6468
600	744.6	0.6873
700	779.4	0.7194
800	807.7	0.7456
900	831.2	0.7673
1000	851	0.7855

Values for T_L = 0°C





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6-80E The sink temperature of a Carnot heat engine, the rate of heat rejection, and the thermal efficiency are given. The power output of the engine and the source temperature are to be determined.

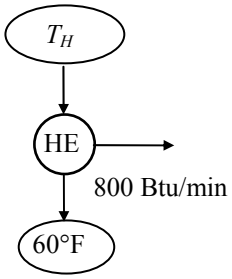
Assumptions The Carnot heat engine operates steadily.

Analysis (a) The rate of heat input to this heat engine is determined from the definition of thermal efficiency,

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} \longrightarrow 0.75 = 1 - \frac{800 \text{ Btu/min}}{\dot{Q}_H} \longrightarrow \dot{Q}_H = 3200 \text{ Btu/min}$$

Then the power output of this heat engine can be determined from

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.75)(3200 \text{ Btu/min}) = 2400 \text{ Btu/min} = \mathbf{56.6\ hp}$$



(*b*) For reversible cyclic devices we have

$$\left(\frac{\dot{Q}_H}{\dot{Q}_L}\right)_{\text{rev}} = \left(\frac{T_H}{T_L}\right)$$

Thus the temperature of the source T_H must be

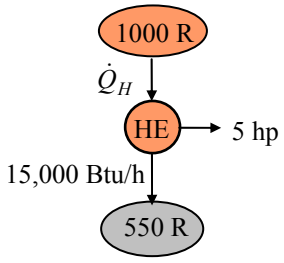
$$T_H = \left(\frac{\dot{Q}_H}{\dot{Q}_L}\right)_{\text{rev}} T_L = \left(\frac{3200 \text{ Btu/min}}{800 \text{ Btu/min}}\right)(520 \text{ R}) = \mathbf{2080\ R}$$

6-81E The claim of an inventor about the operation of a heat engine is to be evaluated.

Assumptions The heat engine operates steadily.

Analysis If this engine were completely reversible, the thermal efficiency would be

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{550 \text{ R}}{1000 \text{ R}} = 0.45$$



When the first law is applied to the engine above,

$$\dot{Q}_H = \dot{W}_{\text{net}} + \dot{Q}_L = (5 \text{ hp})\left(\frac{2544.5 \text{ Btu/h}}{1 \text{ hp}}\right) + 15{,}000 \text{ Btu/h} = 27{,}720 \text{ Btu/h}$$

The actual thermal efficiency of the proposed heat engine is then

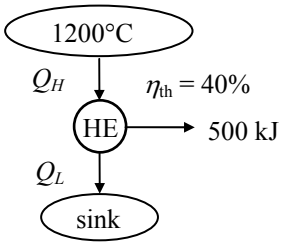
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_H} = \frac{5 \text{ hp}}{27{,}720 \text{ Btu/h}}\left(\frac{2544.5 \text{ Btu/h}}{1 \text{ hp}}\right) = 0.459$$

Since the thermal efficiency of the proposed heat engine is greater than that of a completely reversible heat engine which uses the same isothermal energy reservoirs, **the inventor's claim is invalid**.

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6-82 The work output and thermal efficiency of a Carnot heat engine are given. The heat supplied to the heat engine, the heat rejected and the temperature of heat sink are to be determined.

Assumptions **1** The heat engine operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.



Analysis Applying the definition of the thermal efficiency and an energy balance to the heat engine, the unknown values are determined as follows:

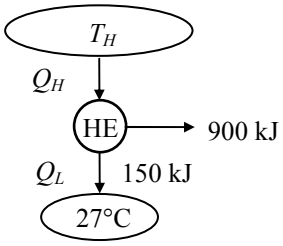
$$Q_H = \frac{W_{net}}{\eta_{th}} = \frac{500\ \text{kJ}}{0.4} = \mathbf{1250\ kJ}$$

$$Q_L = Q_H - W_{net} = 1250 - 500 = \mathbf{750\ kJ}$$

$$\eta_{th,max} = 1 - \frac{T_L}{T_H} \longrightarrow 0.40 = 1 - \frac{T_L}{(1200 + 273)\ \text{K}} \longrightarrow T_L = 883.8\ \text{K} = \mathbf{611°C}$$

6-83 The work output and heat rejection of a a Carnot heat engine are given. The heat supplied to the heat engine and the source temperature are to be determined.

Assumptions **1** The heat engine operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.



Analysis Applying the definition of the thermal efficiency and an energy balance to the heat engine, the unknown values are determined as follows:

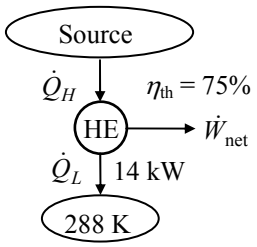
$$Q_H = Q_L + W_{net} = 150 + 900 = \mathbf{1050\ kJ}$$

$$\eta_{th} = \frac{W_{net}}{Q_H} = \frac{900\ \text{kJ}}{1050\ \text{kJ}} = 0.857$$

$$\eta_{th,max} = 1 - \frac{T_L}{T_H} \longrightarrow 0.857 = 1 - \frac{(27 + 273)\ \text{K}}{T_H} \longrightarrow T_H = 2100\ \text{K} = \mathbf{1827°C}$$

6-84 The thermal efficiency and waste heat rejection of a Carnot heat engine are given. The power output and the temperature of the source are to be determined.

Assumptions **1** The heat engine operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.



Analysis Applying the definition of the thermal efficiency and an energy balance to the heat engine, the power output and the source temperature are determined as follows:

$$\eta_{th} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} \longrightarrow 0.75 = 1 - \frac{14\ \text{kW}}{\dot{Q}_H} \longrightarrow \dot{Q}_H = 56\ \text{kW}$$

$$\dot{W}_{net} = \eta_{th}\dot{Q}_H = (0.75)(56\ \text{kW}) = \mathbf{42\ kW}$$

$$\eta_{th} = 1 - \frac{T_L}{T_H} \longrightarrow 0.75 = 1 - \frac{(15 + 273)\ \text{K}}{T_H} \longrightarrow T_H = 1152\ \text{K} = \mathbf{879°C}$$

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6-85 A geothermal power plant uses geothermal liquid water at 150ºC at a specified rate as the heat source. The actual and maximum possible thermal efficiencies and the rate of heat rejected from this power plant are to be determined.

Assumptions **1** The power plant operates steadily. **2** The kinetic and potential energy changes are zero. **3** Steam properties are used for geothermal water.

Properties Using saturated liquid properties, (Table A-4)

$$\left.\begin{matrix} T_{\text{source},1} = 150°\text{C} \\ x_{\text{source}} = 0 \end{matrix}\right\} h_{\text{geo},1} = 632.18 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_{\text{source},2} = 90°\text{C} \\ x_{\text{source}} = 0 \end{matrix}\right\} h_{\text{source}} = 377.04 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_{\text{sink}} = 25°\text{C} \\ x_{\text{sink}} = 0 \end{matrix}\right\} h_{\text{sink}} = 104.83 \text{ kJ/kg}$$

Analysis (*a*) The rate of heat input to the plant is

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{geo}} (h_{\text{geo},1} - h_{\text{geo},2}) = (210 \text{ kg/s})(632.18 - 377.04) \text{ kJ/kg} = 53{,}580 \text{ kW}$$

The actual thermal efficiency is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_{\text{in}}} = \frac{8000 \text{ kW}}{53{,}580 \text{ kW}} = \mathbf{0.1493} = \mathbf{14.9\%}$$

(*b*) The maximum thermal efficiency is the thermal efficiency of a reversible heat engine operating between the source and sink temperatures

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(25 + 273) \text{ K}}{(150 + 273) \text{ K}} = \mathbf{0.2955} = \mathbf{29.6\%}$$

(*c*) Finally, the rate of heat rejection is

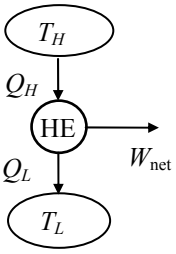
$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net,out}} = 53{,}580 - 8000 = \mathbf{45{,}580 \text{ kW}}$$

6-86 The claim that the efficiency of a completely reversible heat engine can be doubled by doubling the temperature of the energy source is to be evaluated.

Assumptions The heat engine operates steadily.

Analysis The upper limit for the thermal efficiency of any heat engine occurs when a completely reversible engine operates between the same energy reservoirs. The thermal efficiency of this completely reversible engine is given by

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H} = \frac{T_H - T_L}{T_H}$$



If we were to double the absolute temperature of the high temperature energy reservoir, the new thermal efficiency would be

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{2T_H} = \frac{2T_H - T_L}{2T_H} < 2\frac{T_H - T_L}{T_H}$$

The thermal efficiency is then **not doubled** as the temperature of the high temperature reservoir is doubled.

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Carnot Refrigerators and Heat Pumps

6-87C By increasing T_L or by decreasing T_H.

6-88C The difference between the temperature limits is typically much higher for a refrigerator than it is for an air conditioner. The smaller the difference between the temperature limits a refrigerator operates on, the higher is the COP. Therefore, an air-conditioner should have a higher COP.

6-89C The deep freezer should have a lower COP since it operates at a much lower temperature, and in a given environment, the COP decreases with decreasing refrigeration temperature.

6-90C No. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the refrigerator. In reality, the work consumed by the refrigerator will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

6-91C No. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the refrigerator. In reality, the work consumed by the refrigerator will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

6-92C Bad idea. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the heat pump. In reality, the work consumed by the heat pump will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

6-93 The minimum work per unit of heat transfer from the low-temperature source for a refrigerator is to be determined.

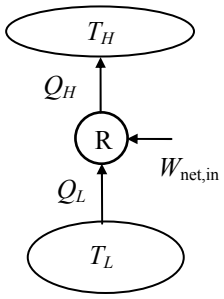
Assumptions The refrigerator operates steadily.

Analysis Application of the first law gives

$$\frac{W_{\text{net,in}}}{Q_L} = \frac{Q_H - Q_L}{Q_L} = \frac{Q_H}{Q_L} - 1$$

For the minimum work input, this refrigerator would be completely reversible and the thermodynamic definition of temperature would reduce the preceding expression to

$$\frac{W_{\text{net,in}}}{Q_L} = \frac{T_H}{T_L} - 1 = \frac{303\ \text{K}}{273\ \text{K}} - 1 = \mathbf{0.110}$$



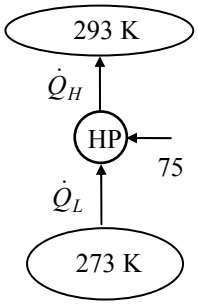
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6-94 The validity of a claim by an inventor related to the operation of a heat pump is to be evaluated.

Assumptions The heat pump operates steadily.

Analysis Applying the definition of the heat pump coefficient of performance,

$$\text{COP}_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_{\text{net,in}}} = \frac{200\ \text{kW}}{75\ \text{kW}} = 2.67$$



The maximum COP of a heat pump operating between the same temperature limits is

$$\text{COP}_{\text{HP,max}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (273\ \text{K})/(293\ \text{K})} = 14.7$$

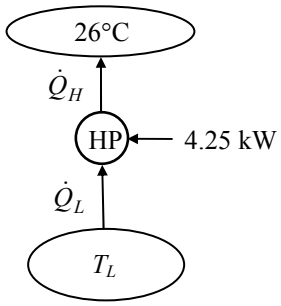
Since the actual COP is less than the maximum COP, the claim is **valid**.

6-95 The power input and the COP of a Carnot heat pump are given. The temperature of the low-temperature reservoir and the heating load are to be determined.

Assumptions The heat pump operates steadily.

Analysis The temperature of the low-temperature reservoir is

$$COP_{\text{HP,max}} = \frac{T_H}{T_H - T_L} \longrightarrow 8.7 = \frac{299\ \text{K}}{(299 - T_L)\ \text{K}} \longrightarrow T_L = \mathbf{264.6\ K}$$



The heating load is

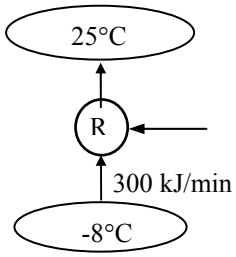
$$COP_{\text{HP,max}} = \frac{\dot{Q}_H}{\dot{W}_{in}} \longrightarrow 8.7 = \frac{\dot{Q}_H}{4.25\ \text{kW}} \longrightarrow \dot{Q}_H = \mathbf{37.0\ kW}$$

6-96 The refrigerated space and the environment temperatures for a refrigerator and the rate of heat removal from the refrigerated space are given. The minimum power input required is to be determined.

Assumptions The refrigerator operates steadily.

Analysis The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only, and is determined from

$$COP_{\text{R,rev}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(25 + 273\ \text{K})/(-8 + 273\ \text{K}) - 1} = 8.03$$

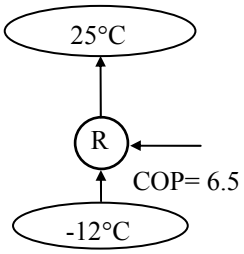


The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_L}{COP_{\text{R,max}}} = \frac{300\ \text{kJ/min}}{8.03} = 37.36\ \text{kJ/min} = \mathbf{0.623\ kW}$$

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6-97 An inventor claims to have developed a refrigerator. The inventor reports temperature and COP measurements. The claim is to be evaluated.



Analysis The highest coefficient of performance a refrigerator can have when removing heat from a cool medium at -12°C to a warmer medium at 25°C is

$$\text{COP}_{\text{R,max}} = \text{COP}_{\text{R,rev}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(25 + 273\ \text{K})/(-12 + 273\ \text{K}) - 1} = 7.1$$

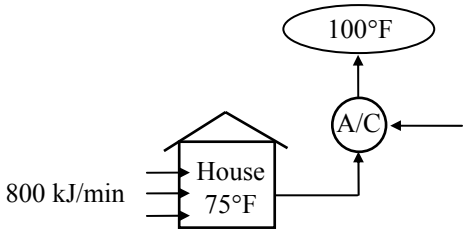
The COP claimed by the inventor is 6.5, which is below this maximum value, thus the claim is **reasonable**. However, it is not probable.

6-98E An air-conditioning system maintains a house at a specified temperature. The rate of heat gain of the house and the rate of internal heat generation are given. The maximum power input required is to be determined.

Assumptions The air-conditioner operates steadily.

Analysis The power input to an air-conditioning system will be a minimum when the air-conditioner operates in a reversible manner. The coefficient of performance of a reversible air-conditioner (or refrigerator) depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(100 + 460\ \text{R})/(70 + 460\ \text{R}) - 1} = 17.67$$



The cooling load of this air-conditioning system is the sum of the heat gain from the outside and the heat generated within the house,

$$\dot{Q}_L = 800 + 100 = 900\ \text{Btu/min}$$

The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{900\ \text{Btu/min}}{17.67} = 50.93\ \text{Btu/min} = \mathbf{1.20\ hp}$$

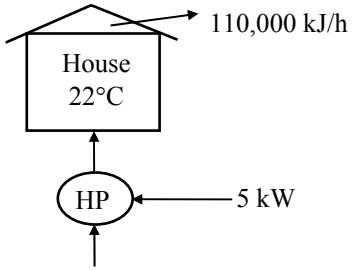
PROPRIETARY MATERIAL. © 2011 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

6-99 A heat pump maintains a house at a specified temperature. The rate of heat loss of the house and the power consumption of the heat pump are given. It is to be determined if this heat pump can do the job.

Assumptions The heat pump operates steadily.

Analysis The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. The coefficient of performance of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\text{ K})/(22+273\text{ K})} = 14.75$$



The required power input to this reversible heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{110{,}000\text{ kJ/h}}{14.75}\left(\frac{1\text{ h}}{3600\text{ s}}\right) = \mathbf{2.07\ kW}$$

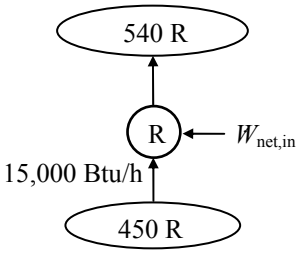
This heat pump is **powerful enough** since 5 kW > 2.07 kW.

6-100E The power required by a reversible refrigerator with specified reservoir temperatures is to be determined.

Assumptions The refrigerator operates steadily.

Analysis The COP of this reversible refrigerator is

$$\text{COP}_{\text{R,max}} = \frac{T_L}{T_H - T_L} = \frac{450\text{ R}}{540\text{ R} - 450\text{ R}} = 5$$



Using this result in the coefficient of performance expression yields

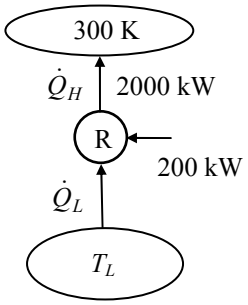
$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{15{,}000\text{ Btu/h}}{5}\left(\frac{1\text{ kW}}{3412.14\text{ Btu/h}}\right) = \mathbf{0.879\ kW}$$

6-101 The power input and heat rejection of a reversed Carnot cycle are given. The cooling load and the source temperature are to be determined.

Assumptions The refrigerator operates steadily.

Analysis Applying the definition of the refrigerator coefficient of performance,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 2000 - 200 = \mathbf{1800\ kW}$$



Applying the definition of the heat pump coefficient of performance,

$$\text{COP}_{\text{R}} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{1800\text{ kW}}{200\text{ kW}} = 9$$

The temperature of the heat source is determined from

$$\text{COP}_{\text{R,max}} = \frac{T_L}{T_H - T_L} \longrightarrow 9 = \frac{T_L}{300 - T_L} \longrightarrow T_L = 270\text{ K} = \mathbf{-3°C}$$

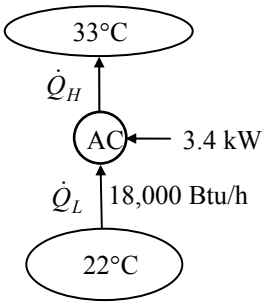
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6-102 The power input and the cooling load of an air conditioner are given. The rate of heat rejected in the condenser, the COP, and the rate of cooling for a reversible operation are to be determined.

Assumptions The air conditioner operates steadily.

Analysis (a) The rate of heat rejected is

$$\begin{aligned}\dot{Q}_H &= \dot{Q}_L + \dot{W}_{\text{in}} \\ &= (18{,}000\ \text{Btu/h})\left(\frac{1.055\ \text{kJ}}{1\ \text{Btu}}\right) + (3.4\ \text{kW})\left(\frac{3600\ \text{kJ/h}}{1\ \text{kW}}\right) \\ &= \mathbf{31{,}230\ kJ/h}\end{aligned}$$



(b) The COP is

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{(18{,}000\ \text{Btu/h})\left(\frac{1\ \text{kW}}{3412\ \text{Btu/h}}\right)}{3.4\ \text{kW}} = \mathbf{1.552}$$

(*c*) The rate of cooling if the air conditioner operated as a Carnot refrigerator for the same power input is

$$\text{COP}_{\text{rev}} = \frac{T_L}{T_H - T_L} = \frac{295\ \text{K}}{(33-22)\ \text{K}} = 26.82$$

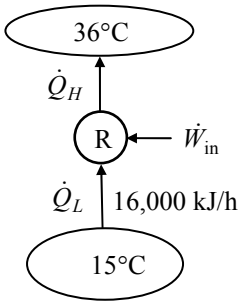
$$\dot{Q}_{L,\max} = \text{COP}_{\text{rev}}\dot{W}_{\text{in,min}} = (26.82)\left[(3.4\ \text{kW})\left(\frac{3412\ \text{Btu/h}}{1\ \text{kW}}\right)\right] = \mathbf{311{,}130\ Btu/h}$$

6-103 The rate of heat removal and the source and sink temperatures are given for a Carnot refrigerator. The COP of the refrigerator and the power input are to be determined.

Assumptions The refrigerator operates steadily.

Analysis The COP of the Carnot refrigerator is determined from

$$COP_{\text{R,max}} = \frac{T_L}{T_H - T_L} = \frac{288\ \text{K}}{(36-15)\ \text{K}} = \mathbf{13.71}$$



The power input is

$$COP_{\text{R,max}} = \frac{\dot{Q}_L}{\dot{W}_{in}} \longrightarrow 13.71 = \frac{16{,}000\ \text{kJ/h}}{\dot{W}_{in}} \longrightarrow \dot{W}_{in} = 1167\ \text{kJ/h} = \mathbf{0.324\ kW}$$

The rate of heat rejected is

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{net,in}} = 16{,}000\ \text{kJ/h} + 1167\ \text{kJ/h} = \mathbf{17{,}167\ kJ/h}$$

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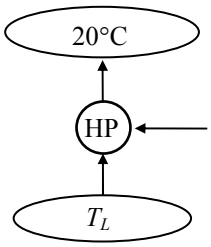
6-104 A heat pump maintains a house at a specified temperature in winter. The maximum COPs of the heat pump for different outdoor temperatures are to be determined.

Analysis The coefficient of performance of a heat pump will be a maximum when the heat pump operates in a reversible manner. The coefficient of performance of a reversible heat pump depends on the temperature limits in the cycle only, and is determined for all three cases above to be

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(10+273\text{K})/(20+273\text{K})} = \mathbf{29.3}$$

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(-5+273\text{K})/(20+273\text{K})} = \mathbf{11.7}$$

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(-30+273\text{K})/(20+273\text{K})} = \mathbf{5.86}$$



6-105E A heat pump maintains a house at a specified temperature. The rate of heat loss of the house is given. The minimum power inputs required for different source temperatures are to be determined.

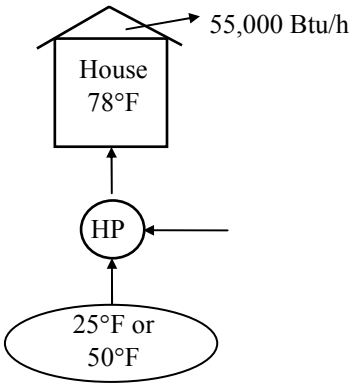
Assumptions The heat pump operates steadily.

Analysis (*a*) The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. If the outdoor air at 25°F is used as the heat source, the COP of the heat pump and the required power input are determined to be

$$\text{COP}_{\text{HP,max}} = \text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(25+460\text{ R})/(78+460\text{ R})} = 10.15$$

and

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,max}}} = \frac{55{,}000\text{ Btu/h}}{10.15}\left(\frac{1\text{ hp}}{2545\text{ Btu/h}}\right) = \mathbf{2.13\ hp}$$



(*b*) If the well-water at 50°F is used as the heat source, the COP of the heat pump and the required power input are determined to be

$$\text{COP}_{\text{HP,max}} = \text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(50+460\text{ R})/(78+460\text{ R})} = 19.2$$

and

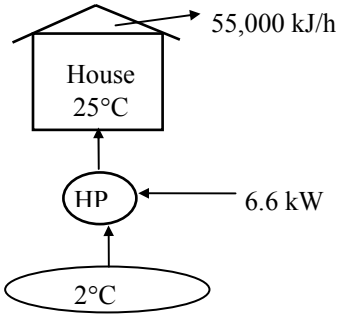
$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,max}}} = \frac{55{,}000\text{ Btu/h}}{19.2}\left(\frac{1\text{ hp}}{2545\text{ Btu/h}}\right) = \mathbf{1.13\ hp}$$

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6-106 A Carnot heat pump consumes 6.6-kW of power when operating, and maintains a house at a specified temperature. The average rate of heat loss of the house in a particular day is given. The actual running time of the heat pump that day, the heating cost, and the cost if resistance heating is used instead are to be determined.

Analysis (*a*) The coefficient of performance of this Carnot heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\ \text{K})/(25+273\ \text{K})} = 12.96$$



The amount of heat the house lost that day is

$$Q_H = \dot{Q}_H(1\ \text{day}) = (55{,}000\ \text{kJ/h})(24\ \text{h}) = 1{,}320{,}000\ \text{kJ}$$

Then the required work input to this Carnot heat pump is determined from the definition of the coefficient of performance to be

$$W_{\text{net,in}} = \frac{Q_H}{\text{COP}_{\text{HP}}} = \frac{1{,}320{,}000\ \text{kJ}}{12.96} = 101{,}880\ \text{kJ}$$

Thus the length of time the heat pump ran that day is

$$\Delta t = \frac{W_{\text{net,in}}}{\dot{W}_{\text{net,in}}} = \frac{101{,}880\ \text{kJ}}{6.6\ \text{kJ/s}} = 15{,}440\ \text{s} = \mathbf{4.29\ h}$$

(*b*) The total heating cost that day is

$$\text{Cost} = W \times \text{price} = (\dot{W}_{\text{net,in}} \times \Delta t)(\text{price}) = (6.6\ \text{kW})(4.29\ \text{h})(0.085\ \$/\text{kWh}) = \mathbf{\$2.41}$$

(*c*) If resistance heating were used, the entire heating load for that day would have to be met by electrical energy. Therefore, the heating system would consume 1,320,000 kJ of electricity that would cost

$$\text{New Cost} = Q_H \times \text{price} = (1{,}320{,}000\text{kJ})\left(\frac{1\ \text{kWh}}{3600\ \text{kJ}}\right)(0.085\ \$/\text{kWh}) = \mathbf{\$31.2}$$

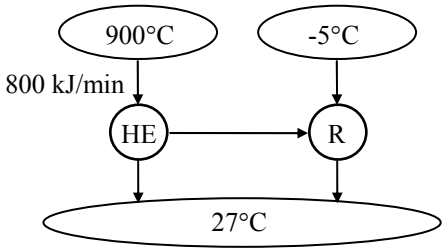
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6-107 A Carnot heat engine is used to drive a Carnot refrigerator. The maximum rate of heat removal from the refrigerated space and the total rate of heat rejection to the ambient air are to be determined.

Assumptions The heat engine and the refrigerator operate steadily.

Analysis (a) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{1173 \text{ K}} = 0.744$$



Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.744)(800 \text{ kJ/min}) = 595.2 \text{ kJ/min}$$

which is also the power input to the refrigerator, $\dot{W}_{\text{net,in}}$.

The rate of heat removal from the refrigerated space will be a maximum if a Carnot refrigerator is used. The COP of the Carnot refrigerator is

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(27 + 273 \text{ K})/(-5 + 273 \text{ K}) - 1} = 8.37$$

Then the rate of heat removal from the refrigerated space becomes

$$\dot{Q}_{L,\text{R}} = (\text{COP}_{\text{R,rev}})(\dot{W}_{\text{net,in}}) = (8.37)(595.2 \text{ kJ/min}) = \mathbf{4982 \text{ kJ/min}}$$

(b) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\text{R}}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 800 - 595.2 = 204.8 \text{ kJ/min}$$
$$\dot{Q}_{H,\text{R}} = \dot{Q}_{L,\text{R}} + \dot{W}_{\text{net,in}} = 4982 + 595.2 = 5577.2 \text{ kJ/min}$$

and

$$\dot{Q}_{\text{ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,\text{R}} = 204.8 + 5577.2 = \mathbf{5782 \text{ kJ/min}}$$

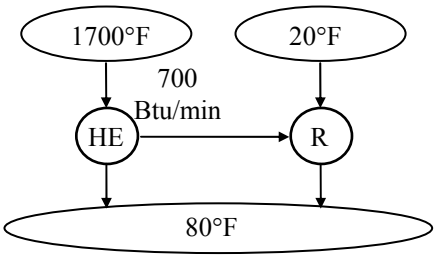
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6-108E A Carnot heat engine is used to drive a Carnot refrigerator. The maximum rate of heat removal from the refrigerated space and the total rate of heat rejection to the ambient air are to be determined.

Assumptions The heat engine and the refrigerator operate steadily.

Analysis (*a*) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540\ \text{R}}{2160\ \text{R}} = 0.75$$



Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.75)(700\ \text{Btu/min}) = 525\ \text{Btu/min}$$

which is also the power input to the refrigerator, $\dot{W}_{\text{net,in}}$.

The rate of heat removal from the refrigerated space will be a maximum if a Carnot refrigerator is used. The COP of the Carnot refrigerator is

$$COP_{\text{R,rev}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(80 + 460\ \text{R})/(20 + 460\ \text{R}) - 1} = 8.0$$

Then the rate of heat removal from the refrigerated space becomes

$$\dot{Q}_{L,\text{R}} = (\text{COP}_{\text{R,rev}})(\dot{W}_{\text{net,in}}) = (8.0)(525\ \text{Btu/min}) = \mathbf{4200\ Btu/min}$$

(*b*) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\text{R}}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 700 - 525 = 175\ \text{Btu/min}$$
$$\dot{Q}_{H,\text{R}} = \dot{Q}_{L,\text{R}} + \dot{W}_{\text{net,in}} = 4200 + 525 = 4725\ \text{Btu/min}$$

and

$$\dot{Q}_{\text{ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,\text{R}} = 175 + 4725 = \mathbf{4900\ \ Btu/min}$$

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6-109 A heat pump that consumes 4-kW of power when operating maintains a house at a specified temperature. The house is losing heat in proportion to the temperature difference between the indoors and the outdoors. The lowest outdoor temperature for which this heat pump can do the job is to be determined.

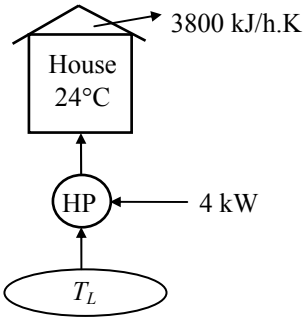
Assumptions The heat pump operates steadily.

Analysis Denoting the outdoor temperature by T_L, the heating load of this house can be expressed as

$$\dot{Q}_H = (3800 \text{ kJ/h} \cdot \text{K})(297 - T_L) = (1.056 \text{ kW/K})(297 - T_L)\text{K}$$

The coefficient of performance of a Carnot heat pump depends on the temperature limits in the cycle only, and can be expressed as

$$\text{COP}_{\text{HP}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - T_L/(297 \text{ K})}$$



or, as

$$\text{COP}_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_{\text{net,in}}} = \frac{(1.056 \text{ kW/K})(297 - T_L)\text{K}}{4 \text{ kW}}$$

Equating the two relations above and solving for T_L, we obtain

$$T_L = 263.5 \text{ K} = \mathbf{-9.5°C}$$

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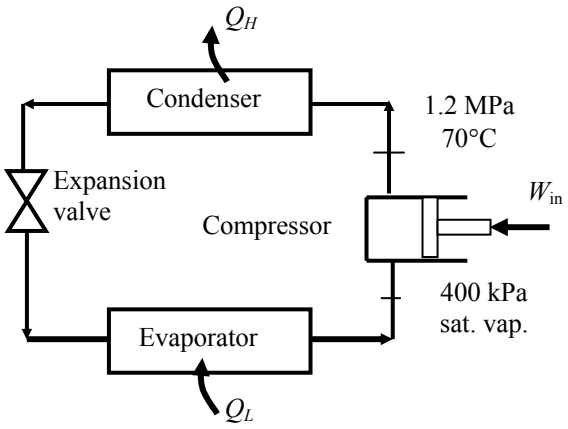
6-110 An air-conditioner with R-134a as the working fluid is considered. The compressor inlet and exit states are specified. The actual and maximum COPs and the minimum volume flow rate of the refrigerant at the compressor inlet are to be determined.

Assumptions **1** The air-conditioner operates steadily. **2** The kinetic and potential energy changes are zero.

Properties The properties of R-134a at the compressor inlet and exit states are (Tables A-11 through A-13)

$$\left.\begin{matrix} P_1 = 400\ \text{kPa} \\ x_1 = 1 \end{matrix}\right\} \begin{matrix} h_1 = 255.55\ \text{kJ/kg} \\ \nu_1 = 0.05120\ \text{m}^3/\text{kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 1.2\ \text{MPa} \\ T_2 = 70°\text{C} \end{matrix}\right\} h_2 = 300.61\ \text{kJ/kg}$$



Analysis (*a*) The mass flow rate of the refrigerant and the power consumption of the compressor are

$$\dot{m}_R = \frac{\dot{\nu}_1}{\nu_1} = \frac{100\ \text{L/min}\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right)}{0.05120\ \text{m}^3/\text{kg}} = 0.03255\ \text{kg/s}$$

$$\dot{W}_{\text{in}} = \dot{m}_R (h_2 - h_1) = (0.03255\ \text{kg/s})(300.61 - 255.55)\ \text{kJ/kg} = 1.467\ \text{kW}$$

The heat gains to the room must be rejected by the air-conditioner. That is,

$$\dot{Q}_L = \dot{Q}_{\text{heat}} + \dot{Q}_{\text{equipment}} = (250\ \text{kJ/min})\left(\frac{1\ \text{min}}{60\ \text{s}}\right) + 0.9\ \text{kW} = 5.067\ \text{kW}$$

Then, the actual COP becomes

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.067\ \text{kW}}{1.467\ \text{kW}} = \mathbf{3.45}$$

(*b*) The COP of a reversible refrigerator operating between the same temperature limits is

$$\text{COP}_{\text{max}}\ \frac{1}{T_H / T_L - 1} = \frac{1}{(37 + 273)/(23 + 273) - 1} = \mathbf{21.14}$$

(*c*) The minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{max}}} = \frac{5.067\ \text{kW}}{21.14} = 0.2396\ \text{kW}$$

The minimum mass flow rate is

$$\dot{m}_{R,\text{min}} = \frac{\dot{W}_{\text{in,min}}}{h_2 - h_1} = \frac{0.2396\ \text{kW}}{(300.61 - 255.55)\ \text{kJ/kg}} = 0.005318\ \text{kg/s}$$

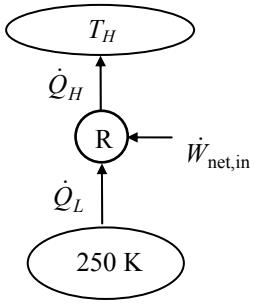
Finally, the minimum volume flow rate at the compressor inlet is

$$\dot{\nu}_{\text{min},1} = \dot{m}_{R,\text{min}} \nu_1 = (0.005318\ \text{kg/s})(0.05120\ \text{m}^3/\text{kg}) = 0.0002723\ \text{m}^3/\text{s} = \mathbf{16.3\ L/min}$$

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6-111 The COP of a completely reversible refrigerator as a function of the temperature of the sink is to be calculated and plotted.

Assumptions The refrigerator operates steadily.

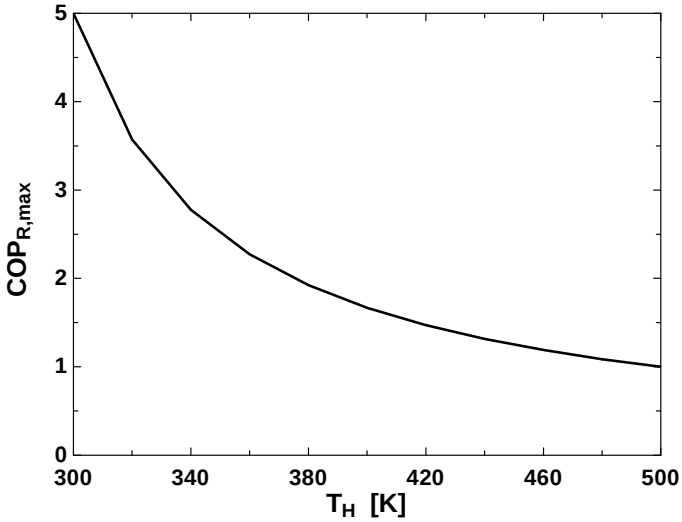


Analysis The coefficient of performance for this completely reversible refrigerator is given by

$$\text{COP}_{\text{R,max}} = \frac{T_L}{T_H - T_L} = \frac{250\ \text{K}}{T_H - 250\ \text{K}}$$

Using EES, we tabulate and plot the variation of COP with the sink temperature as follows:

T_H [K]	$\text{COP}_{\text{R,max}}$
300	5
320	3.571
340	2.778
360	2.273
380	1.923
400	1.667
420	1.471
440	1.316
460	1.19
480	1.087
500	1



6-112 An expression for the COP of a completely reversible refrigerator in terms of the thermal-energy reservoir temperatures, T_L and T_H is to be derived.

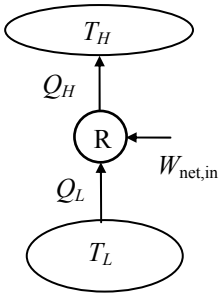
Assumptions The refrigerator operates steadily.

Analysis Application of the first law to the completely reversible refrigerator yields

$$W_{\text{net,in}} = Q_H - Q_L$$

This result may be used to reduce the coefficient of performance,

$$\text{COP}_{\text{R,rev}} = \frac{Q_L}{W_{\text{net,in}}} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{Q_H / Q_L - 1}$$



Since this refrigerator is completely reversible, the thermodynamic definition of temperature tells us that,

$$\frac{Q_H}{Q_L} = \frac{T_H}{T_L}$$

When this is substituted into the COP expression, the result is

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H / T_L - 1} = \frac{T_L}{T_H - T_L}$$

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Special Topic: Household Refrigerators

6-113C The energy consumption of a household refrigerator can be reduced by practicing good conservation measures such as (1) opening the refrigerator door the fewest times possible and for the shortest duration possible, (2) cooling the hot foods to room temperature first before putting them into the refrigerator, (3) cleaning the condenser coils behind the refrigerator, (4) checking the door gasket for air leaks, (5) avoiding unnecessarily low temperature settings, (6) avoiding excessive ice build-up on the interior surfaces of the evaporator, (7) using the power-saver switch that controls the heating coils that prevent condensation on the outside surfaces in humid environments, and (8) not blocking the air flow passages to and from the condenser coils of the refrigerator.

6-114C It is important to clean the condenser coils of a household refrigerator a few times a year since the dust that collects on them serves as insulation and slows down heat transfer. Also, it is important not to block air flow through the condenser coils since heat is rejected through them by natural convection, and blocking the air flow will interfere with this heat rejection process. A refrigerator cannot work unless it can reject the waste heat.

6-115C Today's refrigerators are much more efficient than those built in the past as a result of using smaller and higher efficiency motors and compressors, better insulation materials, larger coil surface areas, and better door seals.

6-116C It is a bad idea to overdesign the refrigeration system of a supermarket so that the entire air-conditioning needs of the store can be met by refrigerated air without installing any air-conditioning system. This is because the refrigerators cool the air to a much lower temperature than needed for air conditioning, and thus their efficiency is much lower, and their operating cost is much higher.

6-117C It is a bad idea to meet the entire refrigerator/freezer requirements of a store by using a large freezer that supplies sufficient cold air at -20°C instead of installing separate refrigerators and freezers . This is because the freezers cool the air to a much lower temperature than needed for refrigeration, and thus their efficiency is much lower, and their operating cost is much higher.

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6-118 A refrigerator consumes 300 W when running, and $74 worth of electricity per year under normal use. The fraction of the time the refrigerator will run in a year is to be determined.

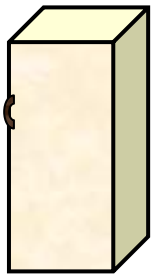
Assumptions The electricity consumed by the light bulb is negligible.

Analysis The total amount of electricity the refrigerator uses a year is

$$\text{Total electric energy used} = W_{e,\text{total}} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$74/\text{year}}{\$0.07/\text{kWh}} = 1057\ \text{kWh/year}$$

The number of hours the refrigerator is on per year is

$$\text{Total operating hours} = \Delta t = \frac{W_{e,\text{total}}}{\dot{W}_e} = \frac{1057\ \text{kWh/year}}{0.3\ \text{kW}} = 3524\ \text{h/year}$$



Noting that there are 365×24=8760 hours in a year, the fraction of the time the refrigerator is on during a year is determined to be

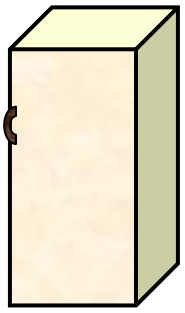
$$\text{Time fraction on} = \frac{\text{Total operating hours}}{\text{Total hours per year}} = \frac{3524/\text{year}}{8760\ \text{h/year}} = \mathbf{0.402}$$

Therefore, the refrigerator remained on 40.2% of the time.

6-119 The light bulb of a refrigerator is to be replaced by a $25 energy efficient bulb that consumes less than half the electricity. It is to be determined if the energy savings of the efficient light bulb justify its cost.

Assumptions The new light bulb remains on the same number of hours a year.

Analysis The lighting energy saved a year by the energy efficient bulb is



$$\begin{aligned}\text{Lighting energy saved} &= (\text{Lighting power saved})(\text{Operating hours})\\ &= [(40-18)\text{W}](60\ \text{h/year})\\ &= 1320\ \text{Wh} = 1.32\ \text{kWh}\end{aligned}$$

This means 1.32 kWh less heat is supplied to the refrigerated space by the light bulb, which must be removed from the refrigerated space. This corresponds to a refrigeration savings of

$$\text{Refrigeration energy saved} = \frac{\text{Lighting energy saved}}{\text{COP}} = \frac{1.32\ \text{kWh}}{1.3} = 1.02\ \text{kWh}$$

Then the total electrical energy and money saved by the energy efficient light bulb become

$$\text{Total energy saved} = (\text{Lighting} + \text{Refrigeration})\ \text{energy saved} = 1.32 + 1.02 = 2.34\ \text{kWh/year}$$

$$\begin{aligned}\text{Money saved} &= (\text{Total energy saved})(\text{Unit cost of energy}) = (2.34\ \text{kWh/year})(\$0.08/\text{kWh})\\ &= \mathbf{\$0.19\ /\ year}\end{aligned}$$

That is, the light bulb will save only 19 cents a year in energy costs, and it will take $25/$0.19 = 132 years for it to pay for itself from the energy it saves. Therefore, it is **not justified** in this case.

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6-120 A person cooks three times a week and places the food into the refrigerator before cooling it first. The amount of money this person will save a year by cooling the hot foods to room temperature before refrigerating them is to be determined.

Assumptions **1** The heat stored in the pan itself is negligible. **2** The specific heat of the food is constant.

Properties The specific heat of food is c = 3.90 kJ/kg.°C (given).

Analysis The amount of hot food refrigerated per year is

$$m_{\text{food}} = (5\text{ kg/pan})(3\text{ pans/week})(52\text{ weeks/year}) = 780\text{ kg/year}$$

The amount of energy removed from food as it is unnecessarily cooled to room temperature in the refrigerator is

$$\text{Energy removed} = Q_{\text{out}} = m_{\text{food}} c\Delta T = (780\text{ kg/year})(3.90\text{ kJ/kg.°C})(95 - 23)°\text{C} = 219{,}024\text{ kJ/year}$$

$$\text{Energy saved} = E_{\text{saved}} = \frac{\text{Energy removed}}{\text{COP}} = \frac{219{,}024\text{ kJ/year}}{1.5}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = 40.56\text{ kWh/year}$$

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (40.56\text{ kWh/year})(\$0.10/\text{kWh}) = \mathbf{\$4.06/year}$$

Therefore, cooling the food to room temperature before putting it into the refrigerator will save about four dollars a year.

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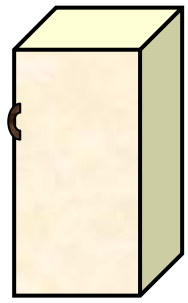
6-121 The door of a refrigerator is opened 8 times a day, and half of the cool air inside is replaced by the warmer room air. The cost of the energy wasted per year as a result of opening the refrigerator door is to be determined for the cases of moist and dry air in the room.

Assumptions **1** The room is maintained at 20°C and 95 kPa at all times. **2** Air is an ideal gas with constant specific heats at room temperature. **3** The moisture is condensed at an average temperature of 4°C. **4** Half of the air volume in the refrigerator is replaced by the warmer kitchen air each time the door is opened.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg·K (Table A-1). The specific heat of air at room temperature is $c_p = 1.005$ kJ/kg·°C (Table A-2a). The heat of vaporization of water at 4°C is $h_{fg} = 2492$ kJ/kg (Table A-4).

Analysis The volume of the refrigerated air replaced each time the refrigerator is opened is 0.3 m^3 (half of the 0.6 m^3 air volume in the refrigerator). Then the total volume of refrigerated air replaced by room air per year is

$$\boldsymbol{V}_{\text{air, replaced}} = (0.3\ \text{m}^3)(8/\text{day})(365\ \text{days/year}) = 876\ \text{m}^3/\text{year}$$



The density of air at the refrigerated space conditions of 95 kPa and 4°C and the mass of air replaced per year are

$$\rho_o = \frac{P_o}{RT_o} = \frac{95\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(4+273\ \text{K})} = 1.195\ \text{kg/m}^3$$

$$m_{\text{air}} = \rho \boldsymbol{V}_{\text{air}} = (1.195\ \text{kg/m}^3)(876\ \text{m}^3/\text{year}) = 1047\ \text{kg/year}$$

The amount of moisture condensed and removed by the refrigerator is

$$\begin{aligned} m_{\text{moisture}} &= m_{\text{air}}\,(\text{moisture removed per kg air}) = (1047\ \text{kg air/year})(0.006\ \text{kg/kg air}) \\ &= 6.28\ \text{kg/year} \end{aligned}$$

The sensible, latent, and total heat gains of the refrigerated space become

$$\begin{aligned} Q_{\text{gain,sensible}} &= m_{\text{air}} c_p (T_{\text{room}} - T_{\text{refrig}}) \\ &= (1047\ \text{kg/year})(1.005\ \text{kJ/kg.°C})(20-4)°\text{C} = 16{,}836\ \text{kJ/year} \\ Q_{\text{gain,latent}} &= m_{\text{moisture}} h_{\text{fg}} \\ &= (6.28\ \text{kg/year})(2492\ \text{kJ/kg}) = 15{,}650\ \text{kJ/year} \\ Q_{\text{gain,total}} &= Q_{\text{gain,sensible}} + Q_{\text{gain,latent}} = 16{,}836 + 15{,}650 = 32{,}486\ \text{kJ/year} \end{aligned}$$

For a COP of 1.4, the amount of electrical energy the refrigerator will consume to remove this heat from the refrigerated space and its cost are

$$\text{Electrical energy used (total)} = \frac{Q_{\text{gain,total}}}{\text{COP}} = \frac{32{,}486\ \text{kJ/year}}{1.4}\left(\frac{1\ \text{kWh}}{3600\ \text{kJ}}\right) = 6.45\ \text{kWh/year}$$

$$\begin{aligned} \text{Cost of energy used (total)} &= (\text{Energy used})(\text{Unit cost of energy}) \\ &= (6.45\ \text{kWh/year})(\$0.075/\text{kWh}) = \mathbf{\$0.48/year} \end{aligned}$$

If the room air is very dry and thus latent heat gain is negligible, then the amount of electrical energy the refrigerator will consume to remove the sensible heat from the refrigerated space and its cost become

$$\text{Electrical energy used (sensible)} = \frac{Q_{\text{gain,sensible}}}{\text{COP}} = \frac{16{,}836\ \text{kJ/year}}{1.4}\left(\frac{1\ \text{kWh}}{3600\ \text{kJ}}\right) = 3.34\ \text{kWh/year}$$

$$\begin{aligned} \text{Cost of energy used (sensible)} &= (\text{Energy used})(\text{Unit cost of energy}) \\ &= (3.34\ \text{kWh/year})(\$0.075/\text{kWh}) = \mathbf{\$0.25/year} \end{aligned}$$

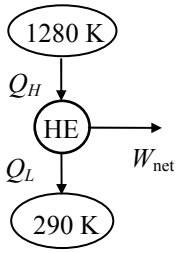
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Review Problems

6-122 The source and sink temperatures of a heat engine are given. The maximum work per unit heat input to the engine is to be determined.

Assumptions The heat engine operates steadily.

Analysis The maximum work per unit of heat that the engine can remove from the source is the Carnot efficiency, which is determined from

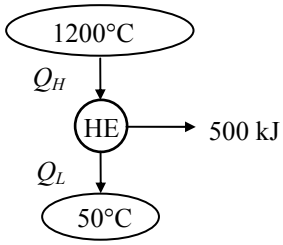


$$\frac{W_{\text{net}}}{Q_H} = \eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{290\ \text{K}}{1280\ \text{K}} = \mathbf{0.773}$$

6-123 The work output and the source and sink temperatures of a Carnot heat engine are given. The heat supplied to and rejected from the heat engine are to be determined.

Assumptions **1** The heat engine operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.

Analysis Applying the definition of the thermal efficiency and an energy balance to the heat engine, the unknown parameters are determined as follows:



$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(50+273)\ \text{K}}{(1200+273)\ \text{K}} 0.781$$

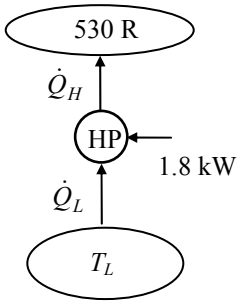
$$Q_H = \frac{W_{\text{net}}}{\eta_{\text{th}}} = \frac{500\ \text{kJ}}{0.781} = \mathbf{640\ kJ}$$

$$Q_L = Q_H - W_{\text{net}} = 640 - 500 = \mathbf{140\ kJ}$$

6-124E The operating conditions of a heat pump are given. The minimum temperature of the source that satisfies the second law of thermodynamics is to be determined.

Assumptions The heat pump operates steadily.

Analysis Applying the first law to this heat pump gives



$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 32{,}000\ \text{Btu/h} - (1.8\ \text{kW})\left(\frac{3412.14\ \text{Btu/h}}{1\ \text{kW}}\right) = 25{,}860\ \text{Btu/h}$$

In the reversible case we have

$$\frac{T_L}{T_H} = \frac{\dot{Q}_L}{\dot{Q}_H}$$

Then the minimum temperature may be determined to be

$$T_L = T_H \frac{\dot{Q}_L}{\dot{Q}_H} = (530\ \text{R})\frac{25{,}860\ \text{Btu/h}}{32{,}000\ \text{Btu/h}} = \mathbf{428\ R}$$

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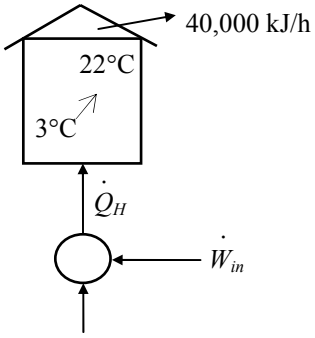
6-125 A heat pump with a specified COP is to heat a house. The rate of heat loss of the house and the power consumption of the heat pump are given. The time it will take for the interior temperature to rise from 3°C to 22°C is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The house is well-sealed so that no air leaks in or out. **3** The COP of the heat pump remains constant during operation.

Properties The constant volume specific heat of air at room temperature is c_v = 0.718 kJ/kg.°C (Table A-2)

Analysis The house is losing heat at a rate of

$$\dot{Q}_{\text{Loss}} = 40{,}000 \text{ kJ/h} = 11.11 \text{ kJ/s}$$



The rate at which this heat pump supplies heat is

$$\dot{Q}_H = \text{COP}_{\text{HP}}\dot{W}_{\text{net,in}} = (2.4)(8 \text{ kW}) = 19.2 \text{ kW}$$

That is, this heat pump can supply heat at a rate of 19.2 kJ/s. Taking the house as the system (a closed system), the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$Q_{\text{in}} - Q_{\text{out}} = mc_v(T_2 - T_1)$$

$$(\dot{Q}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = mc_v(T_2 - T_1)$$

Substituting,

$$(19.2 - 11.11 \text{kJ/s})\Delta t = (2000 \text{kg})(0.718 \text{kJ/kg}\cdot^\circ\text{C})(22 - 3)^\circ\text{C}$$

Solving for Δt, it will take

$$\Delta t = 3373 \text{ s} = \mathbf{0.937 \text{ h}}$$

for the temperature in the house to rise to 22°C.

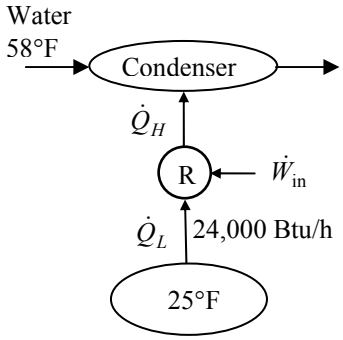
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6-126E A refrigerator with a water-cooled condenser is considered. The cooling load and the COP of a refrigerator are given. The power input, the exit temperature of water, and the maximum possible COP of the refrigerator are to be determined.

Assumptions The refrigerator operates steadily.

Analysis (a) The power input is

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}} = \frac{24{,}000\ \text{Btu/h}}{1.77}\left(\frac{1.055\ \text{kJ}}{1\ \text{Btu}}\right)\left(\frac{1\ \text{h}}{3600\ \text{s}}\right) = \mathbf{3.974\ kW}$$



(b) The rate of heat rejected in the condenser is

$$\begin{aligned}\dot{Q}_H &= \dot{Q}_L + \dot{W}_{\text{in}} \\ &= 24{,}000\ \text{Btu/h} + 3.974\ \text{kW}\left(\frac{1\ \text{Btu}}{1.055\ \text{kJ}}\right)\left(\frac{3600\ \text{s}}{1\ \text{h}}\right) \\ &= 37{,}560\ \text{Btu/h}\end{aligned}$$

The exit temperature of the water is

$$\begin{aligned}\dot{Q}_H &= \dot{m}c_p(T_2 - T_1) \\ T_2 &= T_1 + \frac{\dot{Q}_H}{\dot{m}c_p} \\ &= 58°\text{F} + \frac{37{,}560\ \text{Btu/h}}{(1.45\ \text{lbm/s})\left(\frac{3600\ \text{s}}{1\ \text{h}}\right)(1.0\ \text{Btu/lbm}\cdot°\text{F})} = \mathbf{65.2°F}\end{aligned}$$

(c) Taking the temperature of high-temperature medium to be the average temperature of water in the condenser,

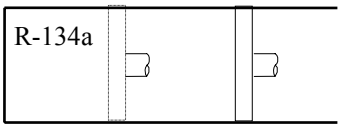
$$\text{COP}_{\text{rev}} = \frac{T_L}{T_H - T_L} = \frac{25 + 460}{0.5(58 + 65.2) - 25} = \mathbf{13.3}$$

6-127 A Carnot heat engine cycle is executed in a closed system with a fixed mass of R-134a. The thermal efficiency of the cycle is given. The net work output of the engine is to be determined.

Assumptions All components operate steadily.

Properties The enthalpy of vaporization of R-134a at 50°C is h_{fg} = 151.79 kJ/kg (Table A-11).

Analysis The enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that T or P. Therefore, the amount of heat transfer to R-134a during the heat addition process of the cycle is



Carnot HE

$$Q_H = mh_{fg@50°\text{C}} = (0.01\ \text{kg})(151.79\ \text{kJ/kg}) = 1.518\ \text{kJ}$$

Then the work output of this heat engine becomes

$$W_{\text{net,out}} = \eta_{\text{th}} Q_H = (0.15)(1.518\ \text{kJ}) = \mathbf{0.228\ kJ}$$

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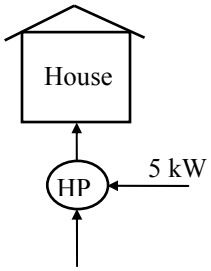
6-128 A heat pump with a specified COP and power consumption is used to heat a house. The time it takes for this heat pump to raise the temperature of a cold house to the desired level is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The heat loss of the house during the warp-up period is negligible. **3** The house is well-sealed so that no air leaks in or out.

Properties The constant volume specific heat of air at room temperature is $c_v = 0.718$ kJ/kg.°C.

Analysis Since the house is well-sealed (constant volume), the total amount of heat that needs to be supplied to the house is

$$Q_H = (mc_v \Delta T)_{\text{house}} = (1500 \text{ kg})(0.718 \text{ kJ/kg} \cdot °\text{C})(22 - 7)°\text{C} = 16{,}155 \text{ kJ}$$



The rate at which this heat pump supplies heat is

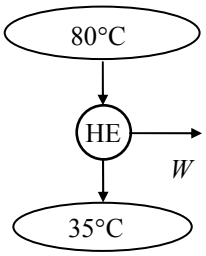
$$\dot{Q}_H = \text{COP}_{\text{HP}} \dot{W}_{\text{net,in}} = (2.8)(5 \text{ kW}) = 14 \text{ kW}$$

That is, this heat pump can supply 14 kJ of heat per second. Thus the time required to supply 16,155 kJ of heat is

$$\Delta t = \frac{Q_H}{\dot{Q}_H} = \frac{16{,}155 \text{ kJ}}{14 \text{ kJ/s}} = 1154 \text{ s} = \mathbf{19.2 \ min}$$

6-129 A solar pond power plant operates by absorbing heat from the hot region near the bottom, and rejecting waste heat to the cold region near the top. The maximum thermal efficiency that the power plant can have is to be determined.

Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from



$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{308 \text{ K}}{353 \text{ K}} = 0.127 \text{ or } \mathbf{12.7\%}$$

In reality, the temperature of the working fluid must be above 35°C in the condenser, and below 80°C in the boiler to allow for any effective heat transfer. Therefore, the maximum efficiency of the actual heat engine will be lower than the value calculated above.

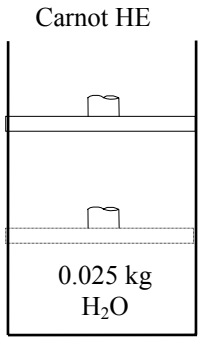
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6-130 A Carnot heat engine cycle is executed in a closed system with a fixed mass of steam. The net work output of the cycle and the ratio of sink and source temperatures are given. The low temperature in the cycle is to be determined.

Assumptions The engine is said to operate on the Carnot cycle, which is totally reversible.

Analysis The thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{1}{2} = 0.5$$



Also,

$$\eta_{\text{th}} = \frac{W}{Q_H} \longrightarrow Q_H = \frac{W}{\eta_{\text{th}}} = \frac{60\text{ kJ}}{0.5} = 120\text{ kJ}$$

$$Q_L = Q_H - W = 120 - 60 = 60\text{ kJ}$$

and

$$q_L = \frac{Q_L}{m} = \frac{60\text{ kJ}}{0.025\text{ kg}} = 2400\text{ kJ/kg} = h_{fg@T_L}$$

since the enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that T or P. Therefore, T_L is the temperature that corresponds to the h_{fg} value of 2400 kJ/kg, and is determined from the steam tables (Table A-4) to be

$$T_L = \mathbf{42.5°C}$$

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6-131 Problem 6-130 is reconsidered. The effect of the net work output on the required temperature of the steam during the heat rejection process as the work output varies from 40 kJ to 60 kJ is to be investigated.

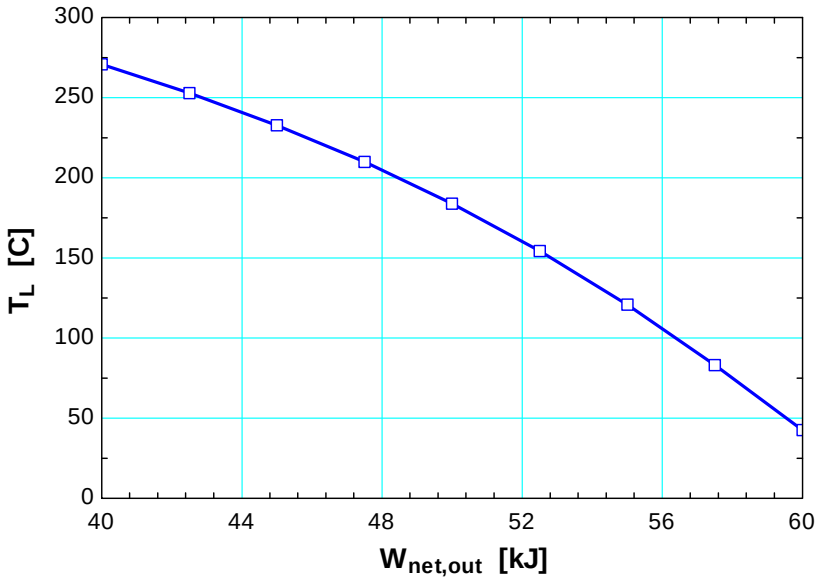
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Given"
m=0.025 [kg]
RatioT=0.5 "RatioT=T_L/T_H"
"W_net_out=60 [kJ]"

"Properties"
Fluid$= 'steam_iapws'
h_f=enthalpy(Fluid$, T=T_L, x=0)
h_g=enthalpy(Fluid$, T=T_L, x=1)
h_fg=h_g-h_f

"Analysis"
eta_th=1-RatioT
eta_th=W_net_out/Q_H
Q_L=Q_H-W_net_out
Q_L=m*h_fg
```

W_{out} [kJ]	$T_{L,C}$ [C]
40	270.8
42.5	252.9
45	232.8
47.5	209.9
50	184
52.5	154.4
55	120.8
57.5	83.17
60	42.5



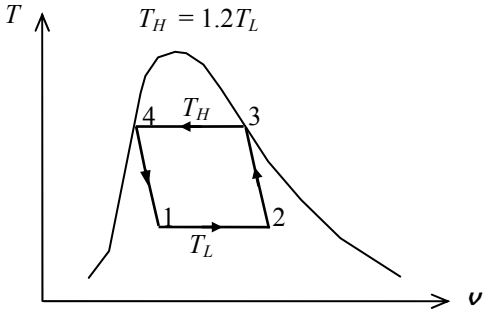
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6-132 A Carnot refrigeration cycle is executed in a closed system with a fixed mass of R-134a. The net work input and the ratio of maximum-to-minimum temperatures are given. The minimum pressure in the cycle is to be determined.

Assumptions The refrigerator is said to operate on the reversed Carnot cycle, which is totally reversible.

Analysis The coefficient of performance of the cycle is

$$\text{COP}_\text{R} = \frac{1}{T_H / T_L - 1} = \frac{1}{1.2 - 1} = 5$$



Also,

$$\text{COP}_\text{R} = \frac{Q_L}{W_\text{in}} \longrightarrow Q_L = \text{COP}_\text{R} \times W_\text{in} = (5)(22\ \text{kJ}) = 110\ \text{kJ}$$

$$Q_H = Q_L + W = 110 + 22 = 132\text{kJ}$$

and

$$q_H = \frac{Q_H}{m} = \frac{132\text{kJ}}{0.96\text{kg}} = 137.5\text{kJ/kg} = h_{fg@T_H}$$

since the enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that T or P. Therefore, T_H is the temperature that corresponds to the h_{fg} value of 137.5 kJ/kg, and is determined from the R-134a tables to be

$$T_H \cong 61.3°\text{C} = 334.3\ \text{K}$$

Then,

$$T_L = \frac{T_H}{1.2} = \frac{334.3\ \text{K}}{1.2} = 278.6\ \text{K} \cong 5.6°\text{C}$$

Therefore,

$$P_\text{min} = P_{\text{sat@5.6°C}} = \mathbf{355\ kPa}$$

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6-133 Problem 6-132 is reconsidered. The effect of the net work input on the minimum pressure as the work input varies from 10 kJ to 30 kJ is to be investigated. The minimum pressure in the refrigeration cycle is to be plotted as a function of net work input.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

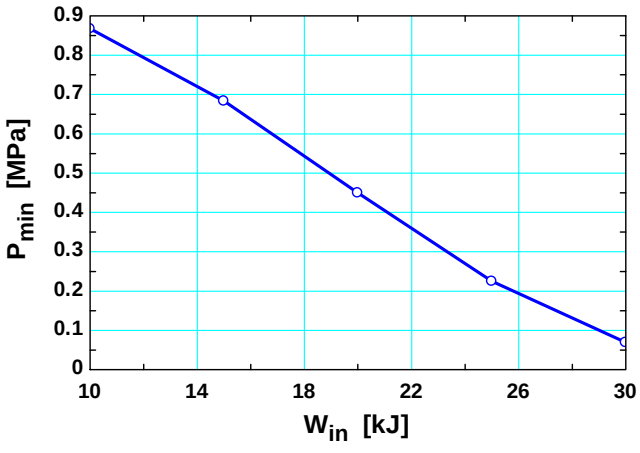
```
Analysis: The coefficient of performance of the cycle is given by"
m_R134a = 0.96 [kg]
THtoTLRatio = 1.2  "T_H = 1.2T_L"
"W_in = 22 [kJ]"  "Depending on the value of W_in, adjust the guess value of T_H."
COP_R = 1/( THtoTLRatio- 1)
Q_L = W_in*COP_R

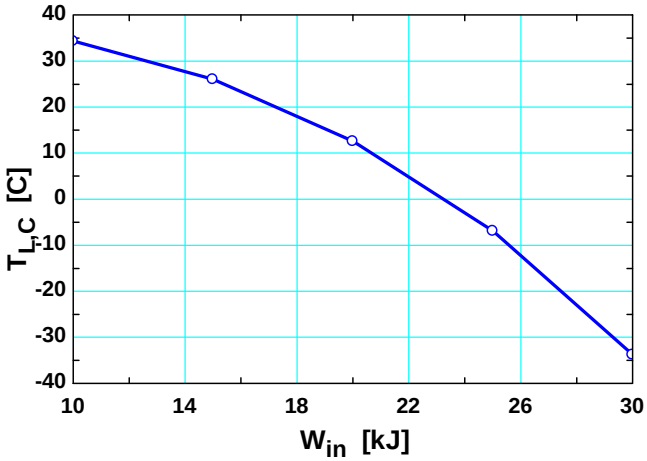
"First law applied to the refrigeration cycle yields:"
Q_L + W_in = Q_H

"Steady-flow analysis of the condenser yields
m_R134a*h_3 = m_R134a*h_4 +Q_H
Q_H = m_R134a*(h_3-h_4)  and h_fg = h_3 - h_4  also  T_H=T_3=T_4"
Q_H=m_R134a*h_fg
h_fg=enthalpy(R134a,T=T_H,x=1) - enthalpy(R134a,T=T_H,x=0)
T_H=THtoTLRatio*T_L

"The minimum pressure is the saturation pressure corresponding to T_L."
P_min = pressure(R134a,T=T_L,x=0)*convert(kPa,MPa)
T_L_C = T_L – 273
```

W_{in} [kJ]	P_{min} [MPa]	T_H [K]	T_L [K]	$T_{L,C}$ [C]
10	0.8673	368.8	307.3	34.32
15	0.6837	358.9	299	26.05
20	0.45	342.7	285.6	12.61
25	0.2251	319.3	266.1	-6.907
30	0.06978	287.1	239.2	-33.78

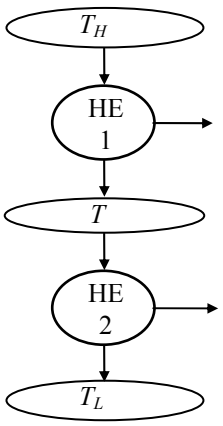




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6-134 Two Carnot heat engines operate in series between specified temperature limits. If the thermal efficiencies of both engines are the same, the temperature of the intermediate medium between the two engines is to be determined.

Assumptions The engines are said to operate on the Carnot cycle, which is totally reversible.



Analysis The thermal efficiency of the two Carnot heat engines can be expressed as

$$\eta_{\text{th,I}} = 1 - \frac{T}{T_H} \quad \text{and} \quad \eta_{\text{th,II}} = 1 - \frac{T_L}{T}$$

Equating,

$$1 - \frac{T}{T_H} = 1 - \frac{T_L}{T}$$

Solving for T,

$$T = \sqrt{T_H T_L} = \sqrt{(1800\ \text{K})(300\ \text{K})} = \mathbf{735\ K}$$

6-135E The thermal efficiency of a completely reversible heat engine as a function of the source temperature is to be calculated and plotted.

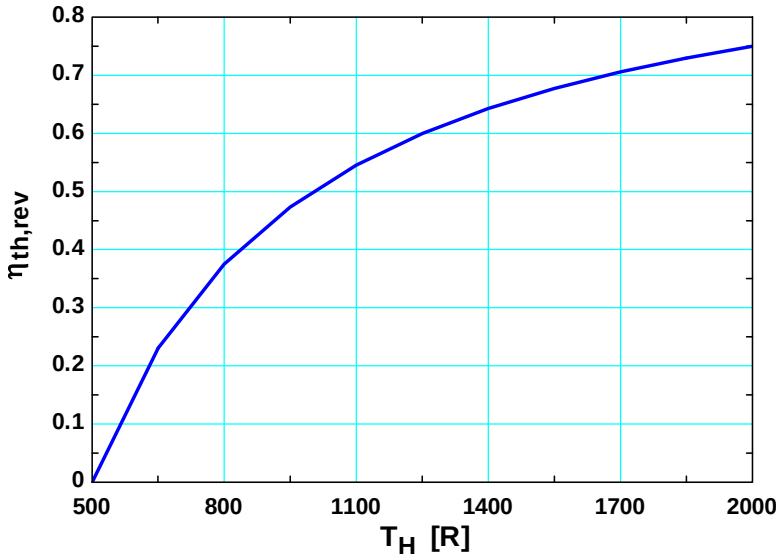
Assumptions The heat engine operates steadily.

Analysis With the specified sink temperature, the thermal efficiency of this completely reversible heat engine is

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H} = 1 - \frac{500\ \text{R}}{T_H}$$

Using EES, we tabulate and plot the variation of thermal efficiency with the source temperature:

T_H [R]	$\eta_{\text{th,rev}}$
500	0
650	0.2308
800	0.375
950	0.4737
1100	0.5455
1250	0.6
1400	0.6429
1550	0.6774
1700	0.7059
1850	0.7297
2000	0.75

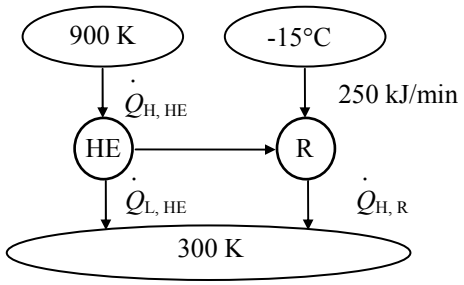


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6-136 A Carnot heat engine drives a Carnot refrigerator that removes heat from a cold medium at a specified rate. The rate of heat supply to the heat engine and the total rate of heat rejection to the environment are to be determined.

Analysis (*a*) The coefficient of performance of the Carnot refrigerator is

$$\text{COP}_{\text{R,C}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(300\ \text{K})/(258\ \text{K}) - 1} = 6.143$$



Then power input to the refrigerator becomes

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,C}}} = \frac{250\ \text{kJ/min}}{6.143} = 40.7\ \text{kJ/min}$$

which is equal to the power output of the heat engine, $\dot{W}_{\text{net,out}}$.

The thermal efficiency of the Carnot heat engine is determined from

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\ \text{K}}{900\ \text{K}} = 0.6667$$

Then the rate of heat input to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,\text{HE}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th,HE}}} = \frac{40.7\ \text{kJ/min}}{0.6667} = \mathbf{61.1\ kJ/min}$$

(*b*) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\text{R}}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 61.1 - 40.7 = 20.4\ \text{kJ/min}$$
$$\dot{Q}_{H,\text{R}} = \dot{Q}_{L,\text{R}} + \dot{W}_{\text{net,in}} = 250 + 40.7 = 290.7\ \text{kJ/min}$$

and

$$\dot{Q}_{\text{Ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,\text{R}} = 20.4 + 290.7 = \mathbf{311\ kJ/min}$$

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6-137 Problem 6-136 is reconsidered. The effects of the heat engine source temperature, the environment temperature, and the cooled space temperature on the required heat supply to the heat engine and the total rate of heat rejection to the environment as the source temperature varies from 500 K to 1000 K, the environment temperature varies from 275 K to 325 K, and the cooled space temperature varies from -20°C to 0°C are to be investigated. The required heat supply is to be plotted against the source temperature for the cooled space temperature of -15°C and environment temperatures of 275, 300, and 325 K.

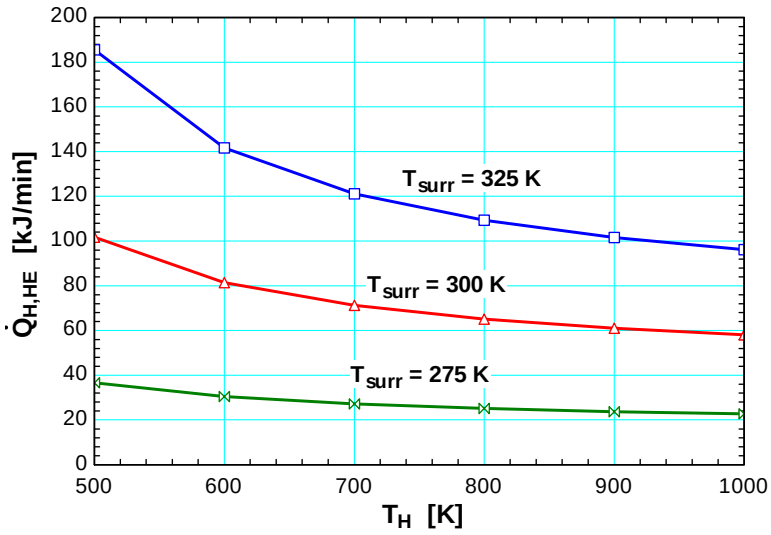
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
Q_dot_L_R = 250 [kJ/min]
T_surr = 300 [K]
T_H = 900 [K]
T_L_C = -15 [C]
T_L =T_L_C+ 273

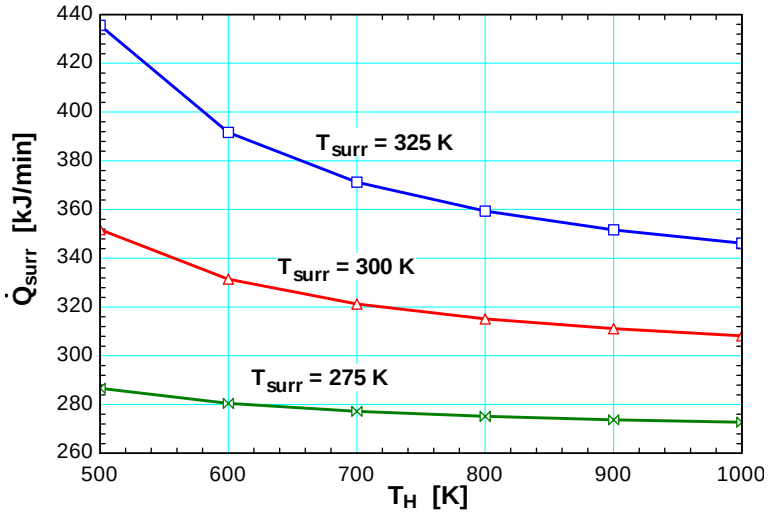
"Coefficient of performance of the Carnot refrigerator:"
T_H_R = T_surr
COP_R = 1/(T_H_R/T_L-1)
"Power input to the refrigerator:"
W_dot_in_R = Q_dot_L_R/COP_R
"Power output from heat engine must be:"
W_dot_out_HE =  W_dot_in_R
"The efficiency of the heat engine is:"
T_L_HE = T_surr
eta_HE = 1 - T_L_HE/T_H
"The rate of heat input to the heat engine is:"
Q_dot_H_HE = W_dot_out_HE/eta_HE
"First law applied to the heat engine and refrigerator:"
Q_dot_L_HE = Q_dot_H_HE - W_dot_out_HE
Q_dot_H_R = Q_dot_L_R + W_dot_in_R
```

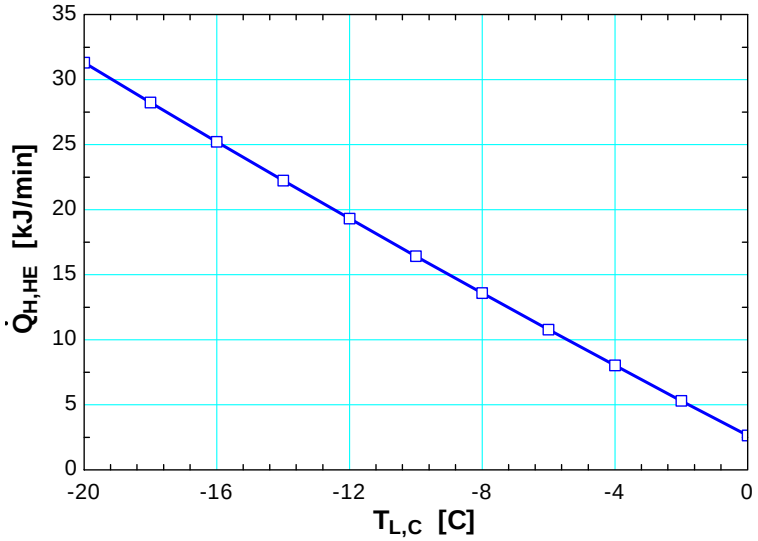
T_H [K]	Q_{HHE} [kJ/min]	Q_{surr} [kJ/min]
500	36.61	286.6
600	30.41	280.4
700	27.13	277.1
800	25.1	275.1
900	23.72	273.7
1000	22.72	272.7

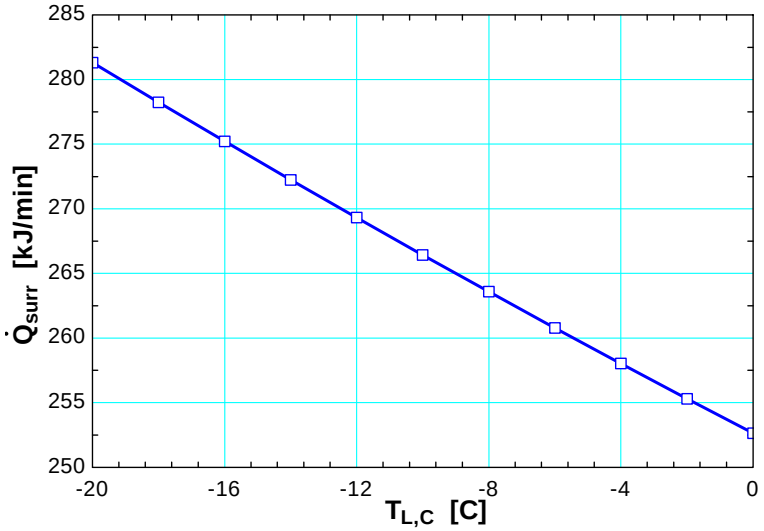
$T_{L,C}$ [C]	Q_{HHE} [kJ/min]	Q_{surr} [kJ/min]
-20	31.3	281.3
-18	28.24	278.2
-16	25.21	275.2
-14	22.24	272.2
-12	19.31	269.3
-10	16.43	266.4
-8	13.58	263.6
-6	10.79	260.8
-4	8.03	258
-2	5.314	255.3
0	2.637	252.6



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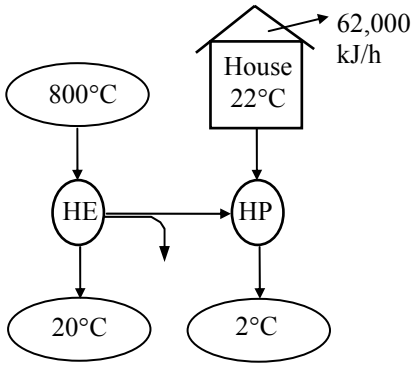
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6-138 Half of the work output of a Carnot heat engine is used to drive a Carnot heat pump that is heating a house. The minimum rate of heat supply to the heat engine is to be determined.

Assumptions Steady operating conditions exist.

Analysis The coefficient of performance of the Carnot heat pump is

$$\text{COP}_{\text{HP,C}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\text{ K})/(22+273\text{ K})} = 14.75$$



Then power input to the heat pump, which is supplying heat to the house at the same rate as the rate of heat loss, becomes

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,C}}} = \frac{62{,}000\text{ kJ/h}}{14.75} = 4203\text{ kJ/h}$$

which is half the power produced by the heat engine. Thus the power output of the heat engine is

$$\dot{W}_{\text{net,out}} = 2\dot{W}_{\text{net,in}} = 2(4203\text{ kJ/h}) = 8406\text{ kJ/h}$$

To minimize the rate of heat supply, we must use a Carnot heat engine whose thermal efficiency is determined from

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{293\text{ K}}{1073\text{ K}} = 0.727$$

Then the rate of heat supply to this heat engine is determined from the definition of thermal efficiency to be

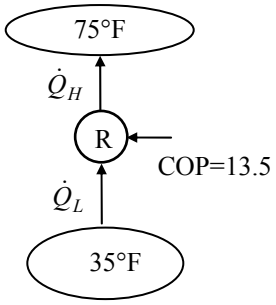
$$\dot{Q}_{H,\text{HE}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th,HE}}} = \frac{8406\text{ kJ/h}}{0.727} = \mathbf{11{,}560\ kJ/h}$$

6-139E An extraordinary claim made for the performance of a refrigerator is to be evaluated.

Assumptions Steady operating conditions exist.

Analysis The performance of this refrigerator can be evaluated by comparing it with a reversible refrigerator operating between the same temperature limits:

$$\text{COP}_{\text{R,max}} = \text{COP}_{\text{R,rev}} = \frac{1}{T_H/T_L - 1} = \frac{1}{(75+460)/(35+460) - 1} = 12.4$$



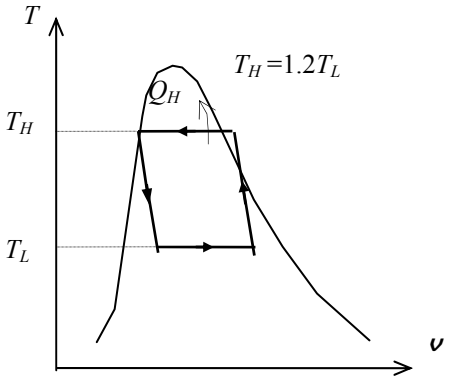
Discussion This is the highest COP a refrigerator can have when absorbing heat from a cool medium at 35°F and rejecting it to a warmer medium at 75°F. Since the COP claimed by the inventor is above this maximum value, **the claim is *false***.

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6-140 A Carnot heat pump cycle is executed in a steady-flow system with R-134a flowing at a specified rate. The net power input and the ratio of the maximum-to-minimum temperatures are given. The ratio of the maximum to minimum pressures is to be determined.

Analysis The coefficient of performance of the cycle is

$$\text{COP}_{\text{HP}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - 1/1.2} = 6.0$$



and

$$\dot{Q}_H = \text{COP}_{\text{HP}} \times \dot{W}_{\text{in}} = (6.0)(5\ \text{kW}) = 30.0\ \text{kJ/s}$$

$$q_H = \frac{\dot{Q}_H}{\dot{m}} = \frac{30.0\ \text{kJ/s}}{0.22\ \text{kg/s}} = 136.36\ \text{kJ/kg} = h_{fg@T_H}$$

since the enthalpy of vaporization h_{fg} at a given T or P represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that T or P. Therefore, T_H is the temperature that corresponds to the h_{fg} value of 136.36 kJ/kg, and is determined from the R-134a tables to be

$$T_H \cong 62.0°\text{C} = 335.1\ \text{K}$$

and

$$P_{\max} = P_{\text{sat@62.0°C}} = 1763\ \text{kPa}$$

$$T_L = \frac{T_H}{1.25} = \frac{335.1\ \text{K}}{1.2} = 291.4\ \text{K} \cong 18.3°\text{C}$$

Also,

$$P_{\min} = P_{\text{sat@18.3°C}} = 542\ \text{kPa}$$

Then the ratio of the maximum to minimum pressures in the cycle is

$$\frac{P_{\max}}{P_{\min}} = \frac{1763\ \text{kPa}}{542\ \text{kPa}} = \mathbf{3.25}$$

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6-141 Switching to energy efficient lighting reduces the electricity consumed for lighting as well as the cooling load in summer, but increases the heating load in winter. It is to be determined if switching to efficient lighting will increase or decrease the total energy cost of a building.

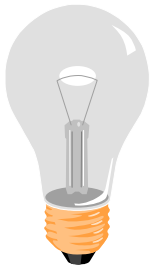
Assumptions The light escaping through the windows is negligible so that the entire lighting energy becomes part of the internal heat generation.

Analysis (*a*) Efficient lighting reduces the amount of electrical energy used for lighting year-around as well as the amount of heat generation in the house since light is eventually converted to heat. As a result, the electrical energy needed to air condition the house is also reduced. Therefore, in summer, the total cost of energy use of the household definitely decreases.

(*b*) In winter, the heating system must make up for the reduction in the heat generation due to reduced energy used for lighting. The total cost of energy used in this case will still decrease if the cost of unit heat energy supplied by the heating system is less than the cost of unit energy provided by lighting.

The cost of 1 kWh heat supplied from lighting is $0.08 since all the energy consumed by lamps is eventually converted to thermal energy. Noting that 1 therm = 105,500 kJ = 29.3 kWh and the furnace is 80% efficient, the cost of 1 kWh heat supplied by the heater is

$$\begin{aligned}\text{Cost of 1 kWh heat supplied by furnace} &= (\text{Amount of useful energy}/\eta_{\text{furnace}})(\text{Price}) \\ &= [(1\,\text{kWh})/0.80](\$1.40/\text{therm})\left(\frac{1\,\text{therm}}{29.3\,\text{kWh}}\right) \\ &= \$0.060 \ \ (\text{per kWh heat})\end{aligned}$$



which is less than $0.08. Thus we conclude that switching to energy efficient lighting will **reduce** the total energy cost of this building both in summer and in winter.

Discussion To determine the amount of cost savings due to switching to energy efficient lighting, consider 10 h of operation of lighting in summer and in winter for 1 kW rated power for lighting.

Current lighting:

Lighting cost: (Energy used)(Unit cost)= (1 kW)(10 h)($0.08/kWh) = $0.80

Increase in air conditioning cost: (Heat from lighting/COP)(unit cost) =(10 kWh/3.5)($0.08/kWh) = $0.23

Decrease in the heating cost = [Heat from lighting/Eff](unit cost)=(10/0.8 kWh)($1.40/29.3/kWh) =$0.60

Total cost in summer = 0.80+0.23 = $1.03; Total cost in winter = $0.80-0.60 = 0.20.

Energy efficient lighting:

Lighting cost: (Energy used)(Unit cost)= (0.25 kW)(10 h)($0.08/kWh) = $0.20

Increase in air conditioning cost: (Heat from lighting/COP)(unit cost) =(2.5 kWh/3.5)($0.08/kWh) = $0.06

Decrease in the heating cost = [Heat from lighting/Eff](unit cost)=(2.5/0.8 kWh)($1.40/29.3/kWh) = $0.15

Total cost in summer = 0.20+0.06 = $0.26; Total cost in winter = $0.20-0.15 = 0.05.

Note that during a day with 10 h of operation, the total energy cost decreases from $1.03 to $0.26 in summer, and from $0.20 to $0.05 in winter when efficient lighting is used.

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6-142 A heat pump is used to heat a house. The maximum money saved by using the lake water instead of outside air as the heat source is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The kinetic and potential energy changes are zero.

Analysis When outside air is used as the heat source, the cost of energy is calculated considering a reversible heat pump as follows:

$$\text{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (0+273)/(25+273)} = 11.92$$

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{(140{,}000/3600)\,\text{kW}}{11.92} = 3.262\,\text{kW}$$

$$\text{Cost}_{\text{air}} = (3.262\,\text{kW})(100\,\text{h})(\$0.085/\text{kWh}) = \$27.73$$

Repeating calculations for lake water,

$$\text{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (10+273)/(25+273)} = 19.87$$

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{(140{,}000/3600)\,\text{kW}}{19.87} = 1.957\,\text{kW}$$

$$\text{Cost}_{\text{lake}} = (1.957\,\text{kW})(100\,\text{h})(\$0.085/\text{kWh}) = \$16.63$$

Then the money saved becomes

$$\text{Money Saved} = \text{Cost}_{\text{air}} - \text{Cost}_{\text{lake}} = \$27.73 - \$16.63 = \mathbf{\$11.10}$$

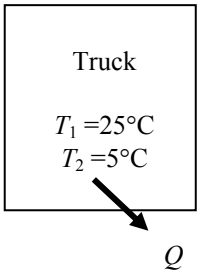
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6-143 The cargo space of a refrigerated truck is to be cooled from 25°C to an average temperature of 5°C. The time it will take for an 8-kW refrigeration system to precool the truck is to be determined.

Assumptions **1** The ambient conditions remain constant during precooling. **2** The doors of the truck are tightly closed so that the infiltration heat gain is negligible. **3** The air inside is sufficiently dry so that the latent heat load on the refrigeration system is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The density of air is taken 1.2 kg/m^3, and its specific heat at the average temperature of 15°C is c_p = 1.0 kJ/kg·°C (Table A-2).

Analysis The mass of air in the truck is



$$m_{\text{air}} = \rho_{\text{air}} V_{\text{truck}} = (1.2 \text{ kg/m}^3)(12 \text{ m} \times 2.3 \text{ m} \times 3.5 \text{ m}) = 116 \text{ kg}$$

The amount of heat removed as the air is cooled from 25 to 5ºC

$$Q_{\text{cooling,air}} = (mc_p \Delta T)_{\text{air}} = (116 \text{ kg})(1.0 \text{ kJ/kg.°C})(25 - 5)°\text{C}$$
$$= 2{,}320 \text{ kJ}$$

Noting that UA is given to be 80 W/ºC and the average air temperature in the truck during precooling is (25+5)/2 = 15ºC, the average rate of heat gain by transmission is determined to be

$$\dot{Q}_{\text{transmission,ave}} = UA\Delta T = (80 \text{ W/}^\circ\text{C})(25 - 15)^\circ\text{C} = 800 \text{ W} = 0.80 \text{ kJ/s}$$

Therefore, the time required to cool the truck from 25 to 5ºC is determined to be

$$\dot{Q}_{\text{refrig.}} \Delta t = Q_{\text{cooling,air}} + \dot{Q}_{\text{transmission}} \Delta t$$

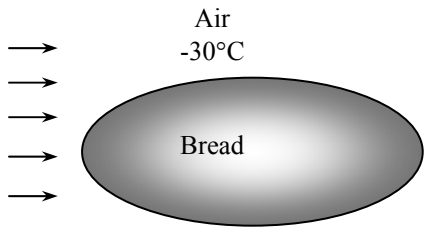
$$\longrightarrow \Delta t = \frac{Q_{\text{cooling,air}}}{\dot{Q}_{\text{refrig.}} - \dot{Q}_{\text{transmission}}} = \frac{2{,}320 \text{kJ}}{(8 - 0.8) \text{ kJ/s}} = 322\text{s} \cong \mathbf{5.4min}$$

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6-144 A refrigeration system is to cool bread loaves at a rate of 1200 per hour by refrigerated air at -30°C. The rate of heat removal from the breads, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the bread loaves are constant. **3** The cooling section is well-insulated so that heat gain through its walls is negligible.

Properties The average specific and latent heats of bread are given to be 2.93 kJ/kg.°C and 109.3 kJ/kg, respectively. The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1), and the specific heat of air at the average temperature of (-30 + -22)/2 = -26°C ≈ 250 K is c_p =1.0 kJ/kg.°C (Table A-2).



Analysis (*a*) Noting that the breads are cooled at a rate of 500 loaves per hour, breads can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{bread}} = (1200\ \text{breads/h})(0.350\ \text{kg/bread}) = 420\ \text{kg/h} = 0.1167\ \text{kg/s}$$

Then the rate of heat removal from the breads as they are cooled from 30°C to -10ºC and frozen becomes

$$\dot{Q}_{\text{bread}} = (\dot{m}c_p\Delta T)_{\text{bread}} = (420\ \text{kg/h})(2.93\ \text{kJ/kg.°C})[(30-(-10)]\text{°C} = 49{,}224\ \text{kJ/h}$$

$$\dot{Q}_{\text{freezing}} = (\dot{m}h_{\text{latent}})_{\text{bread}} = (420\ \text{kg/h})(109.3\,\text{kJ/kg}) = 45{,}906\ \text{kJ/h}$$

and

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{bread}} + \dot{Q}_{\text{freezing}} = 49{,}224 + 45{,}906 = \mathbf{95{,}130\ kJ/h}$$

(*b*) All the heat released by the breads is absorbed by the refrigerated air, and the temperature rise of air is not to exceed 8°C. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p\Delta T)_{\text{air}}} = \frac{95{,}130\ \text{kJ/h}}{(1.0\ \text{kJ/kg.°C})(8\text{°C})} = 11{,}891\ \text{kg/h}$$

$$\rho = \frac{P}{RT} = \frac{101.3\ \text{kPa}}{(0.287\ \text{kPa.m}^3\text{/kg.K})(-30+273)\ \text{K}} = 1.453\ \text{kg/m}^3$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{11{,}891\ \text{kg/h}}{1.453\ \text{kg/m}^3} = \mathbf{8185\ m^3/h}$$

(*c*) For a COP of 1.2, the size of the compressor of the refrigeration system must be

$$\dot{W}_{\text{refrig}} = \frac{\dot{Q}_{\text{refrig}}}{\text{COP}} = \frac{95{,}130\ \text{kJ/h}}{1.2} = 79{,}275\ \text{kJ/h} = \mathbf{22.02\ kW}$$

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6-145 The drinking water needs of a production facility with 20 employees is to be met by a bobbler type water fountain. The size of compressor of the refrigeration system of this water cooler is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The cold water requirement is 0.4 L/h per person.

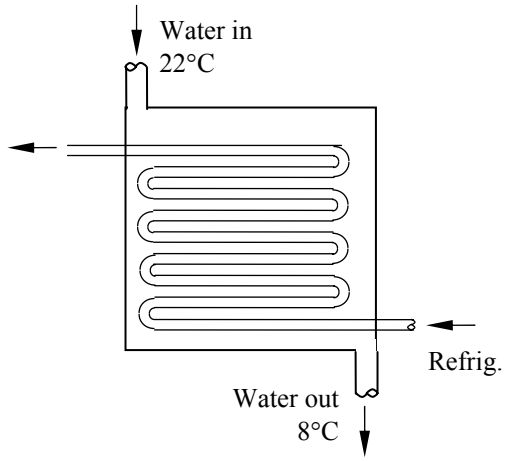
Properties The density and specific heat of water at room temperature are $\rho = 1.0$ kg/L and $c = 4.18$ kJ/kg.°C.C (Table A-3).

Analysis The refrigeration load in this case consists of the heat gain of the reservoir and the cooling of the incoming water. The water fountain must be able to provide water at a rate of

$$\dot{m}_{\text{water}} = \rho \dot{V}_{\text{water}} = (1\,\text{kg/L})(0.4\,\text{L/h}\cdot\text{person})(20\,\text{persons}) = 8.0\,\text{kg/h}$$

To cool this water from 22°C to 8°C, heat must removed from the water at a rate of

$$\begin{aligned}\dot{Q}_{\text{cooling}} &= \dot{m}c_{\text{p}}(T_{\text{in}} - T_{\text{out}}) \\ &= (8.0\,\text{kg/h})(4.18\,\text{kJ/kg.°C})(22 - 8)°\text{C} \\ &= 468\,\text{kJ/h} = 130\,\text{W} \quad (\text{since } 1\,\text{W} = 3.6\,\text{kJ/h})\end{aligned}$$



Then total refrigeration load becomes

$$\dot{Q}_{\text{refrig, total}} = \dot{Q}_{\text{cooling}} + \dot{Q}_{\text{transfer}} = 130 + 45 = 175\,\text{W}$$

Noting that the coefficient of performance of the refrigeration system is 2.9, the required power input is

$$\dot{W}_{\text{refrig}} = \frac{\dot{Q}_{\text{refrig}}}{\text{COP}} = \frac{175\,\text{W}}{2.9} = \mathbf{60.3\ W}$$

Therefore, the power rating of the compressor of this refrigeration system must be at least 60.3 W to meet the cold water requirements of this office.

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6-146E A washing machine uses $33/year worth of hot water heated by a gas water heater. The amount of hot water an average family uses per week is to be determined.

Assumptions **1** The electricity consumed by the motor of the washer is negligible. **2** Water is an incompressible substance with constant properties at room temperature.

Properties The density and specific heat of water at room temperature are $\rho = 62.1$ lbm/ft^3 and $c = 1.00$ Btu/lbm.°F (Table A-3E).

Analysis The amount of electricity used to heat the water and the net amount transferred to water are

$$\text{Total energy used (gas)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$33/\text{year}}{\$1.21/\text{therm}} = 27.27 \text{ therms/year}$$

$$\text{Total energy transfer to water} = \dot{E}_{\text{in}} = (\text{Efficiency})(\text{Total energy used}) = 0.58 \times 27.27 \text{ therms/year}$$

$$= 15.82 \text{ therms/year} = (15.82 \text{ therms/year})\left(\frac{100{,}000 \text{ Btu}}{1 \text{ therm}}\right)\left(\frac{1 \text{ year}}{52 \text{ weeks}}\right)$$

$$= 30{,}420 \text{ Btu/week}$$

Then the mass and the volume of hot water used per week become

$$\dot{E}_{\text{in}} = \dot{m}c(T_{\text{out}} - T_{\text{in}}) \;\rightarrow\; \dot{m} = \frac{\dot{E}_{\text{in}}}{c(T_{\text{out}} - T_{\text{in}})} = \frac{30{,}420 \text{ Btu/week}}{(1.0 \text{ Btu/lbm.°F})(130 - 60)°\text{F}} = 434.6 \text{ lbm/week}$$

and

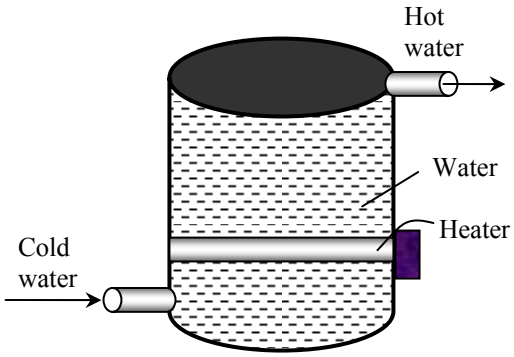
$$\dot{V}_{\text{water}} = \frac{\dot{m}}{\rho} = \frac{434.6 \text{ lbm/week}}{62.1 \text{ lbm/ft}^3} = (7.0 \text{ ft}^3/\text{week})\left(\frac{7.4804 \text{ gal}}{1 \text{ ft}^3}\right) = \mathbf{52.4 \ gal/week}$$

Therefore, an average family uses about 52 gallons of hot water per week for washing clothes.

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6-147 A typical heat pump powered water heater costs about $800 more to install than a typical electric water heater. The number of years it will take for the heat pump water heater to pay for its cost differential from the energy it saves is to be determined.

Assumptions **1** The price of electricity remains constant. **2** Water is an incompressible substance with constant properties at room temperature. **3** Time value of money (interest, inflation) is not considered.



Analysis The amount of electricity used to heat the water and the net amount transferred to water are

$$\text{Total energy used (electrical)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$250/\text{year}}{\$0.080/\text{kWh}} = 3125\ \text{kWh/year}$$

$$\text{Total energy transfer to water} = \dot{E}_{\text{in}} = (\text{Efficiency})(\text{Total energy used}) = 0.95 \times 3125\ \text{kWh/year} = 2969\ \text{kWh/year}$$

The amount of electricity consumed by the heat pump and its cost are

$$\text{Energy usage (of heat pump)} = \frac{\text{Energy transfer to water}}{\text{COP}_{\text{HP}}} = \frac{2969\ \text{kWh/year}}{3.3} = 899.6\ \text{kWh/year}$$

$$\text{Energy cost (of heat pump)} = (\text{Energy usage})(\text{Unit cost of energy}) = (899.6\ \text{kWh/year})(\$0.08/\text{kWh}) = \$71.97/\text{year}$$

Then the money saved per year by the heat pump and the simple payback period become

$$\text{Money saved} = (\text{Energy cost of electric heater}) - (\text{Energy cost of heat pump}) = \$250 - \$71.97 = \$178.0$$

$$\text{Simple payback period} = \frac{\text{Additional installation cost}}{\text{Money saved}} = \frac{\$800}{\$178.0/\text{year}} = \mathbf{4.49\ years}$$

Discussion The economics of heat pump water heater will be even better if the air in the house is used as the heat source for the heat pump in summer, and thus also serving as an air-conditioner.

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6-148 Problem 6-147 is reconsidered. The effect of the heat pump COP on the yearly operation costs and the number of years required to break even are to be considered.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Energy supplied by the water heater to the water per year is E_ElecHeater"

"Cost per year to operate electric water heater for one year is:"
Cost_ElectHeater = 250 [$/year]

"Energy supplied to the water by electric heater is 90% of energy purchased"
eta=0.95
E_ElectHeater = eta*Cost_ElectHeater /UnitCost  "[kWh/year]"
UnitCost=0.08 [$/kWh]

"For the same amont of heated water and assuming that all the heat energy leaving the heat pump goes into the water, then"
"Energy supplied by heat pump heater = Energy supplied by electric heater"
E_HeatPump = E_ElectHeater "[kWh/year]"

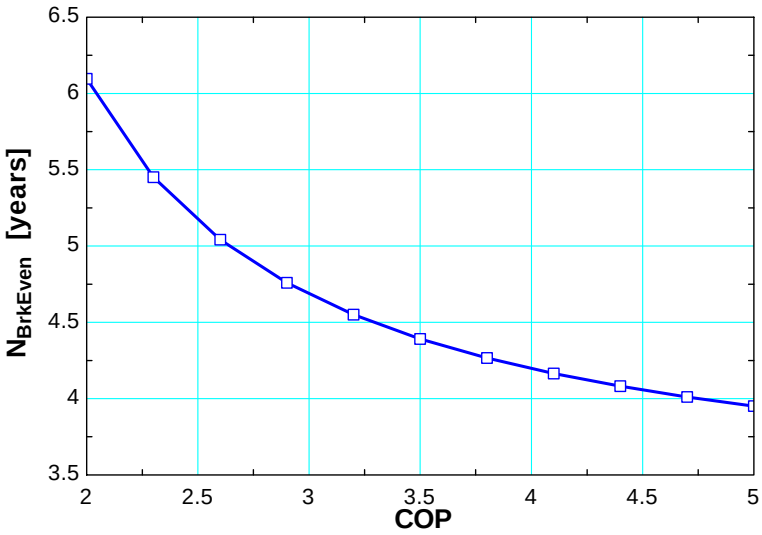
"Electrical Work enegy supplied to heat pump = Heat added to water/COP"
COP=3.3
W_HeatPump = E_HeatPump/COP "[kWh/year]"

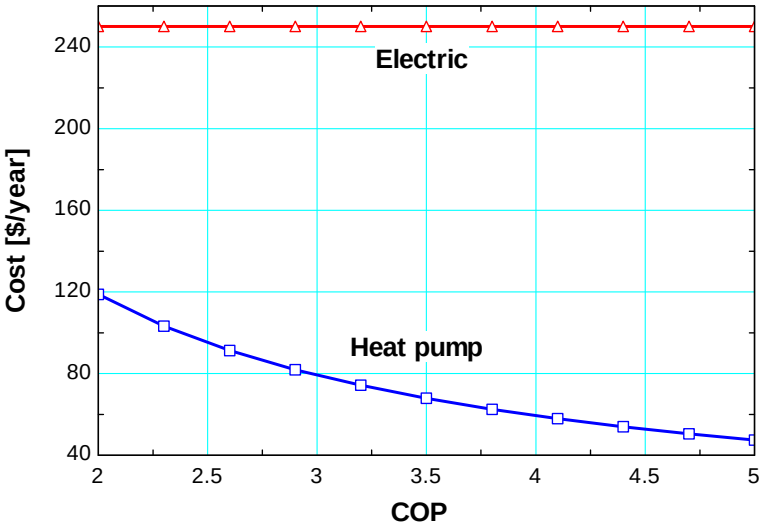
"Cost per year to operate the heat pump is"
Cost_HeatPump=W_HeatPump*UnitCost

"Let N_BrkEven be the number of years to break even:"
"At the break even point, the total cost difference between the two water heaters is zero."
"Years to break even, neglecting the cost to borrow the extra $800 to install heat pump"
CostDiff_total = 0 [$]
CostDiff_total=AddCost+N_BrkEven*(Cost_HeatPump-Cost_ElectHeater) "[$]"
AddCost=800 [$]
```

COP	$B_{BrkEven}$ [years]	$Cost_{HeatPump}$ [$/year]	$Cost_{ElektHeater}$ [$/year]
2	6.095	118.8	250
2.3	5.452	103.3	250
2.6	5.042	91.35	250
2.9	4.759	81.9	250
3.2	4.551	74.22	250
3.5	4.392	67.86	250
3.8	4.267	62.5	250
4.1	4.165	57.93	250
4.4	4.081	53.98	250
4.7	4.011	50.53	250
5	3.951	47.5	250

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6-149 A home owner is to choose between a high-efficiency natural gas furnace and a ground-source heat pump. The system with the lower energy cost is to be determined.

Assumptions The two heater are comparable in all aspects other than the cost of energy.

Analysis The unit cost of each kJ of useful energy supplied to the house by each system is

Natural gas furnace: $$\text{Unit cost of useful energy} = \frac{(\$1.42/\text{therm})}{0.97}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = \$13.8\times10^{-6}/\text{kJ}$$

Heat Pump System: $$\text{Unit cost of useful energy} = \frac{(\$0.092/\text{kWh})}{3.5}\left(\frac{1\ \text{kWh}}{3600\ \text{kJ}}\right) = \$7.3\times10^{-6}/\text{kJ}$$

The energy cost of **ground-source heat pump system** will be lower.

6-150 The ventilating fans of a house discharge a houseful of warmed air in one hour (ACH = 1). For an average outdoor temperature of 5°C during the heating season, the cost of energy "vented out" by the fans in 1 h is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The house is maintained at 22°C and 92 kPa at all times. **3** The infiltrating air is heated to 22°C before it is vented out. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The volume occupied by the people, furniture, etc. is negligible.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg·K (Table A-1). The specific heat of air at room temperature is c_p = 1.0 kJ/kg·°C (Table A-2a).

Analysis The density of air at the indoor conditions of 92 kPa and 22°C is

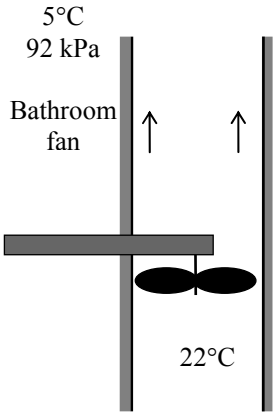
$$\rho_o = \frac{P_o}{RT_o} = \frac{92\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(22+273\ \text{K})} = 1.087\ \text{kg/m}^3$$

Noting that the interior volume of the house is 200 × 2.8 = 560 m^3, the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho\dot{V}_{\text{air}} = (1.087\ \text{kg/m}^3)(560\ \text{m}^3/\text{h}) = 608.7\ \text{kg/h} = 0.169\ \text{kg/s}$$

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 5°C, this corresponds to energy loss at a rate of

$$\dot{Q}_{\text{loss,fan}} = \dot{m}_{\text{air}} c_p (T_{\text{indoors}} - T_{\text{outdoors}})$$
$$= (0.169\ \text{kg/s})(1.0\ \text{kJ/kg.°C})(22-5)°\text{C} = 2.874\ \text{kJ/s} = 2.874\ \text{kW}$$



Then the amount and cost of the heat "vented out" per hour becomes

$$\text{Fuel energy loss} = \dot{Q}_{\text{loss,fan}}\Delta t/\eta_{\text{furnace}} = (2.874\ \text{kW})(1\ \text{h})/0.96 = 2.994\ \text{kWh}$$
$$\text{Money loss} = (\text{Fuel energy loss})(\text{Unit cost of energy})$$
$$= (2.994\ \text{kWh})(\$1.20/\text{therm})\left(\frac{1\ \text{therm}}{29.3\ \text{kWh}}\right) = \mathbf{\$0.123}$$

Discussion Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used sparingly.

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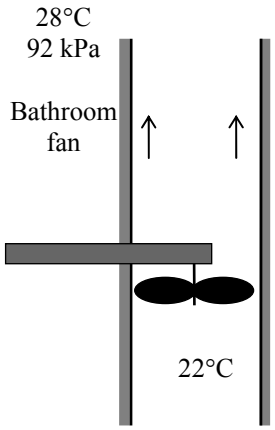
6-151 The ventilating fans of a house discharge a houseful of air-conditioned air in one hour (ACH = 1). For an average outdoor temperature of 28°C during the cooling season, the cost of energy "vented out" by the fans in 1 h is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The house is maintained at 22°C and 92 kPa at all times. **3** The infiltrating air is cooled to 22°C before it is vented out. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The volume occupied by the people, furniture, etc. is negligible. **6** Latent heat load is negligible.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg·K (Table A-1). The specific heat of air at room temperature is $c_p = 1.0$ kJ/kg·°C (Table A-2a).

Analysis The density of air at the indoor conditions of 92 kPa and 22°C is

$$\rho_o = \frac{P_o}{RT_o} = \frac{92\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(22+273\ \text{K})} = 1.087\ \text{kg/m}^3$$



Noting that the interior volume of the house is 200 × 2.8 = 560 m^3, the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (1.087\ \text{kg/m}^3)(560\ \text{m}^3/\text{h}) = 608.7\ \text{kg/h} = 0.169\ \text{kg/s}$$

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 28°C, this corresponds to energy loss at a rate of

$$\begin{aligned}\dot{Q}_{\text{loss,fan}} &= \dot{m}_{\text{air}} c_p (T_{\text{outdoors}} - T_{\text{indoors}}) \\ &= (0.169\ \text{kg/s})(1.0\ \text{kJ/kg.°C})(28-22)°\text{C} = 1.014\ \text{kJ/s} = 1.014\ \text{kW}\end{aligned}$$

Then the amount and cost of the electric energy "vented out" per hour becomes

$$\text{Electric energy loss} = \dot{Q}_{\text{loss,fan}} \Delta t / COP = (1.014\ \text{kW})(1\ \text{h})/2.3 = 0.441\ \text{kWh}$$

$$\begin{aligned}\text{Money loss} &= (\text{Fuel energy loss})(\text{Unit cost of energy}) \\ &= (0.441\ \text{kWh})(\$0.10/\text{kWh}) = \mathbf{\$0.044}\end{aligned}$$

Discussion Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used sparingly.

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6-152 A geothermal heat pump with R-134a as the working fluid is considered. The evaporator inlet and exit states are specified. The mass flow rate of the refrigerant, the heating load, the COP, and the minimum power input to the compressor are to be determined.

Assumptions **1** The heat pump operates steadily. **2** The kinetic and potential energy changes are zero. **3** Steam properties are used for geothermal water.

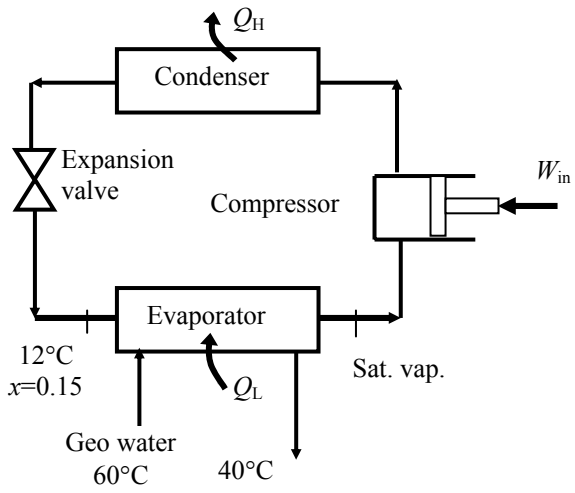
Properties The properties of R-134a and water are (Steam and R-134a tables)

$$\left.\begin{matrix} T_1 = 12°\text{C} \\ x_1 = 0.15 \end{matrix}\right\} \begin{matrix} h_1 = 96.55 \text{ kJ/kg} \\ P_1 = 443.3 \text{ kPa} \end{matrix}$$

$$\left.\begin{matrix} P_2 = P_1 = 443.3 \text{ kPa} \\ x_2 = 1 \end{matrix}\right\} h_2 = 257.27 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_{w,1} = 60°\text{C} \\ x_{w,1} = 0 \end{matrix}\right\} h_{w,1} = 251.18 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_{w,2} = 40°\text{C} \\ x_{w,2} = 0 \end{matrix}\right\} h_{w,2} = 167.53 \text{ kJ/kg}$$



Analysis (*a*) The rate of heat transferred from the water is the energy change of the water from inlet to exit

$$\dot{Q}_L = \dot{m}_w (h_{w,1} - h_{w,2}) = (0.065 \text{ kg/s})(251.18 - 167.53) \text{ kJ/kg} = 5.437 \text{ kW}$$

The energy increase of the refrigerant is equal to the energy decrease of the water in the evaporator. That is,

$$\dot{Q}_L = \dot{m}_R (h_2 - h_1) \longrightarrow \dot{m}_R = \frac{\dot{Q}_L}{h_2 - h_1} = \frac{5.437 \text{ kW}}{(257.27 - 96.55) \text{ kJ/kg}} = \mathbf{0.0338 \text{ kg/s}}$$

(*b*) The heating load is

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 5.437 + 1.6 = \mathbf{7.04 \text{ kW}}$$

(*c*) The COP of the heat pump is determined from its definition,

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{7.04 \text{ kW}}{1.6 \text{ kW}} = \mathbf{4.40}$$

(*d*) The COP of a reversible heat pump operating between the same temperature limits is

$$\text{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (25 + 273)/(60 + 273)} = 9.51$$

Then, the minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{7.04 \text{ kW}}{9.51} = \mathbf{0.740 \text{ kW}}$$

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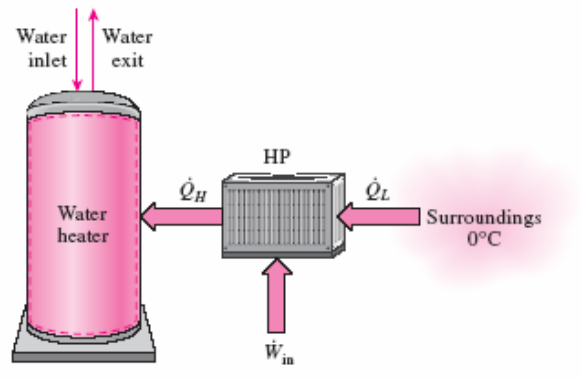
6-153 A heat pump is used as the heat source for a water heater. The rate of heat supplied to the water and the minimum power supplied to the heat pump are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The kinetic and potential energy changes are zero.

Properties The specific heat and specific volume of water at room temperature are c_p = 4.18 kJ/kg.K and v=0.001 m^3/kg (Table A-3).

Analysis (*a*) An energy balance on the water heater gives the rate of heat supplied to the water

$$\begin{aligned}\dot{Q}_H &= \dot{m}c_p(T_2 - T_1)\\ &= \frac{\dot{V}}{v}c_p(T_2 - T_1)\\ &= \frac{(0.02/60)\,\text{m}^3/\text{s}}{0.001\,\text{m}^3/\text{kg}}(4.18\,\text{kJ/kg.°C})(50-10)\,°\text{C}\\ &= \mathbf{55.73\,kW}\end{aligned}$$



(*b*) The COP of a reversible heat pump operating between the specified temperature limits is

$$\text{COP}_{\max} = \frac{1}{1 - T_L/T_H} = \frac{1}{1-(0+273)/(30+273)} = 10.1$$

Then, the minimum power input would be

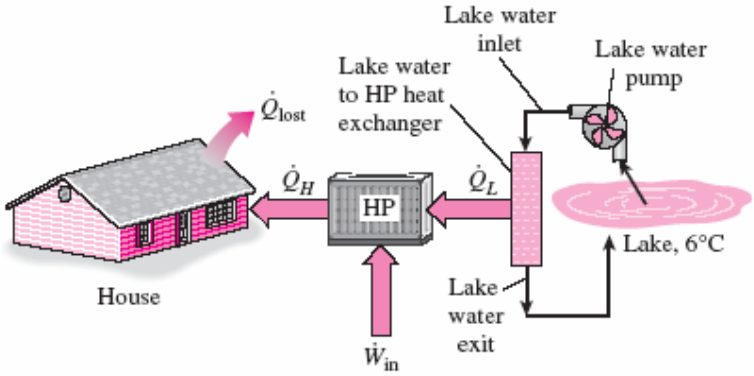
$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{55.73\,\text{kW}}{10.1} = \mathbf{5.52\,kW}$$

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6-154 A heat pump receiving heat from a lake is used to heat a house. The minimum power supplied to the heat pump and the mass flow rate of lake water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The kinetic and potential energy changes are zero.

Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg.K (Table A-3).



Analysis (*a*) The COP of a reversible heat pump operating between the specified temperature limits is

$$\text{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (6+273)/(27+273)} = 14.29$$

Then, the minimum power input would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{(64{,}000/3600)\,\text{kW}}{14.29} = \mathbf{1.244\,kW}$$

(*b*) The rate of heat absorbed from the lake is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in,min}} = 17.78 - 1.244 = 16.53\,\text{kW}$$

An energy balance on the heat exchanger gives the mass flow rate of lake water

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_L}{c_p \Delta T} = \frac{16.53\,\text{kJ/s}}{(4.18\,\text{kJ/kg.°C})(5\,°\text{C})} = \mathbf{0.791\,kg/s}$$

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6-155 It is to be proven that a refrigerator's COP cannot exceed that of a completely reversible refrigerator that shares the same thermal-energy reservoirs.

Assumptions The refrigerator operates steadily.

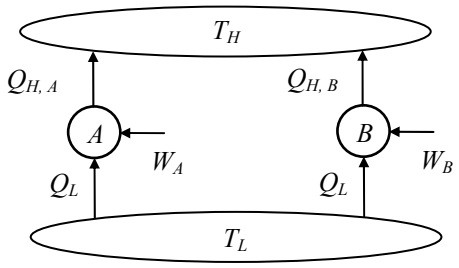
Analysis We begin by assuming that the COP of the general refrigerator B is greater than that of the completely reversible refrigerator A, $\text{COP}_B > \text{COP}_A$. When this is the case, a rearrangement of the coefficient of performance expression yields

$$W_B = \frac{Q_L}{\text{COP}_B} < \frac{Q_L}{\text{COP}_A} = W_A$$

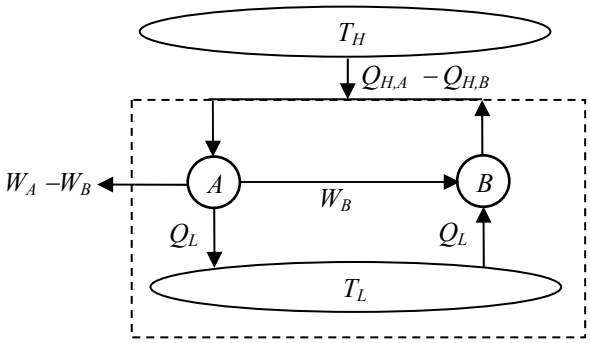
That is, the magnitude of the work required to drive refrigerator B is less than that needed to drive completely reversible refrigerator A. Applying the first law to both refrigerators yields

$$Q_{H,B} < Q_{H,A}$$

since the work supplied to refrigerator B is less than that supplied to refrigerator A, and both have the same cooling effect, Q_L.



Since A is a completely reversible refrigerator, we can reverse it without changing the magnitude of the heat and work transfers. This is illustrated in the figure below. The heat, Q_L , which is rejected by the reversed refrigerator A can now be routed directly to refrigerator B. The net effect when this is done is that no heat is exchanged with the T_L reservoir. The magnitude of the heat supplied to the reversed refrigerator A, $Q_{H,A}$ has been shown to be larger than that rejected by refrigerator B. There is then a net heat transfer from the T_H reservoir to the combined device in the dashed lines of the figure whose magnitude is given by $Q_{H,A} - Q_{H,B}$. Similarly, there is a net work production by the combined device whose magnitude is given by $W_A - W_B$.



The combined cyclic device then exchanges heat with a reservoir at a single temperature and produces work which is clearly a violation of the Kelvin-Planck statement of the second law. Our assumption the $\text{COP}_B > \text{COP}_A$ must then be wrong.

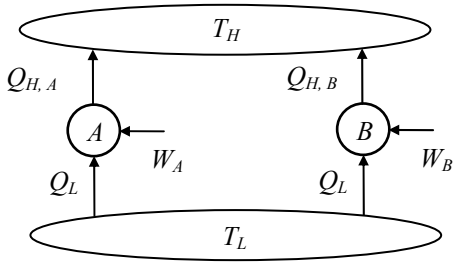
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6-156 It is to be proven that the COP of all completely reversible refrigerators must be the same when the reservoir temperatures are the same.

Assumptions The refrigerators operate steadily.

Analysis We begin by assuming that $\text{COP}_A < \text{COP}_B$. When this is the case, a rearrangement of the coefficient of performance expression yields

$$W_A = \frac{Q_L}{\text{COP}_A} > \frac{Q_L}{\text{COP}_B} = W_B$$

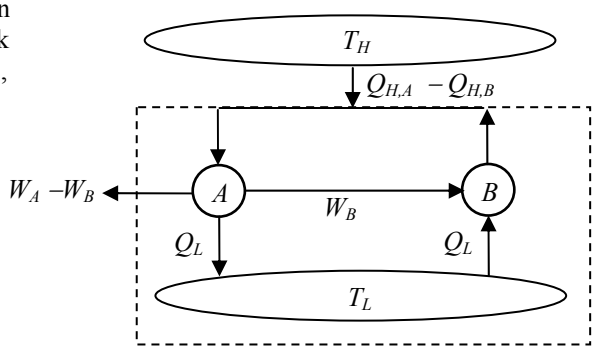


That is, the magnitude of the work required to drive refrigerator *A* is greater than that needed to drive refrigerator *B*. Applying the first law to both refrigerators yields

$$Q_{H,A} > Q_{H,B}$$

since the work supplied to refrigerator *A* is greater than that supplied to refrigerator *B*, and both have the same cooling effect, Q_L.

Since A is a completely reversible refrigerator, we can reverse it without changing the magnitude of the heat and work transfers. This is illustrated in the figure below. The heat, Q_L, which is rejected by the reversed refrigerator *A* can now be routed directly to refrigerator *B*. The net effect when this is done is that no heat is exchanged with the T_L reservoir. The magnitude of the heat supplied to the reversed refrigerator *A*, $Q_{H,A}$ has been shown to be larger than that rejected by refrigerator *B*. There is then a net heat transfer from the T_H reservoir to the combined device in the dashed lines of the figure whose magnitude is given by $Q_{H,A} - Q_{H,B}$. Similarly, there is a net work production by the combined device whose magnitude is given by $W_A - W_B$.



The combined cyclic device then exchanges heat with a reservoir at a single temperature and produces work which is clearly a violation of the Kelvin-Planck statement of the second law. Our assumption the $\text{COP}_A < \text{COP}_B$ must then be wrong.

If we interchange *A* and *B* in the previous argument, we would conclude that the COP_B cannot be less than COP_A. The only alternative left is that

$$\text{COP}_A = \text{COP}_B$$

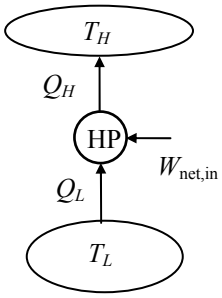
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6-157 An expression for the COP of a completely reversible heat pump in terms of the thermal-energy reservoir temperatures, T_L and T_H is to be derived.

Assumptions The heat pump operates steadily.

Analysis Application of the first law to the completely reversible heat pump yields

$$W_{\text{net,in}} = Q_H - Q_L$$



This result may be used to reduce the coefficient of performance,

$$\text{COP}_{\text{HP,rev}} = \frac{Q_H}{W_{\text{net,in}}} = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - Q_L / Q_H}$$

Since this heat pump is completely reversible, the thermodynamic definition of temperature tells us that,

$$\frac{Q_L}{Q_H} = \frac{T_L}{T_H}$$

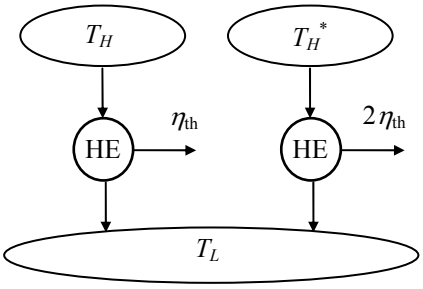
When this is substituted into the COP expression, the result is

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L / T_H} = \frac{T_H}{T_H - T_L}$$

6-158 A Carnot heat engine is operating between specified temperature limits. The source temperature that will double the efficiency is to be determined.

Analysis Denoting the new source temperature by T_H^*, the thermal efficiency of the Carnot heat engine for both cases can be expressed as

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} \quad \text{and} \quad \eta_{\text{th,C}}^* = 1 - \frac{T_L}{T_H^*} = 2\eta_{\text{th,C}}$$



Substituting,

$$1 - \frac{T_L}{T_H^*} = 2\left(1 - \frac{T_L}{T_H}\right)$$

Solving for T_H^*,

$$T_H^* = \frac{T_H T_L}{T_H - 2T_L}$$

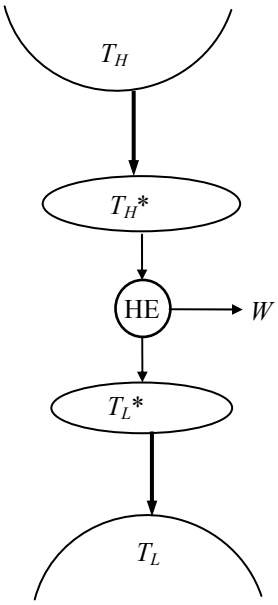
which is the desired relation.

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6-159 A Carnot cycle is analyzed for the case of temperature differences in the boiler and condenser. The ratio of overall temperatures for which the power output will be maximum, and an expression for the maximum net power output are to be determined.

Analysis It is given that

$$\dot{Q}_H = (hA)_H \left(T_H - T_H^*\right).$$



Therefore,

$$\dot{W} = \eta_{\text{th}} \dot{Q}_H = \left(1 - \frac{T_L^*}{T_H^*}\right)(hA)_H \left(T_H - T_H^*\right) = \left(1 - \frac{T_L^*}{T_H^*}\right)(hA)_H \left(1 - \frac{T_H^*}{T_H}\right) T_H$$

or,

$$\frac{\dot{W}}{(hA)_H T_H} = \left(1 - \frac{T_L^*}{T_H^*}\right)\left(1 - \frac{T_H^*}{T_H}\right) = (1 - r)x \qquad (1)$$

where we defined r and x as $r = T_L^*/T_H^*$ and $x = 1 - T_H^*/T_H$.

For a reversible cycle we also have

$$\frac{T_H^*}{T_L^*} = \frac{\dot{Q}_H}{\dot{Q}_L} \longrightarrow \frac{1}{r} = \frac{(hA)_H \left(T_H - T_H^*\right)}{(hA)_L \left(T_L^* - T_L\right)} = \frac{(hA)_H T_H \left(1 - T_H^*/T_H\right)}{(hA)_L T_H \left(T_L^*/T_H - T_L/T_H\right)}$$

but

$$\frac{T_L^*}{T_H} = \frac{T_L^*}{T_H^*}\frac{T_H^*}{T_H} = r(1 - x).$$

Substituting into above relation yields

$$\frac{1}{r} = \frac{(hA)_H x}{(hA)_L \left[r(1 - x) - T_L/T_H\right]}$$

Solving for x,

$$x = \frac{r - T_L/T_H}{r\left[(hA)_H/(hA)_L + 1\right]} \qquad (2)$$

Substitute (2) into (1):

$$\dot{W} = (hA)_H T_H (1 - r)\frac{r - T_L/T_H}{r\left[(hA)_H/(hA)_L + 1\right]} \qquad (3)$$

Taking the partial derivative $\dfrac{\partial \dot{W}}{\partial r}$ holding everything else constant and setting it equal to zero gives

$$r = \frac{T_L^*}{T_H^*} = \left(\frac{T_L}{T_H}\right)^{1/2} \qquad (4)$$

which is the desired relation. The maximum net power output in this case is determined by substituting (4) into (3). It simplifies to

$$\dot{W}_{\max} = \frac{(hA)_H T_H}{1 + (hA)_H/(hA)_L}\left\{1 - \left(\frac{T_L}{T_H}\right)^{1/2}\right\}^2$$

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Fundamentals of Engineering (FE) Exam Problems

6-160 The label on a washing machine indicates that the washer will use $85 worth of hot water if the water is heated by a 90% efficiency electric heater at an electricity rate of $0.09/kWh. If the water is heated from 18°C to 45°C, the amount of hot water an average family uses per year, in metric tons, is

(a) 11.6 tons (b) 15.8 tons (c) 27.1 tons (d) 30.1 tons (e) 33.5 tons

Answer (b) 27.1 tons

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Eff=0.90
C=4.18 "kJ/kg-C"
T1=18 "C"
T2=45 "C"
Cost=85 "$"
Price=0.09 "$/kWh"
Ein=(Cost/Price)*3600 "kJ"
Ein=m*C*(T2-T1)/Eff "kJ"

"Some Wrong Solutions with Common Mistakes:"
Ein=W1_m*C*(T2-T1)*Eff "Multiplying by Eff instead of dividing"
Ein=W2_m*C*(T2-T1) "Ignoring efficiency"
Ein=W3_m*(T2-T1)/Eff "Not using specific heat"
Ein=W4_m*C*(T2+T1)/Eff "Adding temperatures"
```

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6-161 A 2.4-m high 200-m^2 house is maintained at 22°C by an air-conditioning system whose COP is 3.2. It is estimated that the kitchen, bath, and other ventilating fans of the house discharge a houseful of conditioned air once every hour. If the average outdoor temperature is 32°C, the density of air is 1.20 kg/m^3, and the unit cost of electricity is $0.10/kWh, the amount of money "vented out" by the fans in 10 hours is

(a) $0.50 (b) $1.60 (c) $5.00 (d) $11.00 (e) $16.00

Answer (a) $0.50

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP=3.2
T1=22 "C"
T2=32 "C"
Price=0.10 "$/kWh"
Cp=1.005 "kJ/kg-C"
rho=1.20 "kg/m^3"
V=2.4*200 "m^3"
m=rho*V
m_total=m*10
Ein=m_total*Cp*(T2-T1)/COP "kJ"
Cost=(Ein/3600)*Price

"Some Wrong Solutions with Common Mistakes:"
W1_Cost=(Price/3600)*m_total*Cp*(T2-T1)*COP "Multiplying by Eff instead of dividing"
W2_Cost=(Price/3600)*m_total*Cp*(T2-T1)  "Ignoring efficiency"
W3_Cost=(Price/3600)*m*Cp*(T2-T1)/COP "Using m instead of m_total"
W4_Cost=(Price/3600)*m_total*Cp*(T2+T1)/COP  "Adding temperatures"
```

6-162 The drinking water needs of an office are met by cooling tab water in a refrigerated water fountain from 23°C to 6°C at an average rate of 10 kg/h. If the COP of this refrigerator is 3.1, the required power input to this refrigerator is

(a) 197 W (b) 612 W (c) 64 W (d) 109 W (e) 403 W

Answer (c) 64 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP=3.1
Cp=4.18 "kJ/kg-C"
T1=23 "C"
T2=6 "C"
m_dot=10/3600 "kg/s"
Q_L=m_dot*Cp*(T1-T2)  "kW"
W_in=Q_L*1000/COP "W"

"Some Wrong Solutions with Common Mistakes:"
W1_Win=m_dot*Cp*(T1-T2) *1000*COP "Multiplying by COP instead of dividing"
W2_Win=m_dot*Cp*(T1-T2) *1000 "Not using COP"
W3_Win=m_dot*(T1-T2) *1000/COP "Not using specific heat"
W4_Win=m_dot*Cp*(T1+T2) *1000/COP "Adding temperatures"
```

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6-163 A heat pump is absorbing heat from the cold outdoors at 5°C and supplying heat to a house at 25°C at a rate of 18,000 kJ/h. If the power consumed by the heat pump is 1.9 kW, the coefficient of performance of the heat pump is

(a) 1.3 (b) 2.6 (c) 3.0 (d) 3.8 (e) 13.9

Answer (b) 2.6

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=5 "C"
TH=25 "C"
QH=18000/3600 "kJ/s"
Win=1.9 "kW"
COP=QH/Win

"Some Wrong Solutions with Common Mistakes:"
W1_COP=Win/QH "Doing it backwards"
W2_COP=TH/(TH-TL) "Using temperatures in C"
W3_COP=(TH+273)/(TH-TL) "Using temperatures in K"
W4_COP=(TL+273)/(TH-TL) "Finding COP of refrigerator using temperatures in K"
```

6-164 A heat engine cycle is executed with steam in the saturation dome. The pressure of steam is 1 MPa during heat addition, and 0.4 MPa during heat rejection. The highest possible efficiency of this heat engine is

(a) 8.0% (b) 15.6% (c) 20.2% (d) 79.8% (e) 100%

Answer (a) 8.0%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=1000 "kPa"
PL=400 "kPa"
TH=TEMPERATURE(Steam_IAPWS,x=0,P=PH)
TL=TEMPERATURE(Steam_IAPWS,x=0,P=PL)
Eta_Carnot=1-(TL+273)/(TH+273)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta_Carnot=1-PL/PH "Using pressures"
W2_Eta_Carnot=1-TL/TH "Using temperatures in C"
W3_Eta_Carnot=TL/TH "Using temperatures ratio"
```

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6-165 A heat engine receives heat from a source at 1000°C and rejects the waste heat to a sink at 50°C. If heat is supplied to this engine at a rate of 100 kJ/s, the maximum power this heat engine can produce is

(a) 25.4 kW (b) 55.4 kW (c) 74.6 kW (d) 95.0 kW (e) 100.0 kW

Answer (c) 74.6 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TH=1000 "C"
TL=50 "C"
Q_in=100 "kW"
Eta=1-(TL+273)/(TH+273)
W_out=Eta*Q_in

"Some Wrong Solutions with Common Mistakes:"
W1_W_out=(1-TL/TH)*Q_in      "Using temperatures in C"
W2_W_out=Q_in                  "Setting work equal to heat input"
W3_W_out=Q_in/Eta              "Dividing by efficiency instead of multiplying"
W4_W_out=(TL+273)/(TH+273)*Q_in   "Using temperature ratio"
```

6-166 A heat pump cycle is executed with R-134a under the saturation dome between the pressure limits of 1.4 MPa and 0.16 MPa. The maximum coefficient of performance of this heat pump is

(a) 1.1 (b) 3.8 (c) 4.8 (d) 5.3 (e) 2.9

Answer (c) 4.8

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=1400 "kPa"
PL=160 "kPa"
TH=TEMPERATURE(R134a,x=0,P=PH) "C"
TL=TEMPERATURE(R134a,x=0,P=PL) "C"
COP_HP=(TH+273)/(TH-TL)

"Some Wrong Solutions with Common Mistakes:"
W1_COP=PH/(PH-PL)   "Using pressures"
W2_COP=TH/(TH-TL)   "Using temperatures in C"
W3_COP=TL/(TH-TL)   "Refrigeration COP using temperatures in C"
W4_COP=(TL+273)/(TH-TL)   "Refrigeration COP using temperatures in K"
```

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6-167 A refrigeration cycle is executed with R-134a under the saturation dome between the pressure limits of 1.6 MPa and 0.2 MPa. If the power consumption of the refrigerator is 3 kW, the maximum rate of heat removal from the cooled space of this refrigerator is

(a) 0.45 kJ/s (b) 0.78 kJ/s (c) 3.0 kJ/s (d) 11.6 kJ/s (e) 14.6 kJ/s

Answer (d) 11.6 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=1600 "kPa"
PL=200 "kPa"
W_in=3 "kW"
TH=TEMPERATURE(R134a,x=0,P=PH) "C"
TL=TEMPERATURE(R134a,x=0,P=PL) "C"
COP=(TL+273)/(TH-TL)
QL=W_in*COP "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_QL=W_in*TL/(TH-TL)      "Using temperatures in C"
W2_QL=W_in                 "Setting heat removal equal to power input"
W3_QL=W_in/COP             "Dividing by COP instead of multiplying"
W4_QL=W_in*(TH+273)/(TH-TL)  "Using COP definition for Heat pump"
```

6-168 A heat pump with a COP of 3.2 is used to heat a perfectly sealed house (no air leaks). The entire mass within the house (air, furniture, etc.) is equivalent to 1200 kg of air. When running, the heat pump consumes electric power at a rate of 5 kW. The temperature of the house was 7°C when the heat pump was turned on. If heat transfer through the envelope of the house (walls, roof, etc.) is negligible, the length of time the heat pump must run to raise the temperature of the entire contents of the house to 22°C is

(a) 13.5 min (b) 43.1 min (c) 138 min (d) 18.8 min (e) 808 min

Answer (a) 13.5 min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP=3.2
Cv=0.718 "kJ/kg.C"
m=1200 "kg"
T1=7 "C"
T2=22 "C"
QH=m*Cv*(T2-T1)
Win=5 "kW"
Win*time=QH/COP/60

"Some Wrong Solutions with Common Mistakes:"
Win*W1_time*60=m*Cv*(T2-T1) *COP "Multiplying by COP instead of dividing"
Win*W2_time*60=m*Cv*(T2-T1)      "Ignoring COP"
Win*W3_time=m*Cv*(T2-T1) /COP    "Finding time in seconds instead of minutes"
Win*W4_time*60=m*Cp*(T2-T1) /COP "Using Cp instead of Cv"
Cp=1.005 "kJ/kg.K"
```

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6-169 A heat engine cycle is executed with steam in the saturation dome between the pressure limits of 7 MPa and 2 MPa. If heat is supplied to the heat engine at a rate of 150 kJ/s, the maximum power output of this heat engine is

(a) 8.1 kW (b) 19.7 kW (c) 38.6 kW (d) 107 kW (e) 130 kW

Answer (b) 19.7 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=7000 "kPa"
PL=2000 "kPa"
Q_in=150 "kW"
TH=TEMPERATURE(Steam_IAPWS,x=0,P=PH)  "C"
TL=TEMPERATURE(Steam_IAPWS,x=0,P=PL) "C"
Eta=1-(TL+273)/(TH+273)
W_out=Eta*Q_in

"Some Wrong Solutions with Common Mistakes:"
W1_W_out=(1-TL/TH)*Q_in      "Using temperatures in C"
W2_W_out=(1-PL/PH)*Q_in      "Using pressures"
W3_W_out=Q_in/Eta      "Dividing by efficiency instead of multiplying"
W4_W_out=(TL+273)/(TH+273)*Q_in   "Using temperature ratio"
```

6-170 An air-conditioning system operating on the reversed Carnot cycle is required to remove heat from the house at a rate of 32 kJ/s to maintain its temperature constant at 20°C. If the temperature of the outdoors is 35°C, the power required to operate this air-conditioning system is

(a) 0.58 kW (b) 3.20 kW (c) 1.56 kW (d) 2.26 kW (e) 1.64 kW

Answer (e) 1.64 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=20 "C"
TH=35 "C"
QL=32 "kJ/s"
COP=(TL+273)/(TH-TL)
COP=QL/Win

"Some Wrong Solutions with Common Mistakes:"
QL=W1_Win*TL/(TH-TL)     "Using temperatures in C"
QL=W2_Win                    "Setting work equal to heat input"
QL=W3_Win/COP            "Dividing by COP instead of multiplying"
QL=W4_Win*(TH+273)/(TH-TL)     "Using COP of HP"
```

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6-171 A refrigerator is removing heat from a cold medium at 3°C at a rate of 7200 kJ/h and rejecting the waste heat to a medium at 30°C. If the coefficient of performance of the refrigerator is 2, the power consumed by the refrigerator is

(a) 0.1 kW (b) 0.5 kW (c) 1.0 kW (d) 2.0 kW (e) 5.0 kW

Answer (c) 1.0 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=3 "C"
TH=30 "C"
QL=7200/3600 "kJ/s"
COP=2
QL=Win*COP

"Some Wrong Solutions with Common Mistakes:"
QL=W1_Win*(TL+273)/(TH-TL)  "Using Carnot COP"
QL=W2_Win                   "Setting work equal to heat input"
QL=W3_Win/COP               "Dividing by COP instead of multiplying"
QL=W4_Win*TL/(TH-TL)        "Using Carnot COP using C"
```

6-172 Two Carnot heat engines are operating in series such that the heat sink of the first engine serves as the heat source of the second one. If the source temperature of the first engine is 1300 K and the sink temperature of the second engine is 300 K and the thermal efficiencies of both engines are the same, the temperature of the intermediate reservoir is

(a) 625 K (b) 800 K (c) 860 K (d) 453 K (e) 758 K

Answer (a) 625 K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TH=1300 "K"
TL=300 "K"
"Setting thermal efficiencies equal to each other:"
1-Tmid/TH=1-TL/Tmid

"Some Wrong Solutions with Common Mistakes:"
W1_Tmid=(TL+TH)/2 "Using average temperature"
```

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6-173 Consider a Carnot refrigerator and a Carnot heat pump operating between the same two thermal energy reservoirs. If the COP of the refrigerator is 3.4, the COP of the heat pump is

(a) 1.7 (b) 2.4 (c) 3.4 (d) 4.4 (e) 5.0

Answer (d) 4.4

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP_R=3.4
COP_HP=COP_R+1

"Some Wrong Solutions with Common Mistakes:"
W1_COP=COP_R-1 "Subtracting 1 instead of adding 1"
W2_COP=COP_R "Setting COPs equal to each other"
```

6-174 A typical new household refrigerator consumes about 680 kWh of electricity per year, and has a coefficient of performance of 1.4. The amount of heat removed by this refrigerator from the refrigerated space per year is

(*a*) 952 MJ/yr (*b*) 1749 MJ/yr (*c*) 2448 MJ/yr (*d*) 3427 MJ/yr (*e*) 4048 MJ/yr

Answer (*d*) 3427 MJ/yr

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W_in=680*3.6 "MJ"
COP_R=1.4
QL=W_in*COP_R "MJ"

"Some Wrong Solutions with Common Mistakes:"
W1_QL=W_in*COP_R/3.6 "Not using the conversion factor"
W2_QL=W_in "Ignoring COP"
W3_QL=W_in/COP_R "Dividing by COP instead of multiplying"
```

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6-175 A window air conditioner that consumes 1 kW of electricity when running and has a coefficient of performance of 3 is placed in the middle of a room, and is plugged in. The rate of cooling or heating this air conditioner will provide to the air in the room when running is

(*a*) 3 kJ/s, cooling (*b*) 1 kJ/s, cooling (*c*) 0.33 kJ/s, heating (*d*) 1 kJ/s, heating (*e*) 3 kJ/s, heating

Answer (*d*) 1 kJ/s, heating

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W_in=1 "kW"
COP=3
"From energy balance, heat supplied to the room is equal to electricity consumed,"
E_supplied=W_in "kJ/s, heating"

"Some Wrong Solutions with Common Mistakes:"
W1_E=-W_in "kJ/s, cooling"
W2_E=-COP*W_in "kJ/s, cooling"
W3_E=W_in/COP "kJ/s, heating"
W4_E=COP*W_in "kJ/s, heating"
```

6-176 ··· 6-182 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 7
ENTROPY

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Entropy and the Increase of Entropy Principle

7-1C No. A system may produce more (or less) work than it receives during a cycle. A steam power plant, for example, produces more work than it receives during a cycle, the difference being the net work output.

7-2C The entropy change will be the same for both cases since entropy is a property and it has a fixed value at a fixed state.

7-3C No. In general, that integral will have a different value for different processes. However, it will have the same value for all reversible processes.

7-4C That integral should be performed along a reversible path to determine the entropy change.

7-5C No. An isothermal process can be irreversible. Example: A system that involves paddle-wheel work while losing an equivalent amount of heat.

7-6C The value of this integral is always larger for reversible processes.

7-7C No. Because the entropy of the surrounding air increases even more during that process, making the total entropy change positive.

7-8C It is possible to create entropy, but it is not possible to destroy it.

7-9C If the system undergoes a reversible process, the entropy of the system cannot change without a heat transfer. Otherwise, the entropy must increase since there are no offsetting entropy changes associated with reservoirs exchanging heat with the system.

7-10C The claim that work will not change the entropy of a fluid passing through an adiabatic steady-flow system with a single inlet and outlet is true only if the process is also reversible. Since no real process is reversible, there will be an entropy increase in the fluid during the adiabatic process in devices such as pumps, compressors, and turbines.

7-11C Sometimes.

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7-12C Never.

7-13C Always.

7-14C Increase.

7-15C Increases.

7-16C Decreases.

7-17C Sometimes.

7-18C Greater than.

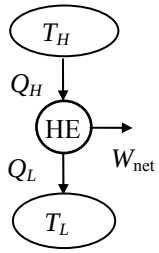
7-19C Yes. This will happen when the system is losing heat, and the decrease in entropy as a result of this heat loss is equal to the increase in entropy as a result of irreversibilities.

7-20C They are heat transfer, irreversibilities, and entropy transport with mass.

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7-21E The source and sink temperatures and the entropy change of the sink for a completely reversible heat engine are given. The entropy decrease of the source and the amount of heat transfer from the source are to be determined.

Assumptions The heat engine operates steadily.



Analysis According to the increase in entropy principle, the entropy change of the source must be equal and opposite to that of the sink. Hence,

$$\Delta S_H = -\Delta S_L = \mathbf{-10\ Btu/R}$$

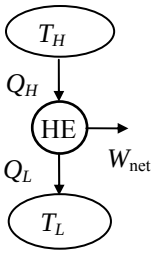
Applying the definition of the entropy to the source gives

$$Q_H = T_H \Delta S_H = (1500\ \text{R})(-10\ \text{Btu/R}) = \mathbf{-15{,}000\ Btu}$$

which is the heat transfer with respect to the source, not the device.

7-22 The source and sink temperatures and the entropy change of the sink for a completely reversible heat engine are given. The amount of heat transfer from the source are to be determined.

Assumptions The heat engine operates steadily.



Analysis According to the increase in entropy principle, the entropy change of the source must be equal and opposite to that of the sink. Hence,

$$\Delta S_H = -\Delta S_L = -20\ \text{kJ/K}$$

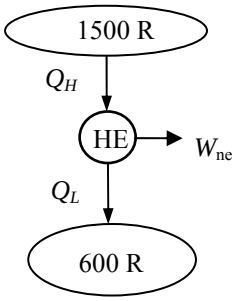
Applying the definition of the entropy to the source gives

$$Q_H = T_H \Delta S_H = (1000\ \text{K})(-20\ \text{kJ/K}) = \mathbf{-20{,}000\ kJ}$$

which is the heat transfer with respect to the source, not the device.

7-23E The operating conditions of a heat engine are given. The entropy change of all components and the work input are to be calculated and it is to be determined if this heat engine is reversible.

Assumptions The heat engine operates steadily.



Analysis The entropy change of all the components is

$$\Delta S_{\text{total}} = \Delta S_H + \Delta S_L + \Delta S_{\text{device}}$$

where the last term is zero each time the engine completes a cycle. Applying the definition of the entropy to the two reservoirs reduces this to

$$\Delta S_{\text{total}} = \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{-200{,}000\ \text{Btu}}{1500\ \text{R}} + \frac{100{,}000\ \text{Btu}}{600\ \text{R}} = \mathbf{33.3\ Btu/R}$$

Since the entropy of everything involved with this engine has increased, the engine is not reversible, but possible. Applying the first law to this engine,

$$W_{\text{net}} = Q_H - Q_L = 200{,}000\ \text{Btu} - 100{,}000\ \text{Btu} = \mathbf{100{,}000\ Btu}$$

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7-24 Air is compressed steadily by a compressor. The air temperature is maintained constant by heat rejection to the surroundings. The rate of entropy change of air is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas. **4** The process involves no internal irreversibilities such as friction, and thus it is an isothermal, internally reversible process.

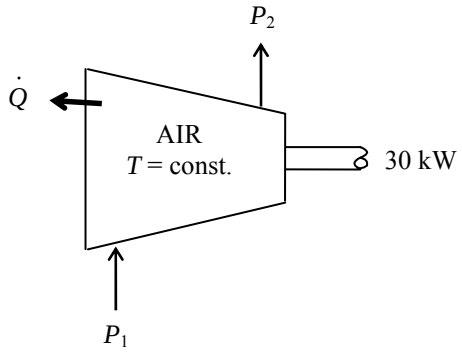
Properties Noting that $h = h(T)$ for ideal gases, we have $h_1 = h_2$ since $T_1 = T_2 = 25°C$.

Analysis We take the compressor as the system. Noting that the enthalpy of air remains constant, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}}$$



Therefore,

$$\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 30\,\text{kW}$$

Noting that the process is assumed to be an isothermal and internally reversible process, the rate of entropy change of air is determined to be

$$\Delta \dot{S}_{\text{air}} = -\frac{\dot{Q}_{\text{out,air}}}{T_{\text{sys}}} = -\frac{30\,\text{kW}}{298\,\text{K}} = \mathbf{-0.101\,kW/K}$$

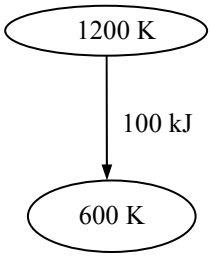
7-25 Heat is transferred directly from an energy-source reservoir to an energy-sink. The entropy change of the two reservoirs is to be calculated and it is to be determined if the increase of entropy principle is satisfied.

Assumptions The reservoirs operate steadily.

Analysis The entropy change of the source and sink is given by

$$\Delta S = \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{-100\,\text{kJ}}{1200\,\text{K}} + \frac{100\,\text{kJ}}{600\,\text{K}} = \mathbf{0.0833\,kJ/K}$$

Since the entropy of everything involved in this process has increased, this transfer of heat is **possible**.



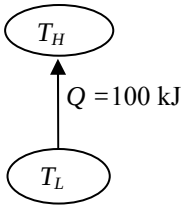
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7-26 It is assumed that heat is transferred from a cold reservoir to the hot reservoir contrary to the Clausius statement of the second law. It is to be proven that this violates the increase in entropy principle.

Assumptions The reservoirs operate steadily.

Analysis According to the definition of the entropy, the entropy change of the high-temperature reservoir shown below is

$$\Delta S_H = \frac{Q}{T_H} = \frac{100\ \text{kJ}}{1200\ \text{K}} = 0.08333\ \text{kJ/K}$$



and the entropy change of the low-temperature reservoir is

$$\Delta S_L = \frac{Q}{T_L} = \frac{-100\ \text{kJ}}{600\ \text{K}} = -0.1667\ \text{kJ/K}$$

The total entropy change of everything involved with this system is then

$$\Delta S_{\text{total}} = \Delta S_H + \Delta S_L = 0.08333 - 0.1667 = \mathbf{-0.0833\ kJ/K}$$

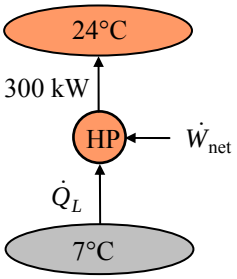
which violates the increase in entropy principle since the total entropy change is negative.

7-27 A reversible heat pump with specified reservoir temperatures is considered. The entropy change of two reservoirs is to be calculated and it is to be determined if this heat pump satisfies the increase in entropy principle.

Assumptions The heat pump operates steadily.

Analysis Since the heat pump is completely reversible, the combination of the coefficient of performance expression, first Law, and thermodynamic temperature scale gives

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (280\ \text{K})/(297\ \text{K})} = 17.47$$



The power required to drive this heat pump, according to the coefficient of performance, is then

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,rev}}} = \frac{300\ \text{kW}}{17.47} = 17.17\ \text{kW}$$

According to the first law, the rate at which heat is removed from the low-temperature energy reservoir is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 300\ \text{kW} - 17.17\ \text{kW} = 282.8\ \text{kW}$$

The rate at which the entropy of the high temperature reservoir changes, according to the definition of the entropy, is

$$\Delta \dot{S}_H = \frac{\dot{Q}_H}{T_H} = \frac{300\ \text{kW}}{297\ \text{K}} = \mathbf{1.01\ kW/K}$$

and that of the low-temperature reservoir is

$$\Delta \dot{S}_L = \frac{\dot{Q}_L}{T_L} = \frac{-17.17\ \text{kW}}{280\ \text{K}} = \mathbf{-1.01\ kW/K}$$

The net rate of entropy change of everything in this system is

$$\Delta \dot{S}_{\text{total}} = \Delta \dot{S}_H + \Delta \dot{S}_L = 1.01 - 1.01 = \mathbf{0\ kW/K}$$

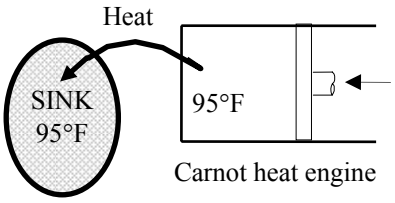
as it must be since the heat pump is completely reversible.

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7-28E Heat is transferred isothermally from the working fluid of a Carnot engine to a heat sink. The entropy change of the working fluid is given. The amount of heat transfer, the entropy change of the sink, and the total entropy change during the process are to be determined.

Analysis (*a*) This is a reversible isothermal process, and the entropy change during such a process is given by

$$\Delta S = \frac{Q}{T}$$



Noting that heat transferred from the working fluid is equal to the heat transferred to the sink, the heat transfer become

$$Q_{\text{fluid}} = T_{\text{fluid}}\Delta S_{\text{fluid}} = (555\ \text{R})(-0.7\ \text{Btu/R}) = -388.5\ \text{Btu} \ \rightarrow \ Q_{\text{fluid,out}} = \mathbf{388.5\ Btu}$$

(*b*) The entropy change of the sink is determined from

$$\Delta S_{\text{sink}} = \frac{Q_{\text{sink,in}}}{T_{\text{sink}}} = \frac{388.5\ \text{Btu}}{555\ \text{R}} = \mathbf{0.7\ Btu/R}$$

(*c*) Thus the total entropy change of the process is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{fluid}} + \Delta S_{\text{sink}} = -0.7 + 0.7 = \mathbf{0}$$

This is expected since all processes of the Carnot cycle are reversible processes, and no entropy is generated during a reversible process.

7-29 R-134a enters an evaporator as a saturated liquid-vapor at a specified pressure. Heat is transferred to the refrigerant from the cooled space, and the liquid is vaporized. The entropy change of the refrigerant, the entropy change of the cooled space, and the total entropy change for this process are to be determined.

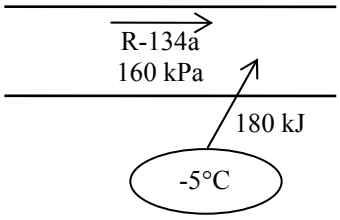
Assumptions **1** Both the refrigerant and the cooled space involve no internal irreversibilities such as friction. **2** Any temperature change occurs within the wall of the tube, and thus both the refrigerant and the cooled space remain isothermal during this process. Thus it is an isothermal, internally reversible process.

Analysis Noting that both the refrigerant and the cooled space undergo reversible isothermal processes, the entropy change for them can be determined from

$$\Delta S = \frac{Q}{T}$$

(*a*) The pressure of the refrigerant is maintained constant. Therefore, the temperature of the refrigerant also remains constant at the saturation value,

$$T = T_{\text{sat@160 kPa}} = -15.6°\text{C} = 257.4\ \text{K} \qquad \text{(Table A-12)}$$



Then, $$\Delta S_{\text{refrigerant}} = \frac{Q_{\text{refrigerant,in}}}{T_{\text{refrigerant}}} = \frac{180\ \text{kJ}}{257.4\ \text{K}} = \mathbf{0.699\ kJ/K}$$

(*b*) Similarly,

$$\Delta S_{\text{space}} = -\frac{Q_{\text{space,out}}}{T_{\text{space}}} = -\frac{180\ \text{kJ}}{268\ \text{K}} = \mathbf{-0.672\ kJ/K}$$

(*c*) The total entropy change of the process is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{refrigerant}} + \Delta S_{\text{space}} = 0.699 - 0.672 = \mathbf{0.027\ kJ/K}$$

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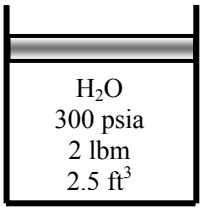
Entropy Changes of Pure Substances

7-30C Yes, because an internally reversible, adiabatic process involves no irreversibilities or heat transfer.

7-31E A piston-cylinder device that is filled with water is heated. The total entropy change is to be determined.

Analysis The initial specific volume is

$$v_1 = \frac{V_1}{m} = \frac{2.5\ \text{ft}^3}{2\ \text{lbm}} = 1.25\ \text{ft}^3/\text{lbm}$$



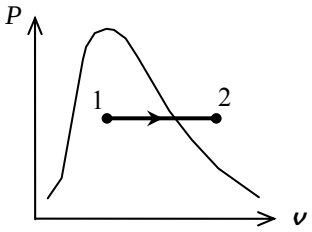
which is between v_f and v_g for 300 psia. The initial quality and the entropy are then (Table A-5E)

$$x_1 = \frac{v_1 - v_f}{v_{fg}} = \frac{(1.25 - 0.01890)\ \text{ft}^3/\text{lbm}}{(1.5435 - 0.01890)\ \text{ft}^3/\text{lbm}} = 0.8075$$

$$s_1 = s_f + x_1 s_{fg} = 0.58818\ \text{Btu/lbm}\cdot\text{R} + (0.8075)(0.92289\ \text{Btu/lbm}\cdot\text{R}) = 1.3334\ \text{Btu/lbm}\cdot\text{R}$$

The final state is superheated vapor and

$$\left.\begin{array}{l} T_2 = 500°\text{F} \\ P_2 = P_1 = 300\ \text{psia} \end{array}\right\} \quad s_2 = 1.5706\ \text{Btu/lbm}\cdot\text{R} \quad \text{(Table A - 6E)}$$



Hence, the change in the total entropy is

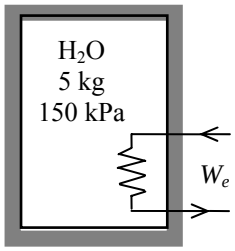
$$\begin{aligned} \Delta S &= m(s_2 - s_1) \\ &= (2\ \text{lbm})(1.5706 - 1.3334)\ \text{Btu/lbm}\cdot\text{R} \\ &= \mathbf{0.4744\ Btu/R} \end{aligned}$$

7-32 An insulated rigid tank contains a saturated liquid-vapor mixture of water at a specified pressure. An electric heater inside is turned on and kept on until all the liquid vaporized. The entropy change of the water during this process is to be determined.

Analysis From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 150\ \text{kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.001053 + (0.25)(1.1594 - 0.001053) = 0.29065\ \text{m}^3/\text{kg} \\ s_1 = s_f + x_1 s_{fg} = 1.4337 + (0.25)(5.7894) = 2.8810\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} v_2 = v_1 \\ \text{sat. vapor} \end{array}\right\} s_2 = 6.7298\ \text{kJ/kg}\cdot\text{K}$$

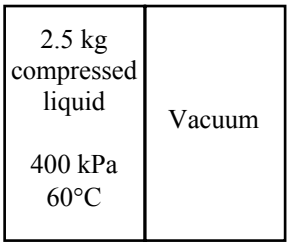


Then the entropy change of the steam becomes

$$\Delta S = m(s_2 - s_1) = (5\ \text{kg})(6.7298 - 2.8810)\ \text{kJ/kg}\cdot\text{K} = \mathbf{19.2\ kJ/K}$$

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7-33 A rigid tank is divided into two equal parts by a partition. One part is filled with compressed liquid water while the other side is evacuated. The partition is removed and water expands into the entire tank. The entropy change of the water during this process is to be determined.



Analysis The properties of the water are (Table A-4)

$$\left.\begin{array}{l} P_1 = 400\ \text{kPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} \nu_1 \cong \nu_{f@60°\text{C}} = 0.001017\,\text{m}^3/\text{kg} \\ s_1 = s_{f@60°\text{C}} = 0.8313\ \text{kJ/kg}\cdot\text{K} \end{array}$$

Noting that

$$\nu_2 = 2\nu_1 = (2)(0.001017) = 0.002034\ \text{m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 40\ \text{kPa} \\ \nu_2 = 0.002034\ \text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{\nu_2 - \nu_f}{\nu_{fg}} = \dfrac{0.002034 - 0.001026}{3.993 - 0.001026} = 0.0002524 \\ s_2 = s_f + x_2 s_{fg} = 1.0261 + (0.0002524)(6.6430) = 1.0278\ \text{kJ/kg}\cdot\text{K} \end{array}$$

Then the entropy change of the water becomes

$$\Delta S = m(s_2 - s_1) = (2.5\ \text{kg})(1.0278 - 0.8313)\,\text{kJ/kg}\cdot\text{K} = \mathbf{0.492\ kJ/K}$$

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7-34 Problem 7-33 is reconsidered. The entropy generated is to be evaluated and plotted as a function of surroundings temperature, and the values of the surroundings temperatures that are valid for this problem are to be determined. The surrounding temperature is to vary from 0°C to 100°C.

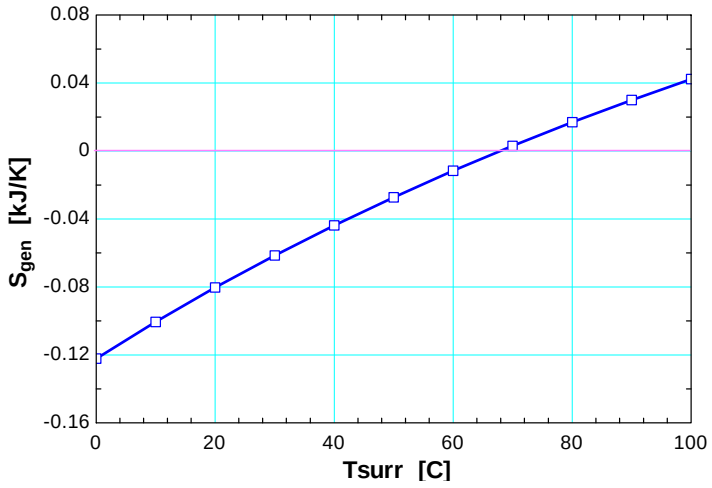
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Input Data"
P[1]=400 [kPa]
T[1]=60 [C]
m=2.5 [kg]
P[2]=40 [kPa]

Fluid$='Steam_IAPWS'
V[1]=m*spv[1]
spv[1]=volume(Fluid$,T=T[1], P=P[1]) "specific volume of steam at state 1, m^3/kg"
s[1]=entropy(Fluid$,T=T[1],P=P[1]) "entropy of steam at state 1, kJ/kgK"
V[2]=2*V[1] "Steam expands to fill entire volume at state 2"
"State 2 is identified by P[2] and spv[2]"
spv[2]=V[2]/m "specific volume of steam at state 2, m^3/kg"
s[2]=entropy(Fluid$,P=P[2],v=spv[2])  "entropy of steam at state 2, kJ/kgK"
T[2]=temperature(Fluid$,P=P[2],v=spv[2])
DELTAS_sys=m*(s[2]-s[1])  "Total entopy change of steam, kJ/K"
"What does the first law tell us about this problem?"
E_in - E_out = DELTAE_sys "Conservation of Energy for the entire, closed system"
"neglecting changes in KE and PE for the system:"
DELTAE_sys=m*(intenergy(Fluid$, P=P[2], v=spv[2]) - intenergy(Fluid$,T=T[1],P=P[1]))
E_in = 0
"How do you interpert the energy leaving the system, E_out?  Recall this is a constant volume system."
Q_out = E_out
"What is the maximum value of the Surroundings temperature?"
"The maximum possible value for the surroundings temperature occurs when we set S_gen = 0=Delta
S_sys+sum(DeltaS_surr)"
Q_net_surr=Q_out
S_gen = 0
S_gen = DELTAS_sys+Q_net_surr/(Tsurr+273)

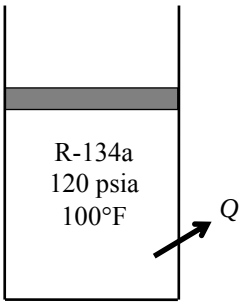
"Establish a parametric table for the variables S_gen, Q_net_surr, T_surr, and DELTAS_sys.  In the Parametric
Table window select T_surr and insert a range of values.  Then place '{' and '}'  about the S_gen = 0 line; press
F3 to solve the table.  The results are shown in Plot Window 1.  What values of T_surr are valid for this
problem?"
```

T_{surr} [C]	S_{gen} [kJ/K]
0	-0.1222
10	-0.1005
20	-0.08033
30	-0.06145
40	-0.04378
50	-0.0272
60	-0.01162
70	0.003049
80	0.01689
90	0.02997
100	0.04235



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7-35E A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled and condensed at constant pressure. The entropy change of refrigerant during this process is to be determined



Analysis From the refrigerant tables (Tables A-11E through A-13E),

$$\left.\begin{matrix} P_1 = 120 \text{ psia} \\ T_1 = 100°\text{F} \end{matrix}\right\} s_1 = 0.22361 \text{ Btu/lbm} \cdot \text{R}$$

$$\left.\begin{matrix} T_2 = 50°\text{F} \\ P_2 = 120 \text{ psia} \end{matrix}\right\} s_2 \cong s_{f@90°\text{F}} = 0.06039 \text{ Btu/lbm} \cdot \text{R}$$

Then the entropy change of the refrigerant becomes

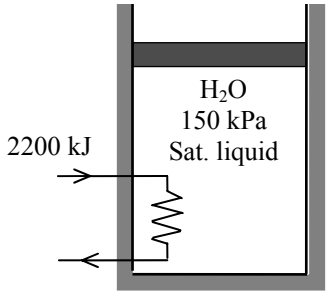
$$\Delta S = m(s_2 - s_1) = (2 \text{ lbm})(0.06039 - 0.22361)\text{Btu/lbm} \cdot \text{R} = \mathbf{-0.3264\ Btu/R}$$

7-36 An insulated cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically at constant pressure. The entropy change of the water during this process is to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis From the steam tables (Tables A-4 through A-6),

$$\left.\begin{matrix} P_1 = 150 \text{ kPa} \\ sat.liquid \end{matrix}\right\} \begin{matrix} v_1 = v_{f@150 \text{ kPa}} = 0.001053 \text{ m}^3/\text{kg} \\ h_1 = h_{f@150 \text{ kPa}} = 467.13 \text{ kJ/kg} \\ s_1 = s_{f@150 \text{ kPa}} = 1.4337 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$



Also, $$m = \frac{V}{v_1} = \frac{0.005 \text{ m}^3}{0.001053 \text{ m}^3/\text{kg}} = 4.75 \text{ kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\begin{matrix}\text{Net energy transfer} \\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta E_{\text{system}}}_{\begin{matrix}\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}\end{matrix}}$$

$$W_{e,\text{in}} - W_{b,\text{out}} = \Delta U$$

$$W_{e,\text{in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Solving for h_2,

$$h_2 = h_1 + \frac{W_{e,\text{in}}}{m} = 467.13 + \frac{2200 \text{ kJ}}{4.75 \text{ kg}} = 930.33 \text{ kJ/kg}$$

Thus,

$$\left.\begin{matrix} P_2 = 150 \text{ kPa} \\ h_2 = 930.33 \text{ kJ/kg} \end{matrix}\right\} \begin{matrix} x_2 = \dfrac{h_2 - h_f}{h_{fg}} = \dfrac{930.33 - 467.13}{2226.0} = 0.2081 \\ s_2 = s_f + x_2 s_{fg} = 1.4337 + (0.2081)(5.7894) = 2.6384 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$

Then the entropy change of the water becomes

$$\Delta S = m(s_2 - s_1) = (4.75 \text{ kg})(2.6384 - 1.4337)\text{kJ/kg} \cdot \text{K} = \mathbf{5.72\ kJ/K}$$

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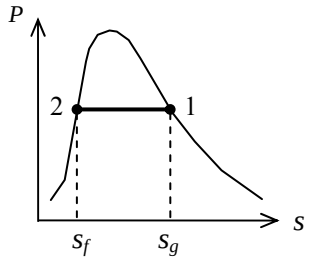
7-37 Entropy change of water as it is cooled at constant pressure is to be determined using Gibbs equation and to be compared to steam tables.

Analysis The Gibbs equation is

$$Tds = dh - vdP$$

As water is converted from a saturated liquid to a saturated vapor, both the pressure and temperature remain constant. Then, the Gibbs equation reduces to

$$ds = \frac{dh}{T}$$



When the result is integrated between the saturated liquid and saturated vapor states, the result is

$$s_g - s_f = \frac{h_g - h_f}{T} = \frac{h_g - h_f}{T} = \frac{h_{fg@300\,\text{kPa}}}{T_{\text{sat}@300\,\text{kPa}}} = \frac{2163.5\,\text{kJ/kg}}{(133.52+273)\text{K}} = \mathbf{5.322\,kJ/kg\cdot K}$$

Where enthalpy and temperature data are obtained from Table A-5. The entropy change from the steam tables is

$$s_{fg@300\,\text{kPa}} = \mathbf{5.320\,kJ/kg\cdot K} \quad (\text{Table A - 5})$$

The result is practically the same.

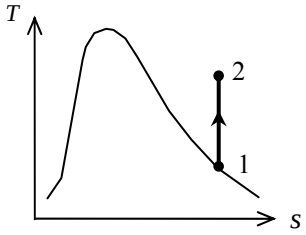
7-38E R-134a is compressed in a compressor during which the entropy remains constant. The final temperature and enthalpy change are to be determined.

Analysis The initial state is saturated vapor and the properties are (Table A-11E)

$$h_1 = h_{g@0°\text{F}} = 103.08\,\text{Btu/lbm}$$
$$s_1 = s_{g@0°\text{F}} = 0.22539\,\text{Btu/lbm}\cdot\text{R}$$

The final state is superheated vapor and the properties are (Table A-13E)

$$\left.\begin{array}{ll} P_2 = 60\,\text{psia} & \\ s_2 = s_1 = 0.22539\,\text{Btu/lbm}\cdot\text{R} & \end{array}\right\} \begin{array}{l} T_2 = \mathbf{59.3°F} \\ h_2 = 112.23\,\text{Btu/lbm} \end{array}$$



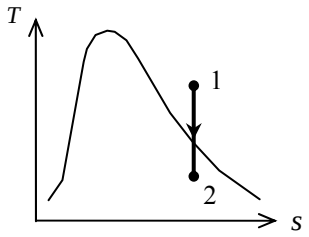
The change in the enthalpy across the compressor is then

$$\Delta h = h_2 - h_1 = 112.23 - 103.08 = \mathbf{9.15\,Btu/lbm}$$

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7-39 Water vapor is expanded in a turbine during which the entropy remains constant. The enthalpy difference is to be determined.

Analysis The initial state is superheated vapor and thus



$$\left.\begin{array}{l} P_1 = 6\ \text{MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3178.3\ \text{kJ/kg} \\ s_1 = 6.5432\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 6)}$$

The entropy is constant during the process. The final state is a mixture since the entropy is between s_f and s_g for 100 kPa. The properties at this state are (Table A-5)

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{(6.5432 - 1.3028)\ \text{kJ/kg}\cdot\text{K}}{6.0562\ \text{kJ/kg}\cdot\text{K}} = 0.8653$$

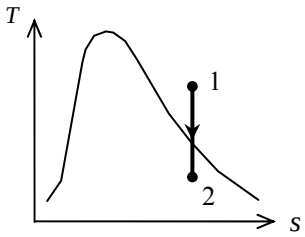
$$h_2 = h_f + x_2 h_{fg} = 417.51 + (0.8653)(2257.5) = 2370.9\ \text{kJ/kg}$$

The change in the enthalpy across the turbine is then

$$\Delta h = h_2 - h_1 = 2370.9 - 3178.3 = \mathbf{-807.4\ kJ/kg}$$

7-40 R-134a undergoes a process during which the entropy is kept constant. The final temperature and internal energy are to be determined.

Analysis The initial entropy is



$$\left.\begin{array}{l} T_1 = 25°\text{C} \\ P_1 = 600\ \text{kPa} \end{array}\right\} \ s_1 = 0.9341\ \text{kJ/kg}\cdot\text{K} \quad \text{(Table A -13)}$$

The entropy is constant during the process. The final state is a mixture since the entropy is between s_f and s_g for 100 kPa. The properties at this state are (Table A-12)

$$T_2 = T_{\text{sat @ 100 kPa}} = \mathbf{-26.37°C}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{(0.9341 - 0.07188)\ \text{kJ/kg}\cdot\text{K}}{0.87995\ \text{kJ/kg}\cdot\text{K}} = 0.9799$$

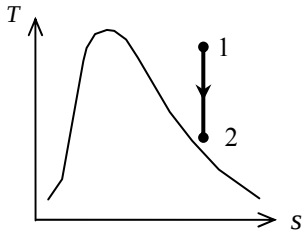
$$u_2 = u_f + x_2 u_{fg} = 17.21 + (0.9799)(197.98) = \mathbf{211.2\ kJ/kg}$$

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7-41 Refrigerant-134a is is expanded in a turbine during which the entropy remains constant. The inlet and outlet velocities are to be determined.

Analysis The initial state is superheated vapor and thus

$$\left.\begin{array}{l} P_1 = 800\ \text{kPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.02997\ \text{m}^3/\text{kg} \\ s_1 = 1.0110\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 13)}$$



The entropy is constant during the process. The properties at the exit state are

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ s_2 = s_1 = 1.0110\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \quad v_2 = 0.2098\ \text{m}^3/\text{kg} \ \text{(Table A - 13)}$$

The inlet and outlet veloicites are

$$V_1 = \frac{\dot{m}v_1}{A_1} = \frac{(0.5\ \text{kg/s})(0.02997\ \text{m}^3/\text{kg})}{0.5\ \text{m}^2} = \mathbf{0.030\ m/s}$$

$$V_2 = \frac{\dot{m}v_2}{A_2} = \frac{(0.5\ \text{kg/s})(0.2098\ \text{m}^3/\text{kg})}{1.0\ \text{m}^2} = \mathbf{0.105\ m/s}$$

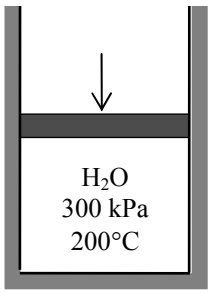
7-42 An insulated cylinder is initially filled with superheated steam at a specified state. The steam is compressed in a reversible manner until the pressure drops to a specified value. The work input during this process is to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

Analysis This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 300\ \text{kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.71643\ \text{m}^3/\text{kg} \\ u_1 = 2651.0\ \text{kJ/kg} \\ s_1 = 7.3132\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} u_2 = 2921.6\ \text{kJ/kg}$$



Also,

$$m = \frac{V}{v_1} = \frac{0.02\ \text{m}^3}{0.71643\ \text{m}^3/\text{kg}} = 0.02792\ \text{kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this adiabatic closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$

Substituting, the work input during this adiabatic process is determined to be

$$W_{\text{b,in}} = m(u_2 - u_1) = (0.02792\ \text{kg})(2921.6 - 2651.0)\ \text{kJ/kg} = \mathbf{7.55\ kJ}$$

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7-43 Problem 7-42 is reconsidered. The work done on the steam is to be determined and plotted as a function of final pressure as the pressure varies from 300 kPa to 1.2 MPa.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

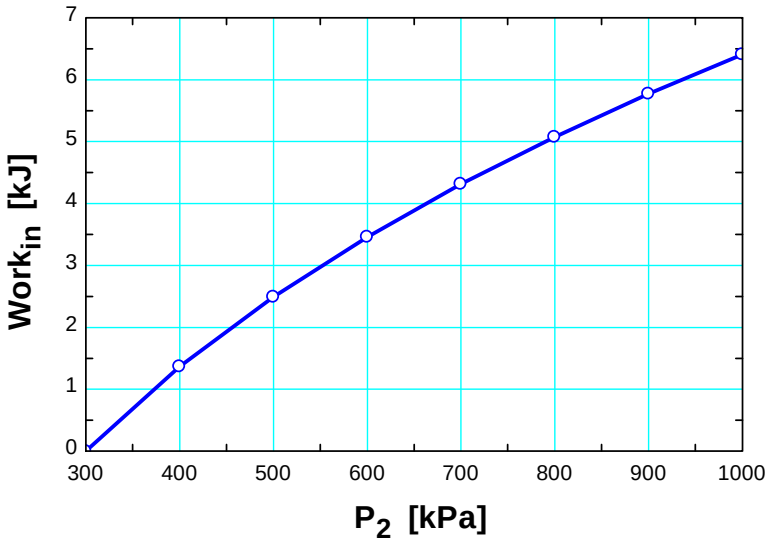
```
"Knowns:"
P_1 = 300 [kPa]
T_1 = 200 [C]
V_sys = 0.02 [m^3]
"P_2 = 1200 [kPa]"

"Analysis: "
Fluid$='Steam_IAPWS'

" Treat the piston-cylinder as a closed system, with no heat transfer in, neglect
changes in KE and PE of the Steam.  The process is reversible and adiabatic thus isentropic."
"The isentropic work is determined from:"
E_in - E_out = DELTAE_sys
E_out = 0 [kJ]
E_in = Work_in
DELTAE_sys = m_sys*(u_2 - u_1)
u_1 = INTENERGY(Fluid$,P=P_1,T=T_1)
v_1 = volume(Fluid$,P=P_1,T=T_1)
s_1 = entropy(Fluid$,P=P_1,T=T_1)
V_sys = m_sys*v_1

" The process is reversible and adiabatic or isentropic.
s_2 = s_1
u_2 = INTENERGY(Fluid$,P=P_2,s=s_2)
T_2_isen = temperature(Fluid$,P=P_2,s=s_2)
```

P_2 [kPa]	$Work_{in}$ [kJ]
300	0
400	1.366
500	2.494
600	3.462
700	4.314
800	5.078
900	5.773
1000	6.411

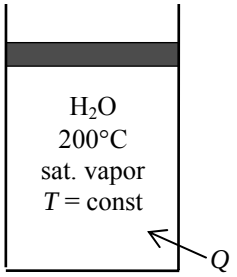


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7-44 A cylinder is initially filled with saturated water vapor at a specified temperature. Heat is transferred to the steam, and it expands in a reversible and isothermal manner until the pressure drops to a specified value. The heat transfer and the work output for this process are to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible and isothermal.

Analysis From the steam tables (Tables A-4 through A-6),



$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ \textit{sat.vapor} \end{array}\right\} \begin{array}{l} u_1 = u_{g@200°\text{C}} = 2594.2 \text{ kJ/kg} \\ s_1 = s_{g@200°\text{C}} = 6.4302 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = T_1 \end{array}\right\} \begin{array}{l} u_2 = 2631.1 \text{ kJ/kg} \\ s_2 = 6.8177 \text{ kJ/kg} \cdot \text{K} \end{array}$$

The heat transfer for this reversible isothermal process can be determined from

$$Q = T\Delta S = Tm(s_2 - s_1) = (473 \text{ K})(1.2 \text{ kg})(6.8177 - 6.4302)\text{kJ/kg} \cdot \text{K} = \mathbf{219.9 \ \ kJ}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$W_{\text{b,out}} = Q_{\text{in}} - m(u_2 - u_1)$$

Substituting, the work done during this process is determined to be

$$W_{\text{b,out}} = 219.9 \text{ kJ} - (1.2 \text{ kg})(2631.1 - 2594.2) \text{ kJ/kg} = \mathbf{175.6 \ \ kJ}$$

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7-45 Problem 7-44 is reconsidered. The heat transferred to the steam and the work done are to be determined and plotted as a function of final pressure as the pressure varies from the initial value to the final value of 800 kPa.

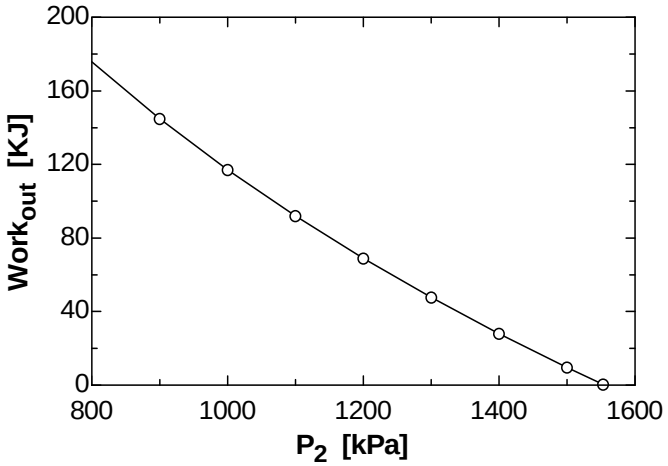
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
T_1 = 200 [C]
x_1 = 1.0
m_sys = 1.2 [kg]
{P_2 = 800"[kPa]"}

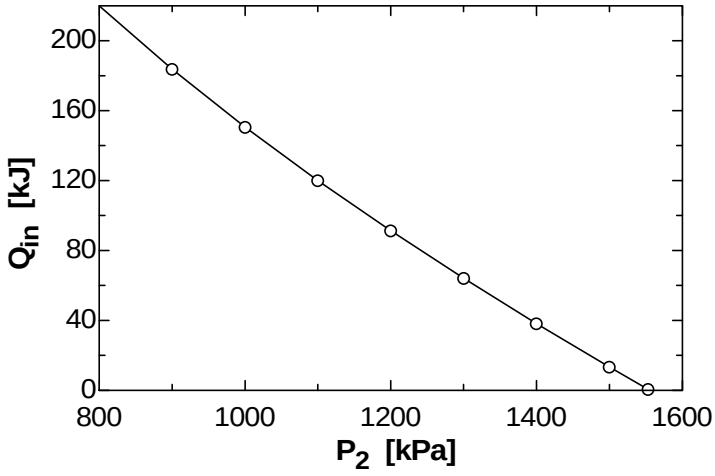
"Analysis: "
Fluid$='Steam_IAPWS'

" Treat the piston-cylinder as a closed system,  neglect changes in KE and PE of the Steam.  The process is reversible and isothermal ."
T_2 = T_1
E_in - E_out = DELTAE_sys
E_in = Q_in
E_out = Work_out
DELTAE_sys = m_sys*(u_2 - u_1)
P_1 = pressure(Fluid$,T=T_1,x=1.0)
u_1 = INTENERGY(Fluid$,T=T_1,x=1.0)
v_1 = volume(Fluid$,T=T_1,x=1.0)
s_1 = entropy(Fluid$,T=T_1,x=1.0)
V_sys = m_sys*v_1

" The process is reversible and isothermal.
Then P_2 and T_2 specify state 2."
u_2 = INTENERGY(Fluid$,P=P_2,T=T_2)
s_2 = entropy(Fluid$,P=P_2,T=T_2)
Q_in= (T_1+273)*m_sys*(s_2-s_1)
```



P_2 [kPa]	Q_{in} [kJ]	$Work_{out}$ [kJ]
800	219.9	175.7
900	183.7	144.7
1000	150.6	117
1100	120	91.84
1200	91.23	68.85
1300	64.08	47.65
1400	38.2	27.98
1500	13.32	9.605
1553	0.4645	0.3319



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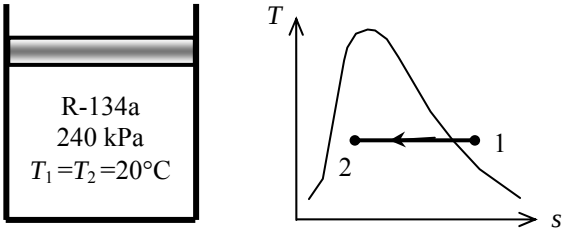
7-46 R-134a undergoes an isothermal process in a closed system. The work and heat transfer are to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$



The initial state properties are

$$\left.\begin{array}{l} P_1 = 240\ \text{kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} u_1 = 246.74\ \text{kJ/kg} \\ s_1 = 1.0134\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A -13)}$$

For this isothermal process, the final state properties are (Table A-11)

$$\left.\begin{array}{l} T_2 = T_1 = 20°\text{C} \\ x_2 = 0.20 \end{array}\right\} \begin{array}{l} u_2 = u_f + x_2 u_{fg} = 78.86 + (0.20)(162.16) = 111.29\ \text{kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 0.30063 + (0.20)(0.62172) = 0.42497\ \text{kJ/kg}\cdot\text{K} \end{array}$$

The heat transfer is determined from

$$q_{\text{in}} = T_0(s_2 - s_1) = (293\ \text{K})(0.42497 - 1.0134)\ \text{kJ/kg}\cdot\text{K} = -172.4\ \text{kJ/kg}$$

The negative sign shows that the heat is actually transferred from the system. That is,

$$q_{\text{out}} = \mathbf{172.4\ kJ/kg}$$

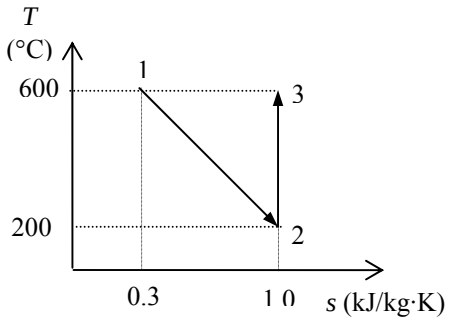
The work required is determined from the energy balance to be

$$w_{\text{in}} = q_{\text{out}} + (u_2 - u_1) = 172.4\ \text{kJ/kg} + (111.29 - 246.74)\ \text{kJ/kg} = \mathbf{36.95\ kJ/kg}$$

7-47 The heat transfer during the process shown in the figure is to be determined.

Assumptions The process is reversible.

Analysis No heat is transferred during the process 2-3 since the area under process line is zero. Then the heat transfer is equal to the area under the process line 1-2:



$$q_{12} = \int_1^2 T\,ds = \text{Area} = \frac{T_1 + T_2}{2}(s_2 - s_1)$$

$$= \frac{(600 + 273)\text{K} + (200 + 273)\text{K}}{2}(1.0 - 0.3)\text{kJ/kg}\cdot\text{K}$$

$$= \mathbf{471\ kJ/kg}$$

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7-48E The heat transfer during the process shown in the figure is to be determined.

Assumptions The process is reversible.

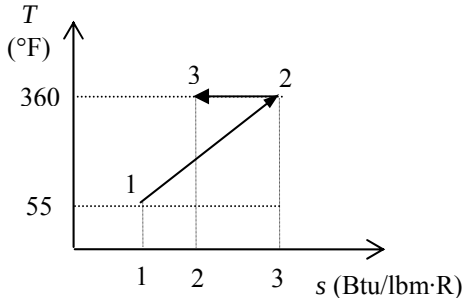
Analysis Heat transfer is equal to the sum of the areas under the process 1-2 and 2-3.

$$q_{12} = \int_1^2 Tds + \int_2^3 Tds = \frac{T_1 + T_2}{2}(s_2 - s_1) + T_2(s_3 - s_2)$$

$$= \frac{(55 + 460)\text{R} + (360 + 460)\text{R}}{2}(3.0 - 1.0)\text{Btu/lbm} \cdot \text{R}$$

$$+ \left[(360 + 460)\text{R}\right](2.0 - 3.0)\text{Btu/lbm} \cdot \text{R}$$

$$= \mathbf{515\ Btu/lbm}$$



7-49 The heat transfer during the process shown in the figure is to be determined.

Assumptions The process is reversible.

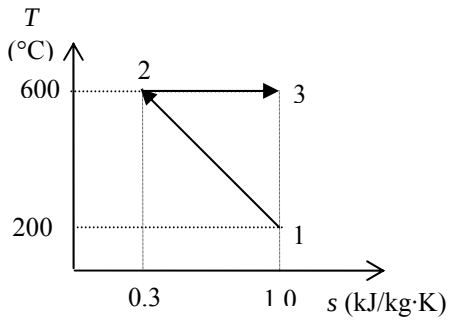
Analysis Heat transfer is equal to the sum of the areas under the process 1-2 and 2-3.

$$q_{12} = \int_1^2 Tds + \int_2^3 Tds = \frac{T_1 + T_2}{2}(s_2 - s_1) + T_2(s_3 - s_2)$$

$$= \frac{(200 + 273)\text{K} + (600 + 273)\text{K}}{2}(0.3 - 1.0)\text{kJ/kg} \cdot \text{K}$$

$$+ \left[(600 + 273)\text{K}\right](1.0 - 0.3)\text{kJ/kg} \cdot \text{K}$$

$$= \mathbf{140\ kJ/kg}$$



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7-50 Steam is expanded in an isentropic turbine. The work produced is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic).

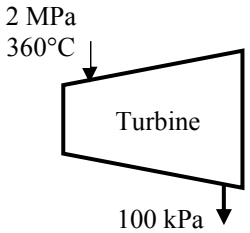
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

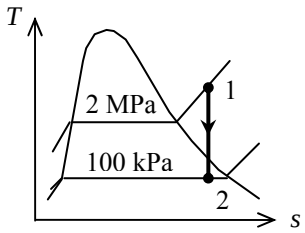
$$\dot{m}h_1 = \dot{m}h_2 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$



The inlet state properties are

$$\left.\begin{array}{l} P_1 = 2\ \text{MPa} \\ T_1 = 360°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3159.9\ \text{kJ/kg} \\ s_1 = 6.9938\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 6)}$$



For this isentropic process, the final state properties are (Table A-5)

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ s_2 = s_1 = 6.9938\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{6.9938 - 1.3028}{6.0562} = 0.9397 \\ h_2 = h_f + x_2 h_{fg} = 417.51 + (0.9397)(2257.5) = 2538.9\ \text{kJ/kg} \end{array}$$

Substituting,

$$w_{\text{out}} = h_1 - h_2 = (3159.9 - 2538.9)\ \text{kJ/kg} = \mathbf{621.0\ kJ/kg}$$

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7-51 Steam is expanded in an isentropic turbine. The work produced is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic).

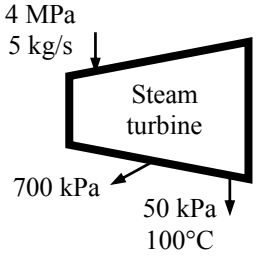
Analysis There is one inlet and two exits. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{\text{out}}$$

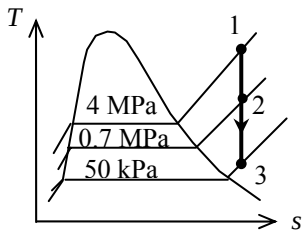
$$\dot{W}_{\text{out}} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$



From a mass balance,

$$\dot{m}_2 = 0.05\dot{m}_1 = (0.05)(5 \text{ kg/s}) = 0.25 \text{ kg/s}$$

$$\dot{m}_3 = 0.95\dot{m}_1 = (0.95)(5 \text{ kg/s}) = 4.75 \text{ kg/s}$$



Noting that the expansion process is isentropic, the enthalpies at three states are determined as follows:

$$\left.\begin{array}{l} P_3 = 50 \text{ kPa} \\ T_3 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 2682.4 \text{ kJ/kg} \\ s_3 = 7.6953 \text{ kJ/kg} \cdot \text{K} \end{array} \quad \text{(Table A - 6)}$$

$$\left.\begin{array}{l} P_1 = 4 \text{ MPa} \\ s_1 = s_3 = 7.6953 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_1 = 3979.3 \text{ kJ/kg} \quad \text{(Table A - 6)}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ s_2 = s_3 = 7.6953 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_2 = 3309.1 \text{ kJ/kg} \quad \text{(Table A - 6)}$$

Substituting,

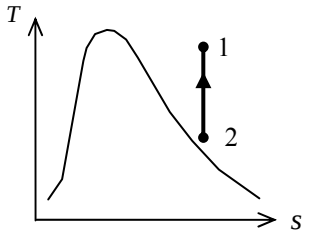
$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3 \\ &= (5 \text{ kg/s})(3979.3 \text{ kJ/kg}) - (0.25 \text{ kg/s})(3309.1 \text{ kJ/kg}) - (4.75 \text{ kg/s})(2682.4 \text{ kJ/kg}) \\ &= \mathbf{6328 \text{ kW}} \end{aligned}$$

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7-52 Water is compressed in a closed system during which the entropy remains constant. The final temperature and the work required are to be determined.

Analysis The initial state is superheated vapor and thus

$$\left.\begin{array}{l} P_1 = 70\ \text{kPa} \\ T_1 = 100°\text{C} \end{array}\right\} \begin{array}{l} u_1 = 2509.4\ \text{kJ/kg} \\ s_1 = 7.5344\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 6)}$$



The entropy is constant during the process. The properties at the exit state are

$$\left.\begin{array}{l} P_2 = 4000\ \text{kPa} \\ s_2 = s_1 = 7.5344\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} u_2 = 3396.5\ \text{kJ/kg} \\ T_2 = \mathbf{664°C} \end{array} \quad \text{(Table A - 6)}$$

To determine the work done, we take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$w_{\text{in}} = \Delta u = u_2 - u_1 \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

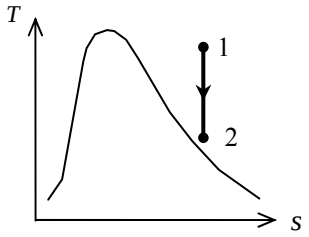
Substituting,

$$w_{\text{in}} = u_2 - u_1 = (3396.5 - 2509.4)\text{kJ/kg} = \mathbf{887.1\ kJ/kg}$$

7-53 Refrigerant-134a is expanded in a closed system during which the entropy remains constant. The heat transfer and the work production are to be determined.

Analysis The initial state is superheated vapor and thus

$$\left.\begin{array}{l} P_1 = 600\ \text{kPa} \\ T_1 = 30°\text{C} \end{array}\right\} \begin{array}{l} u_1 = 249.22\ \text{kJ/kg} \\ s_1 = 0.9499\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 13)}$$



The entropy is constant during the process. The properties at the exit state are

$$\left.\begin{array}{l} P_2 = 140\ \text{kPa} \\ s_2 = s_1 = 0.9499\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} u_2 = 220.75\ \text{kJ/kg} \quad \text{(Table A - 13)}$$

Since the process is isentropic, and thus the heat transfer is zero.

$$Q = \mathbf{0\ kJ}$$

To determine the work done, we take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

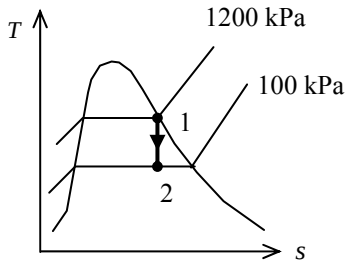
Substituting,

$$W_{\text{out}} = m(u_1 - u_2) = (0.5\ \text{kg})(249.22 - 220.75)\text{kJ/kg} = \mathbf{14.2\ kJ}$$

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7-54 Refrigerant-134a is expanded in an adiabatic turbine reversibly. The process is to be sketched on the *T*-*s* diagram and the volume flow rate at the exit is to be determined.

Assumptions The process is steady.



Analysis (b) Noting that the process is isentropic (constant entropy) the inlet and exit states are obtained from R-134a tables (Table A-12) as follows:

$$\left.\begin{array}{l} P_1 = 1200\ \text{kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 273.87\ \text{kJ/kg} \\ s_1 = 0.9130\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ s_2 = s_1 = 0.9130\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{0.9130 - 0.07188}{0.87995} = 0.9559 \\ h_2 = h_f + xh_{fg} = 17.28 + 0.9559 \times 217.16 = 224.87\ \text{kJ/kg} \\ \nu_2 = \nu_f + x\nu_{fg} = 0.0007259 + 0.9559 \times (0.19254 - 0.0007259) = 0.1841\ \text{m}^3/\text{kg} \end{array}$$

We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the turbine, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2 + \dot{W}_{\text{out}} \qquad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$

Solving for the mass flow rate and substituting,

$$\dot{m} = \frac{\dot{W}_{\text{out}}}{h_1 - h_2} = \frac{100\ \text{kW}}{(273.87 - 224.87)\text{kJ/kg}} = 2.041\ \text{kg/s}$$

The volume flow rate at the exit is then,

$$\dot{V}_2 = \dot{m}\nu_2 = (2.041\ \text{kg/s})(0.1841\ \text{m}^3/\text{kg}) = \mathbf{0.376\ m^3/s}$$

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7-55 Water vapor is expanded adiabatically in a piston-cylinder device. The entropy change is to be determined and it is to be discussed if this process is realistic.

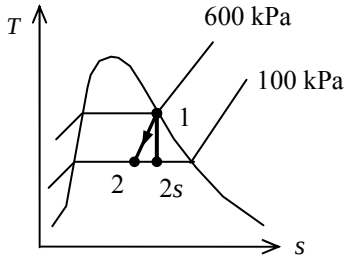
Analysis (a) The properties at the initial state are

$$\left.\begin{array}{l} P_1 = 600\ \text{kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} u_1 = 2566.8\ \text{kJ/kg} \\ s_1 = 6.7593\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 5)}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$



Solving for the final state internal energy,

$$u_2 = u_1 + \frac{W_{\text{out}}}{m} = 2566.8\ \text{kJ/kg} + \frac{700\ \text{kJ}}{2\ \text{kg}} = 2216.8\ \text{kJ/kg}$$

The entropy at the final state is (from Table A-5)

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ u_2 = 2216.8\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{u_2 - u_f}{u_{fg}} = \dfrac{2216.8 - 417.40}{2088.2} = 0.8617 \\ s_2 = s_f + x s_{fg} = 1.3028 + 0.8617 \times 6.0562 = 6.5215\ \text{kJ/kg}\cdot\text{K} \end{array}$$

The entropy change is

$$\Delta s = s_2 - s_1 = 6.5215 - 6.7593 = \mathbf{-0.238\ kJ/kg\cdot K}$$

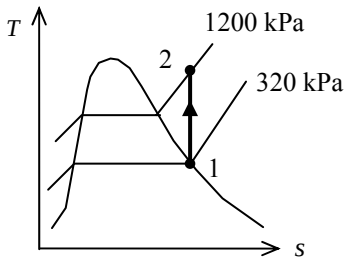
(b) The process is not realistic since entropy cannot decrease during an adiabatic process. In the limiting case of a reversible (and adiabatic) process, the entropy would remain constant.

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7-56 Refrigerant-134a is compressed in an adiabatic compressor reversibly. The process is to be sketched on the *T*-*s* diagram and the volume flow rate at the inlet is to be determined.

Assumptions The process is steady.

Analysis (b) Noting that the process is isentropic (constant entropy) the inlet and exit states are obtained from R-134a tables (Tables A-12 and A-13) as follows:



$$\left.\begin{array}{l} P_1 = 320\ \text{kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 251.88\ \text{kJ/kg} \\ s_1 = 0.9301\ \text{kJ/kg}\cdot\text{K} \\ \nu_1 = 0.06360\ \text{m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1200\ \text{kPa} \\ s_2 = s_1 = 0.9301\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} h_2 = 279.35\ \text{kJ/kg}$$

We take the compressor as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the compressor, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\not{0}\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2 \qquad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Solving for the mass flow rate and substituting,

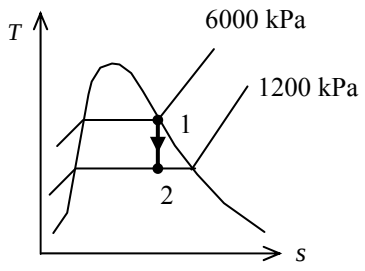
$$\dot{m} = \frac{\dot{W}_{\text{in}}}{h_2 - h_1} = \frac{100\ \text{kW}}{(279.35 - 251.88)\text{kJ/kg}} = 3.640\ \text{kg/s}$$

The volume flow rate at the inlet is then,

$$\dot{V}_1 = \dot{m}\nu_1 = (3.640\ \text{kg/s})(0.06360\ \text{m}^3/\text{kg}) = \mathbf{0.232\ m^3/s}$$

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7-57 Steam enters a nozzle at a specified state and leaves at a specified pressure. The process is to be sketched on the *T*-*s* diagram and the maximum outlet velocity is to be determined.



Analysis (b) The inlet state properties are

$$\left.\begin{array}{l} P_1 = 6000\ \text{kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 2784.6\ \text{kJ/kg} \\ s_1 = 5.8902\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 5)}$$

For the maximum velocity at the exit, the entropy will be constant during the process. The exit state enthalpy is (Table A-6)

$$\left.\begin{array}{l} P_2 = 1200\ \text{kPa} \\ s_2 = s_1 = 5.8902\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{5.8902 - 2.2159}{4.3058} = 0.8533 \\ h_2 = h_f + x h_{fg} = 798.33 + 0.8533 \times 1985.4 = 2492.5\ \text{kJ/kg} \end{array}$$

We take the nozzle as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the nozzle, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\not{0}\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) \qquad (\text{since } \dot{W} \cong \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$h_1 - h_2 = \left(\frac{V_2^2 - V_1^2}{2}\right)$$

Solving for the exit velocity and substituting,

$$h_1 - h_2 = \left(\frac{V_2^2 - V_1^2}{2}\right)$$

$$V_2 = \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} = \left[(0\ \text{m/s})^2 + 2(2784.6 - 2492.5)\ \text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)\right]^{0.5}$$

$$= \mathbf{764.3\ m/s}$$

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7-58 Heat is added to a pressure cooker that is maintained at a specified pressure. The minimum entropy change of the thermal-energy reservoir supplying this heat is to be determined.

Assumptions **1** Only water vapor escapes through the pressure relief valve.

Analysis According to the conservation of mass principle,

$$\frac{dm_{CV}}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m}$$
$$\frac{dm}{dt} = -\dot{m}_{out}$$

An entropy balance adapted to this system becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} + \dot{m}_{out} s \geq 0$$

When this is combined with the mass balance, it becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} - s\frac{dm}{dt} \geq 0$$

Multiplying by *dt* and integrating the result yields

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out}(m_2 - m_1) \geq 0$$

The properties at the initial and final states are (from Table A-5 at P_1 = 175 kPa and P_2 = 150 kPa)

$$v_1 = v_f + x v_{fg} = 0.001057 + (0.10)(1.0037 - 0.001057) = 0.1013 \text{ m}^3/\text{kg}$$
$$s_1 = s_f + x s_{fg} = 1.4850 + (0.10)(5.6865) = 2.0537 \text{ kJ/kg} \cdot \text{K}$$
$$v_2 = v_f + x v_{fg} = 0.001053 + (0.40)(1.1594 - 0.001053) = 0.4644 \text{ m}^3/\text{kg}$$
$$s_2 = s_f + x s_{fg} = 1.4337 + (0.40)(5.7894) = 3.7494 \text{ kJ/kg} \cdot \text{K}$$

The initial and final masses are

$$m_1 = \frac{V}{v_1} = \frac{0.020 \text{ m}^3}{0.01013 \text{ m}^3/\text{kg}} = 0.1974 \text{ kg}$$

$$m_2 = \frac{V}{v_2} = \frac{0.020 \text{ m}^3}{0.4644 \text{ m}^3/\text{kg}} = 0.04307 \text{ kg}$$

The entropy of escaping water vapor is

$$s_{out} = s_{g@150 \text{ kPa}} = 7.2231 \text{ kJ/kg} \cdot \text{K}$$

Substituting,

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out}(m_2 - m_1) \geq 0$$
$$\Delta S_{surr} + (0.04307)(3.7494) - (0.1974)(2.0537) - (7.2231)(0.04307 - 0.1974) \geq 0$$
$$\Delta S_{surr} + 0.8708 \geq 0$$

The entropy change of the thermal energy reservoir must then satisfy

$$\Delta S_{surr} \geq \mathbf{-0.8708 \text{ kJ/K}}$$

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7-59 Heat is added to a pressure cooker that is maintained at a specified pressure. Work is also done on water. The minimum entropy change of the thermal-energy reservoir supplying this heat is to be determined.

***Assumptions* 1** Only water vapor escapes through the pressure relief valve.

Analysis According to the conservation of mass principle,

$$\frac{dm_{CV}}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m}$$
$$\frac{dm}{dt} = -\dot{m}_{out}$$

An entropy balance adapted to this system becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} + \dot{m}_{out} s \geq 0$$

When this is combined with the mass balance, it becomes

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} - s\frac{dm}{dt} \geq 0$$

Multiplying by *dt* and integrating the result yields

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out}(m_2 - m_1) \geq 0$$

Note that work done on the water has no effect on this entropy balance since work transfer does not involve any entropy transfer. The properties at the initial and final states are (from Table A-5 at P_1 = 175 kPa and P_2 = 150 kPa)

$$v_1 = v_f + x v_{fg} = 0.001057 + (0.10)(1.0037 - 0.001057) = 0.1013 \text{ m}^3/\text{kg}$$
$$s_1 = s_f + x s_{fg} = 1.4850 + (0.10)(5.6865) = 2.0537 \text{ kJ/kg} \cdot \text{K}$$
$$v_2 = v_f + x v_{fg} = 0.001053 + (0.40)(1.1594 - 0.001053) = 0.4644 \text{ m}^3/\text{kg}$$
$$s_2 = s_f + x s_{fg} = 1.4337 + (0.40)(5.7894) = 3.7494 \text{ kJ/kg} \cdot \text{K}$$

The initial and final masses are

$$m_1 = \frac{V}{v_1} = \frac{0.020 \text{ m}^3}{0.01013 \text{ m}^3/\text{kg}} = 0.1974 \text{ kg}$$

$$m_2 = \frac{V}{v_2} = \frac{0.020 \text{ m}^3}{0.4644 \text{ m}^3/\text{kg}} = 0.04307 \text{ kg}$$

The entropy of escaping water vapor is

$$s_{out} = s_{g@150 \text{ kPa}} = 7.2231 \text{ kJ/kg} \cdot \text{K}$$

Substituting,

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out}(m_2 - m_1) \geq 0$$
$$\Delta S_{surr} + (0.04307)(3.7494) - (0.1974)(2.0537) - (7.2231)(0.04307 - 0.1974) \geq 0$$
$$\Delta S_{surr} + 0.8708 \geq 0$$

The entropy change of the thermal energy reservoir must then satisfy

$$\Delta S_{surr} \geq \mathbf{-0.8708 \text{ kJ/K}}$$

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7-60 A cylinder is initially filled with saturated water vapor mixture at a specified temperature. Steam undergoes a reversible heat addition and an isentropic process. The processes are to be sketched and heat transfer for the first process and work done during the second process are to be determined.

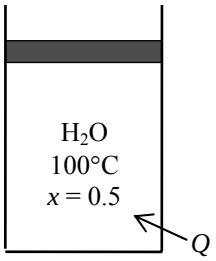
Assumptions **1** The kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself is negligible. **3** Both processes are reversible.

Analysis (b) From the steam tables (Tables A-4 through A-6),

$$\left.\begin{matrix} T_1 = 100°\text{C} \\ x = 0.5 \end{matrix}\right\} h_1 = h_f + xh_{fg} = 419.17 + (0.5)(2256.4) = 1547.4 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_2 = 100°\text{C} \\ x_2 = 1 \end{matrix}\right\} \begin{matrix} h_2 = h_g = 2675.6 \text{ kJ/kg} \\ u_2 = u_g = 2506.0 \text{ kJ/kg} \\ s_2 = s_g = 7.3542 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$

$$\left.\begin{matrix} P_3 = 15 \text{ kPa} \\ s_3 = s_2 \end{matrix}\right\} u_3 = 2247.9 \text{ kJ/kg}$$



We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

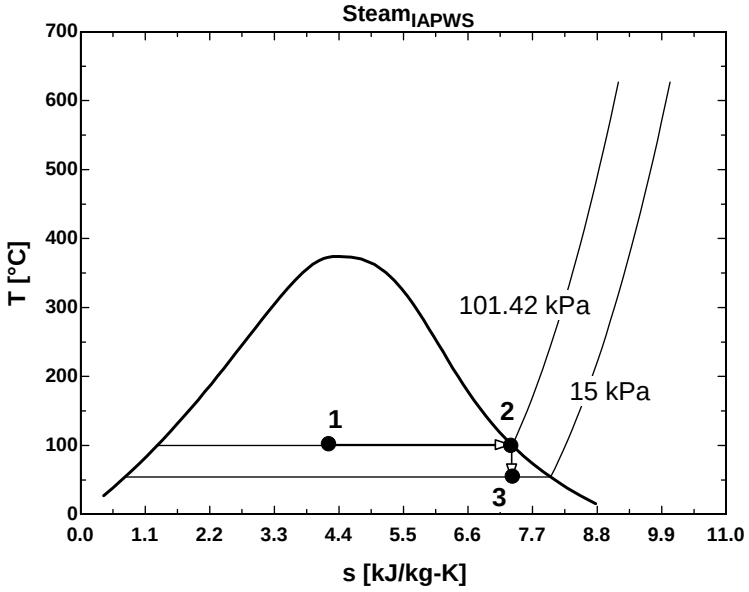
$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

For process 1-2, it reduces to

$$Q_{12,\text{in}} = m(h_2 - h_1) = (5 \text{ kg})(2675.6 - 1547.4)\text{kJ/kg} = \mathbf{5641\,kJ}$$

(c) For process 2-3, it reduces to

$$W_{23,\text{b,out}} = m(u_2 - u_3) = (5 \text{ kg})(2506.0 - 2247.9)\text{kJ/kg} = \mathbf{1291\,kJ}$$

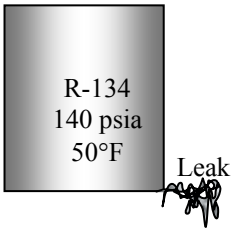


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7-61E An insulated rigid can initially contains R-134a at a specified state. A crack develops, and refrigerant escapes slowly. The final mass in the can is to be determined when the pressure inside drops to a specified value.

Assumptions **1** The can is well-insulated and thus heat transfer is negligible. **2** The refrigerant that remains in the can underwent a reversible adiabatic process.

Analysis Noting that for a reversible adiabatic (i.e., isentropic) process, $s_1 = s_2$, the properties of the refrigerant in the can are (Tables A-11E through A-13E)



$$\left.\begin{array}{l} P_1 = 140 \text{ psia} \\ T_1 = 50°\text{F} \end{array}\right\} s_1 \cong s_{f@50°\text{F}} = 0.06039 \text{ Btu/lbm} \cdot \text{R}$$

$$\left.\begin{array}{l} P_2 = 30 \text{ psia} \\ s_2 = s_1 \end{array}\right\} \quad x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{0.06039 - 0.03793}{0.1859} = 0.1208$$

$$v_2 = v_f + x_2 v_{fg} = 0.01209 + (0.1208)(1.5492 - 0.01209) = 0.1978 \text{ ft}^3/\text{lbm}$$

Thus the final mass of the refrigerant in the can is

$$m = \frac{V}{v_2} = \frac{0.8 \text{ ft}^3}{0.1978 \text{ ft}^3/\text{lbm}} = \mathbf{4.04 \text{ lbm}}$$

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7-62E An electric windshield defroster used to remove ice is considered. The electrical energy required and the minimum temperature of the defroster are to be determined.

Assumptions No no heat is transferred from the defroster or ice to the surroundings.

Analysis The conservation of mass principle is

$$\frac{dm_{cv}}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m}$$

which reduces to

$$\frac{dm_{cv}}{dt} = -\dot{m}_{\text{out}}$$

while the the first law reduces to

$$-W_{\text{out}} = \frac{d(mu)_{cv}}{dt} + \dot{m}_{\text{out}} h_{\text{out}}$$

Combining these two expressions yield

$$W_{\text{out}} = h_{\text{out}} \frac{dm_{cv}}{dt} - \frac{d(mu)_{cv}}{dt}$$

When this is multiplied by *dt* and integrated from time when the ice layer is present until it is removed ($m = 0$) gives

$$W_{\text{out}} = h_{\text{out}}(-m_i) + (mu)_i$$

The original mass of the ice layer is

$$m_i = \frac{\boldsymbol{V}}{\boldsymbol{v}} = \frac{tA}{\boldsymbol{v}}$$

The work required per unit of windshield area is then

$$\frac{W_{\text{out}}}{A} = \frac{t}{\boldsymbol{v}}(u_i - h_{\text{out}}) = \frac{t}{\boldsymbol{v}}(u_i - u_f) = \frac{t}{\boldsymbol{v}} u_{if} = \frac{(0.25/12)\text{ ft}}{0.01602\text{ ft}^3/\text{lbm}}(-144\text{ Btu/lbm}) = -187.3\text{ Btu/ft}^2$$

That is,

$$W_{\text{in}} = \mathbf{187.3\ Btu/ft^2}$$

The second law as stated by Clasius tells us that the temperature of the defroster cannot be lower than the temperature of the ice being melted. Then,

$$T_{\text{min}} = \mathbf{32°F}$$

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Entropy Change of Incompressible Substances

7-63C No, because entropy is not a conserved property.

7-64 A hot copper block is dropped into water in an insulated tank. The final equilibrium temperature of the tank and the total entropy change are to be determined.

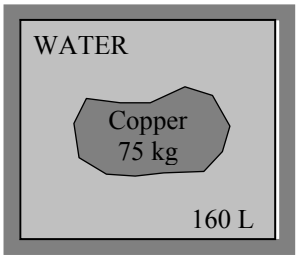
Assumptions **1** Both the water and the copper block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer.

Properties The density and specific heat of water at 25°C are $\rho = 997$ kg/m^3 and $c_p = 4.18$ kJ/kg.°C. The specific heat of copper at 27°C is $c_p = 0.386$ kJ/kg.°C (Table A-3).

Analysis We take the entire contents of the tank, water + copper block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$



or,

$$\Delta U_{\text{Cu}} + \Delta U_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{Cu}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

where

$$m_{\text{water}} = \rho V = (997 \text{ kg/m}^3)(0.160 \text{ m}^3) = 159.5 \text{ kg}$$

Using specific heat values for copper and liquid water at room temperature and substituting,

$$(75 \text{ kg})(0.386 \text{ kJ/kg} \cdot \text{°C})(T_2 - 110)\text{°C} + (159.5 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{°C})(T_2 - 15)\text{°C} = 0$$

$$T_2 = \mathbf{19.0°C} = 292 \text{ K}$$

The entropy generated during this process is determined from

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (75 \text{ kg})(0.386 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{292.0 \text{ K}}{383 \text{ K}}\right) = -7.85 \text{ kJ/K}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (159.5 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln\left(\frac{292.0 \text{ K}}{288 \text{ K}}\right) = 9.20 \text{ kJ/K}$$

Thus,

$$\Delta S_{\text{total}} = \Delta S_{\text{copper}} + \Delta S_{\text{water}} = -7.85 + 9.20 = \mathbf{1.35 \text{ kJ/K}}$$

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7-65 Computer chips are cooled by placing them in saturated liquid R-134a. The entropy changes of the chips, R-134a, and the entire system are to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved. **3** There is no heat transfer between the system and the surroundings.

Analysis (a) The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = [m(u_2 - u_1)]_{\text{chips}} + [m(u_2 - u_1)]_{\text{R-134a}}$$

$$[m(u_1 - u_2)]_{\text{chips}} = [m(u_2 - u_1)]_{\text{R-134a}}$$

The heat released by the chips is

$$Q_{\text{chips}} = mc(T_1 - T_2) = (0.010\,\text{kg})(0.3\,\text{kJ/kg}\cdot\text{K})[20 - (-40)]\,\text{K} = 0.18\,\text{kJ}$$

The mass of the refrigerant vaporized during this heat exchange process is

$$m_{g,2} = \frac{Q_{\text{R-134a}}}{u_g - u_f} = \frac{Q_{\text{R-134a}}}{u_{fg\,@-40°\text{C}}} = \frac{0.18\,\text{kJ}}{207.40\,\text{kJ/kg}} = 0.0008679\,\text{kg}$$

Only a small fraction of R-134a is vaporized during the process. Therefore, the temperature of R-134a remains constant during the process. The change in the entropy of the R-134a is (at -40°F from Table A-11)

$$\begin{aligned}\Delta S_{\text{R-134a}} &= m_{g,2}s_{g,2} + m_{f,2}s_{f,2} - m_{f,1}s_{f,1}\\ &= (0.0008679)(0.96866) + (0.005 - 0.0008679)(0) - (0.005)(0)\\ &= \mathbf{0.000841\,kJ/K}\end{aligned}$$

(b) The entropy change of the chips is

$$\Delta S_{\text{chips}} = mc\ln\frac{T_2}{T_1} = (0.010\,\text{kg})(0.3\,\text{kJ/kg}\cdot\text{K})\ln\frac{(-40+273)\text{K}}{(20+273)\text{K}} = \mathbf{-0.000687\,kJ/K}$$

(c) The total entropy change is

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{R-134a}} + \Delta S_{\text{chips}} = 0.000841 + (-0.000687) = \mathbf{0.000154\,kJ/K}$$

The positive result for the total entropy change (i.e., entropy generation) indicates that this process is possible.

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7-66 A hot iron block is dropped into water in an insulated tank. The total entropy change during this process is to be determined.

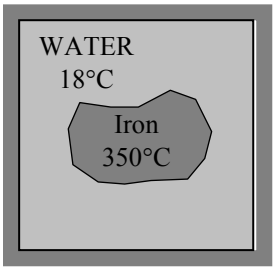
Assumptions **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer. **4** The water that evaporates, condenses back.

Properties The specific heat of water at 25°C is c_p = 4.18 kJ/kg.°C. The specific heat of iron at room temperature is c_p = 0.45 kJ/kg.°C (Table A-3).

Analysis We take the entire contents of the tank, water + iron block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$



or,

$$\Delta U_{\text{iron}} + \Delta U_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{iron}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(25\ \text{kg})(0.45\ \text{kJ/kg}\cdot\text{K})(T_2 - 350^\circ\text{C}) + (100\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K})(T_2 - 18^\circ\text{C}) = 0$$

$$T_2 = \mathbf{26.7°C}$$

The entropy generated during this process is determined from

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (25\ \text{kg})(0.45\ \text{kJ/kg}\cdot\text{K}) \ln\left(\frac{299.7\ \text{K}}{623\ \text{K}}\right) = -8.232\ \text{kJ/K}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (100\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K}) \ln\left(\frac{299.7\ \text{K}}{291\ \text{K}}\right) = 12.314\ \text{kJ/K}$$

Thus,

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{water}} = -8.232 + 12.314 = \mathbf{4.08\ kJ/K}$$

Discussion The results can be improved somewhat by using specific heats at average temperature.

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7-67 An aluminum block is brought into contact with an iron block in an insulated enclosure. The final equilibrium temperature and the total entropy change for this process are to be determined.

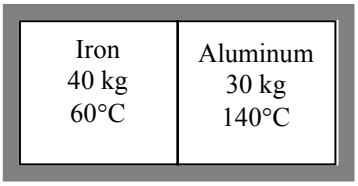
Assumptions **1** Both the aluminum and the iron block are incompressible substances with constant specific heats. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The system is well-insulated and thus there is no heat transfer.

Properties The specific heat of aluminum at the anticipated average temperature of 400 K is c_p = 0.949 kJ/kg.°C. The specific heat of iron at room temperature (the only value available in the tables) is c_p = 0.45 kJ/kg.°C (Table A-3).

Analysis We take the iron+aluminum blocks as the system, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$



or,

$$\Delta U_{\text{alum}} + \Delta U_{\text{iron}} = 0$$
$$[mc(T_2 - T_1)]_{\text{alum}} + [mc(T_2 - T_1)]_{\text{iron}} = 0$$

Substituting,

$$(30\text{ kg})(0.949\text{ kJ/kg}\cdot\text{K})(T_2 - 140°\text{C}) + (40\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})(T_2 - 60°\text{C}) = 0$$

$$T_2 = \mathbf{109°C} = 382\text{ K}$$

The total entropy change for this process is determined from

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (40\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{382\text{ K}}{333\text{ K}}\right) = 2.472\text{ kJ/K}$$

$$\Delta S_{\text{alum}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (30\text{ kg})(0.949\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{382\text{ K}}{413\text{ K}}\right) = -2.221\text{ kJ/K}$$

Thus,

$$\Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{alum}} = 2.472 - 2.221 = \mathbf{0.251\,kJ/K}$$

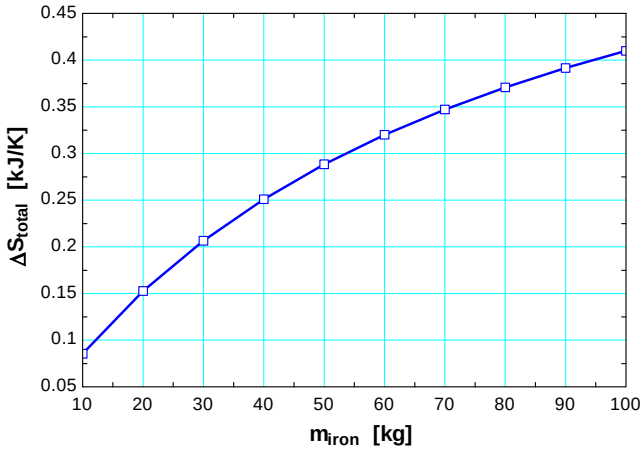
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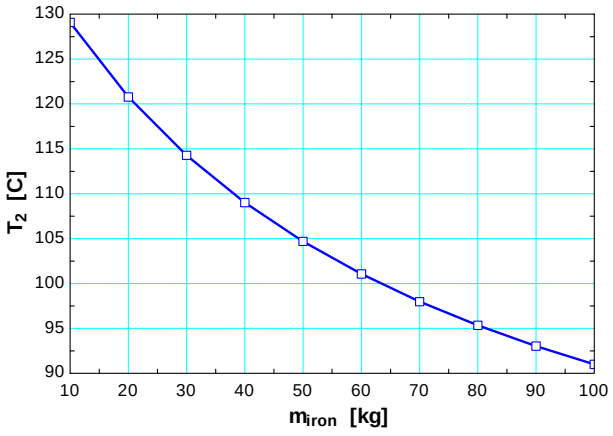
7-68 Problem 7-67 is reconsidered. The effect of the mass of the iron block on the final equilibrium temperature and the total entropy change for the process is to be studied. The mass of the iron is to vary from 10 to 100 kg. The equilibrium temperature and the total entropy change are to be plotted as a function of iron mass.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
T_1_iron = 60 [C]
"m_iron = 40 [kg]"
T_1_al = 140 [C]
m_al = 30 [kg]
C_al = 0.949 [kJ/kg-K]  "FromTable A-3 at the anticipated average temperature of 450 K."
C_iron= 0.45 [kJ/kg-K]  "FromTable A-3 at room temperature, the only value available."
"Analysis: "
" Treat the iron plus aluminum as a closed system, with no heat transfer in, no work out, neglect changes in KE and PE of the system.  "
"The final temperature is found from the energy balance."
E_in - E_out = DELTAE_sys
E_out = 0
E_in = 0
DELTAE_sys = m_iron*DELTAu_iron  + m_al*DELTAu_al
DELTAu_iron = C_iron*(T_2_iron - T_1_iron)
DELTAu_al = C_al*(T_2_al - T_1_al)
"the iron and aluminum reach thermal equilibrium:"
T_2_iron = T_2
T_2_al = T_2
DELTAS_iron = m_iron*C_iron*ln((T_2_iron+273) / (T_1_iron+273))
DELTAS_al = m_al*C_al*ln((T_2_al+273) / (T_1_al+273))
DELTAS_total = DELTAS_iron + DELTAS_al
```

ΔS_{total} kJ/kg]	m_{iron} [kg]	T_2 [C]
0.08547	10	129.1
0.1525	20	120.8
0.2066	30	114.3
0.2511	40	109
0.2883	50	104.7
0.32	60	101.1
0.3472	70	97.98
0.3709	80	95.33
0.3916	90	93.02
0.41	100	91





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7-69 An iron block and a copper block are dropped into a large lake. The total amount of entropy change when both blocks cool to the lake temperature is to be determined.

Assumptions **1** The water, the iron block and the copper block are incompressible substances with constant specific heats at room temperature. **2** Kinetic and potential energies are negligible.

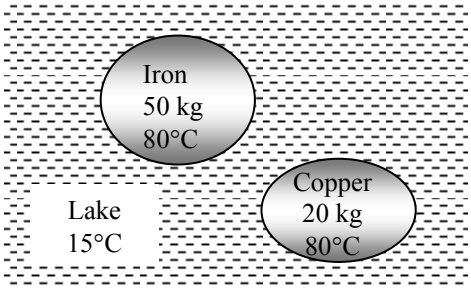
Properties The specific heats of iron and copper at room temperature are $c_{\text{iron}} = 0.45$ kJ/kg.°C and $c_{\text{copper}} = 0.386$ kJ/kg.°C (Table A-3).

Analysis The thermal-energy capacity of the lake is very large, and thus the temperatures of both the iron and the copper blocks will drop to the lake temperature (15°C) when the thermal equilibrium is established. Then the entropy changes of the blocks become

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (50\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288\text{ K}}{353\text{ K}}\right) = -4.579\text{ kJ/K}$$

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (20\text{ kg})(0.386\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288\text{ K}}{353\text{ K}}\right) = -1.571\text{ kJ/K}$$

We take both the iron and the copper blocks, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = \Delta U_{\text{iron}} + \Delta U_{\text{copper}}$$

or,

$$Q_{\text{out}} = [mc(T_1 - T_2)]_{\text{iron}} + [mc(T_1 - T_2)]_{\text{copper}}$$

Substituting,

$$Q_{\text{out}} = (50\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})(353 - 288)\text{K} + (20\text{ kg})(0.386\text{ kJ/kg}\cdot\text{K})(353 - 288)\text{K}$$
$$= 1964\text{ kJ}$$

Thus,

$$\Delta S_{\text{lake}} = \frac{Q_{\text{lake,in}}}{T_{\text{lake}}} = \frac{1964\text{ kJ}}{288\text{ K}} = 6.820\text{ kJ/K}$$

Then the total entropy change for this process is

$$\Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{copper}} + \Delta S_{\text{lake}} = -4.579 - 1.571 + 6.820 = \mathbf{0.670\,kJ/K}$$

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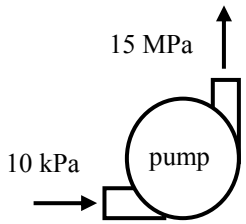
7-70 An adiabatic pump is used to compress saturated liquid water in a reversible manner. The work input is to be determined by different approaches.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible.

Analysis The properties of water at the inlet and exit of the pump are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10\ \text{kPa} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 191.81\ \text{kJ/kg} \\ s_1 = 0.6492\ \text{kJ/kg} \\ \nu_1 = 0.001010\ \text{m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 15\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} h_2 = 206.90\ \text{kJ/kg} \\ \nu_2 = 0.001004\ \text{m}^3/\text{kg} \end{array}$$



(*a*) Using the entropy data from the compressed liquid water table

$$w_{\text{P}} = h_2 - h_1 = 206.90 - 191.81 = \mathbf{15.10\ kJ/kg}$$

(*b*) Using inlet specific volume and pressure values

$$w_{\text{P}} = \nu_1 (P_2 - P_1) = (0.001010\ \text{m}^3/\text{kg})(15{,}000 - 10)\text{kPa} = \mathbf{15.14\ kJ/kg}$$

Error = **0.3%**

(*c*) Using average specific volume and pressure values

$$w_{\text{P}} = \nu_{\text{avg}} (P_2 - P_1) = \left[1/2(0.001010 + 0.001004)\ \text{m}^3/\text{kg}\right](15{,}000 - 10)\text{kPa} = \mathbf{15.10\ kJ/kg}$$

Error = **0%**

Discussion The results show that any of the method may be used to calculate reversible pump work.

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Entropy Changes of Ideal Gases

7-71C No. The entropy of an ideal gas depends on the pressure as well as the temperature.

7-72C The entropy of a gas *can* change during an isothermal process since entropy of an ideal gas depends on the pressure as well as the temperature.

7-73C The entropy change relations of an ideal gas simplify to

$\Delta s = c_p \ln(T_2/T_1)$ for a constant pressure process

and $\Delta s = c_v \ln(T_2/T_1)$ for a constant volume process.

Noting that $c_p > c_v$, the entropy change will be larger for a constant pressure process.

7-74 For ideal gases, $c_p = c_v + R$ and

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow \frac{v_2}{v_1} = \frac{T_2 P_1}{T_1 P_2}$$

Thus,

$$\begin{aligned} s_2 - s_1 &= c_v \ln\left(\frac{T_2}{T_1}\right) + R \ln\left(\frac{v_2}{v_1}\right) \\ &= c_v \ln\left(\frac{T_2}{T_1}\right) + R \ln\left(\frac{T_2 P_1}{T_1 P_2}\right) \\ &= c_v \ln\left(\frac{T_2}{T_1}\right) + R \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right) \\ &= c_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right) \end{aligned}$$

7-75 For an ideal gas, $dh = c_p\, dT$ and $v = RT/P$. From the second Tds relation,

$$ds = \frac{dh}{T} - \frac{v dP}{T} = \frac{c_p\, dP}{T} - \frac{RT}{P}\frac{dP}{T} = c_p \frac{dT}{T} - R\frac{dP}{P}$$

Integrating,

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$

Since c_p is assumed to be constant.

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7-76 Setting $\Delta s = 0$ gives

$$c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) = 0 \longrightarrow \ln\left(\frac{T_2}{T_1}\right) = \frac{R}{c_p}\ln\left(\frac{P_2}{P_1}\right) \longrightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{R/C_p}$$

but

$$\frac{R}{c_p} = \frac{c_p - c_v}{c_p} = 1 - \frac{1}{k} = \frac{k-1}{k} \qquad \text{since} \quad k = c_p / c_v. \text{ Thus,} \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

7-77 The entropy changes of helium and nitrogen is to be compared for the same initial and final states.

Assumptions Helium and nitrogen are ideal gases with constant specific heats.

Properties The properties of helium are $c_p = 5.1926$ kJ/kg·K, $R = 2.0769$ kJ/kg·K (Table A-2a). The specific heat of nitrogen at the average temperature of (427+27)/2=227°C=500 K is $c_p = 1.056$ kJ/kg·K (Table A-2b). The gas constant of nitrogen is $R = 0.2968$ kJ/kg·K (Table A-2a).

Analysis From the entropy change relation of an ideal gas,

$$\begin{aligned}\Delta s_{\text{He}} &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (5.1926\,\text{kJ/kg}\cdot\text{K})\ln\frac{(27+273)\text{K}}{(427+273)\text{K}} - (2.0769\,\text{kJ/kg}\cdot\text{K})\ln\frac{200\,\text{kPa}}{2000\,\text{kPa}} \\ &= \mathbf{0.3826\ kJ/kg\cdot K}\end{aligned}$$

$$\begin{aligned}\Delta s_{\text{N2}} &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.056\,\text{kJ/kg}\cdot\text{K})\ln\frac{(27+273)\text{K}}{(427+273)\text{K}} - (0.2968\,\text{kJ/kg}\cdot\text{K})\ln\frac{200\,\text{kPa}}{2000\,\text{kPa}} \\ &= -\mathbf{0.2113\ kJ/kg\cdot K}\end{aligned}$$

Hence, helium undergoes the largest change in entropy.

7-78 The entropy difference between the two states of air is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The specific heat of air at the average temperature of (500+50)/2=275°C = 548 K ≅ 550 K is $c_p = 1.040$ kJ/kg·K (Table A-2b). The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-2a).

Analysis From the entropy change relation of an ideal gas,

$$\begin{aligned}\Delta s_{\text{air}} &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.040\,\text{kJ/kg}\cdot\text{K})\ln\frac{(50+273)\text{K}}{(500+273)\text{K}} - (0.287\,\text{kJ/kg}\cdot\text{K})\ln\frac{100\,\text{kPa}}{2000\,\text{kPa}} \\ &= -\mathbf{0.0478\ kJ/kg\cdot K}\end{aligned}$$

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7-79E The entropy difference between the two states of air is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The specific heat of air at the average temperature of (70+250)/2=160°F is c_p = 0.241 Btu/lbm·R (Table A-2Eb). The gas constant of air is R = 0.06855 Btu/lbm·R (Table A-2Ea).

Analysis From the entropy change relation of an ideal gas,

$$\begin{aligned}\Delta s_{\text{air}} &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\\ &= (0.241\,\text{Btu/lbm}\cdot\text{R})\ln\frac{(250+460)\text{R}}{(70+460)\text{R}} - (0.06855\,\text{Btu/lbm}\cdot\text{R})\ln\frac{40\,\text{psia}}{15\,\text{psia}}\\ &= \mathbf{0.00323\,Btu/lbm\cdot R}\end{aligned}$$

7-80 The final temperature of nitrogen when it is expanded isentropically is to be determined.

Assumptions Nitrogen is an ideal gas with constant specific heats.

Properties The specific heat ratio of nitrogen at an anticipated average temperature of 450 K is k = 1.391 (Table A-2b).

Analysis From the isentropic relation of an ideal gas under constant specific heat assumption,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300+273\,\text{K})\left(\frac{100\,\text{kPa}}{900\,\text{kPa}}\right)^{0.391/1.391} = \mathbf{309\,K}$$

Discussion The average air temperature is (573+309)/2=441 K, which is sufficiently close to the assumed average temperature of 450 K.

7-81E The final temperature of air when it is compressed isentropically is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air at an anticipated average temperature of 400°F is k = 1.389 (Table A-2Eb).

Analysis From the isentropic relation of an ideal gas under constant specific heat assumption,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (70+460\,\text{R})\left(\frac{200\,\text{psia}}{15\,\text{psia}}\right)^{0.389/1.389} = \mathbf{1095\,R}$$

Discussion The average air temperature is (530+1095)/2=813 K = 353°F, which is sufficiently close to the assumed average temperature of 400°F.

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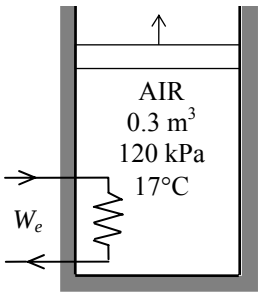
7-82 An insulated cylinder initially contains air at a specified state. A resistance heater inside the cylinder is turned on, and air is heated for 15 min at constant pressure. The entropy change of air during this process is to be determined for the cases of constant and variable specific heats.

Assumptions At specified conditions, air can be treated as an ideal gas.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

Analysis The mass of the air and the electrical work done during this process are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(120\ \text{kPa})(0.3\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\ \text{K})} = 0.4325\ \text{kg}$$
$$W_{e,in} = \dot{W}_{e,in}\Delta t = (0.2\ \text{kJ/s})(15\times 60\ \text{s}) = 180\ \text{kJ}$$



The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} - W_{b,out} = \Delta U \longrightarrow W_{e,in} = m(h_2 - h_1) \cong c_p(T_2 - T_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.

(*a*) Using a constant c_p value at the anticipated average temperature of 450 K, the final temperature becomes

$$\text{Thus,}\quad T_2 = T_1 + \frac{W_{e,in}}{mc_p} = 290\ \text{K} + \frac{180\ \text{kJ}}{(0.4325\ \text{kg})(1.02\ \text{kJ/kg}\cdot\text{K})} = 698\ \text{K}$$

Then the entropy change becomes

$$\Delta S_{sys} = m(s_2 - s_1) = m\left(c_{p,avg}\ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}^{\nearrow 0}\right) = mc_{p,avg}\ln\frac{T_2}{T_1}$$
$$= (0.4325\ \text{kg})(1.020\ \text{kJ/kg}\cdot\text{K})\ln\left(\frac{698\ \text{K}}{290\ \text{K}}\right) = \mathbf{0.387\ kJ/K}$$

(*b*) Assuming variable specific heats,

$$W_{e,in} = m(h_2 - h_1) \longrightarrow h_2 = h_1 + \frac{W_{e,in}}{m} = 290.16\ \text{kJ/kg} + \frac{180\ \text{kJ}}{0.4325\ \text{kg}} = 706.34\ \text{kJ/kg}$$

From the air table (Table A-17, we read $s_2^\circ = 2.5628$ kJ/kg·K corresponding to this h_2 value. Then,

$$\Delta S_{sys} = m\left(s_2^\circ - s_1^\circ + R\ln\frac{P_2}{P_1}^{\nearrow 0}\right) = m(s_2^\circ - s_1^\circ) = (0.4325\ \text{kg})(2.5628 - 1.66802)\text{kJ/kg}\cdot\text{K} = \mathbf{0.387\ kJ/K}$$

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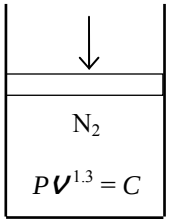
7-83 A cylinder contains N_2 gas at a specified pressure and temperature. The gas is compressed polytropically until the volume is reduced by half. The entropy change of nitrogen during this process is to be determined.

Assumptions **1** At specified conditions, N_2 can be treated as an ideal gas. **2** Nitrogen has constant specific heats at room temperature.

Properties The gas constant of nitrogen is $R = 0.2968$ kJ/kg.K (Table A-1). The constant volume specific heat of nitrogen at room temperature is $c_v = 0.743$ kJ/kg.K (Table A-2).

Analysis From the polytropic relation,

$$\frac{T_2}{T_1}=\left(\frac{v_1}{v_2}\right)^{n-1} \longrightarrow T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = (310\ \text{K})(2)^{1.3-1} = 381.7\ \text{K}$$



Then the entropy change of nitrogen becomes

$$\Delta S_{N_2} = m\left(c_{v,\text{avg}}\ln\frac{T_2}{T_1} + R\ln\frac{v_2}{v_1}\right)$$

$$= (0.75\ \text{kg})\left((0.743\ \text{kJ/kg}\cdot\text{K})\ln\frac{381.7\ \text{K}}{310\ \text{K}} + (0.2968\ \text{kJ/kg}\cdot\text{K})\ln(0.5)\right) = \mathbf{-0.0384\ kJ/K}$$

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7-84 Problem 7-83 is reconsidered. The effect of varying the polytropic exponent from 1 to 1.4 on the entropy change of the nitrogen is to be investigated, and the processes are to be shown on a common *P-v* diagram.

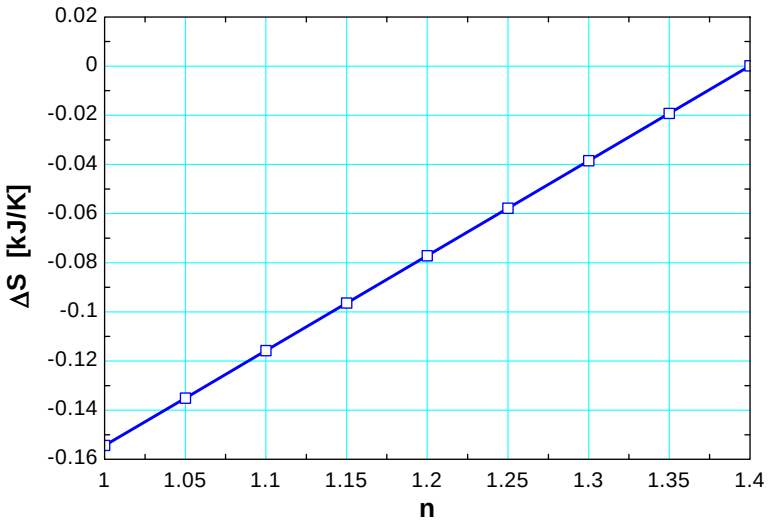
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

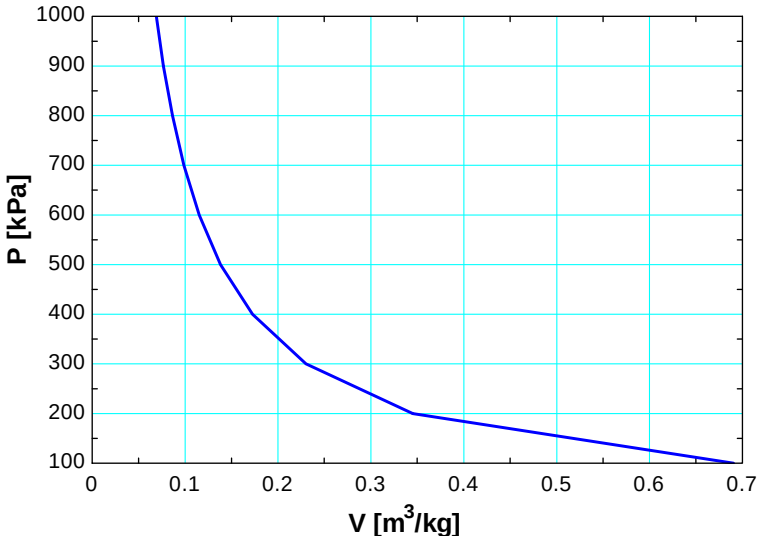
```
"Given"
m=0.75 [kg]
P1=140 [kPa]
T1=(37+273) [K]
n=1.3
RatioV=0.5 "RatioV=V2/V1"

"Properties"
c_v=0.743 [kJ/kg-K]
R=0.297 [kJ/kg-K]

"Analysis"
T2=T1*(1/RatioV)^(n-1) "from polytropic relation"
DELTAS=m*(c_v*ln(T2/T1)+R*ln(RatioV))
P1*V1=m*R*T1
```

n	ΔS [kJ/kg]
1	-0.1544
1.05	-0.1351
1.1	-0.1158
1.15	-0.09646
1.2	-0.07715
1.25	-0.05783
1.3	-0.03852
1.35	-0.01921
1.4	0.000104



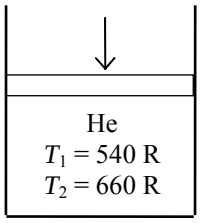


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7-85E A fixed mass of helium undergoes a process from one specified state to another specified state. The entropy change of helium is to be determined for the cases of reversible and irreversible processes.

Assumptions **1** At specified conditions, helium can be treated as an ideal gas. **2** Helium has constant specific heats at room temperature.

Properties The gas constant of helium is $R = 0.4961$ Btu/lbm.R (Table A-1E). The constant volume specific heat of helium is $c_v = 0.753$ Btu/lbm.R (Table A-2E).



Analysis From the ideal-gas entropy change relation,

$$\begin{aligned}\Delta S_{\text{He}} &= m\left(c_{v,\text{ave}}\ln\frac{T_2}{T_1}+R\ln\frac{v_2}{v_1}\right)\\ &= (15\text{ lbm})\left((0.753\text{ Btu/lbm}\cdot\text{R})\ln\frac{660\text{ R}}{540\text{ R}}+(0.4961\text{ Btu/lbm}\cdot\text{R})\ln\left(\frac{10\text{ ft}^3/\text{lbm}}{50\text{ ft}^3/\text{lbm}}\right)\right)\\ &= \mathbf{-9.71\text{ Btu/R}}\end{aligned}$$

The entropy change will be the same for both cases.

7-86 Air is expanded in a piston-cylinder device isothermally until a final pressure. The amount of heat transfer is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The specific heat of air at the given temperature of 127°C = 400 K is $c_p = 1.013$ kJ/kg·K (Table A-2b). The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-2a).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}}-E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$\begin{aligned}Q_{\text{in}}-W_{\text{out}} &= \Delta U = m(u_2-u_1 \quad (\text{since KE} = \text{PE} = 0)\\ Q_{\text{in}}-W_{\text{out}} &= \Delta U = m(u_2-u_1) = 0 \quad \text{since } T_1 = T_1\\ Q_{\text{in}} &= W_{\text{out}}\end{aligned}$$

The boundary work output during this isothermal process is

$$W_{\text{out}} = mRT\ln\frac{P_1}{P_2} = (1\text{ kg})(0.287\text{ kJ/kg}\cdot\text{K})(127+273\text{ K})\ln\frac{200\text{ kPa}}{100\text{ kPa}} = 79.6\text{ kJ}$$

Thus,

$$Q_{\text{in}} = W_{\text{out}} = \mathbf{79.6\text{ kJ}}$$

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7-87 Argon is expanded in an adiabatic turbine. The exit temperature and the maximum work output is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is adiabatic, and thus there is no heat transfer. **3** Argon is an ideal gas with constant specific heats.

Properties The properties of argon are $c_p = 0.2081$ kJ/kg·K and $k = 1.667$ (Table A-2*b*).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

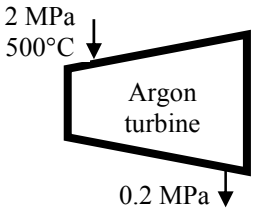
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

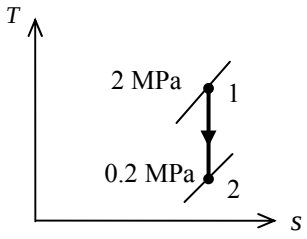
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{m}h_2 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$

$$w_{\text{out}} = h_1 - h_2$$





For the minimum work input to the compressor, the process must be reversible as well as adiabatic (i.e., isentropic). This being the case, the exit temperature will be

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (500 + 273\,\text{K})\left(\frac{200\,\text{kPa}}{2000\,\text{kPa}}\right)^{0.667/1.667} = \mathbf{308\,K}$$

Substituting into the energy balance equation gives

$$w_{\text{out}} = h_1 - h_2 = c_p(T_1 - T_2) = (0.2081\,\text{kJ/kg}\cdot\text{K})(773 - 308)\text{K} = \mathbf{96.8\,kJ/kg}$$

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7-88E Air is compressed in an isentropic compressor. The outlet temperature and the work input are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is adiabatic, and thus there is no heat transfer. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at an anticipated average temperature of 400°F are c_p = 0.245 Btu/lbm·R and k = 1.389 (Table A-2E*b*).

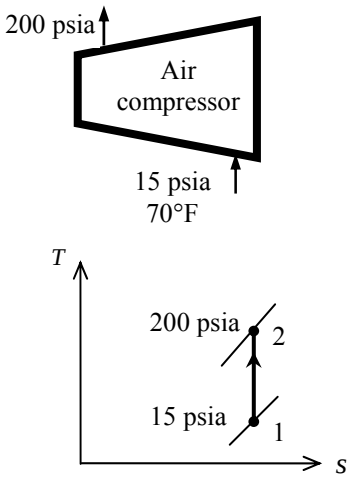
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$



The process is reversible as well as adiabatic (i.e., isentropic). This being the case, the exit temperature will be

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (70 + 460\ \text{R})\left(\frac{200\ \text{psia}}{15\ \text{psia}}\right)^{0.389/1.389} = \mathbf{1095\ R}$$

Substituting into the energy balance equation gives

$$w_{\text{in}} = h_2 - h_1 = c_p(T_2 - T_1) = (0.245\ \text{Btu/lbm} \cdot \text{R})(1095 - 530)\text{R} = \mathbf{138\ Btu/lbm}$$

Discussion The average air temperature is (530+1095)/2=813 K = 353°F, which is sufficiently close to the assumed average temperature of 400°F.

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7-89 One side of a partitioned insulated rigid tank contains an ideal gas at a specified temperature and pressure while the other side is evacuated. The partition is removed, and the gas fills the entire tank. The total entropy change during this process is to be determined.

Assumptions The gas in the tank is given to be an ideal gas, and thus ideal gas relations apply.

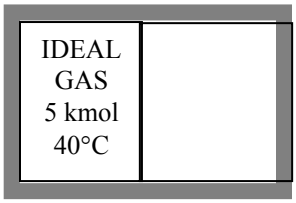
Analysis Taking the entire rigid tank as the system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1)$$

$$u_2 = u_1$$

$$T_2 = T_1$$



since $u = u(T)$ for an ideal gas. Then the entropy change of the gas becomes

$$\Delta S = N\left(\bar{c}_{v,\text{avg}} \ln \frac{T_2}{T_1}^{\nearrow 0} + R_u \ln \frac{v_2}{v_1}\right) = NR_u \ln \frac{v_2}{v_1}$$

$$= (5 \text{ kmol})(8.314 \text{ kJ/kmol} \cdot \text{K}) \ln(2)$$

$$= \mathbf{28.81 \text{ kJ/K}}$$

This also represents the **total entropy change** since the tank does not contain anything else, and there are no interactions with the surroundings.

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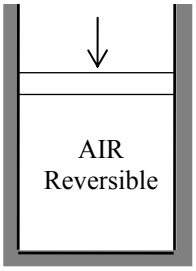
7-90 Air is compressed in a piston-cylinder device in a reversible and adiabatic manner. The final temperature and the work are to be determined for the cases of constant and variable specific heats.

Assumptions **1** At specified conditions, air can be treated as an ideal gas. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1). The specific heat ratio of air at the anticipated average temperature of 425 K is $k = 1.393$ and $R = 0.730$ kJ/kg.K (Table A-2a).

Analysis (*a*) Assuming constant specific heats, the ideal gas isentropic relations give

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (295\text{ K})\left(\frac{900\text{ kPa}}{90\text{ kPa}}\right)^{0.393/1.393} = \mathbf{564.9\ K}$$



Since $T_{avg} = (295 + 565)/2 = 430\text{ K}$

the assumed average temperature (425 K) is close enough to his value.

We take the air in the cylinder as the system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{in} = \Delta U = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$

Thus,

$$w_{in} = c_{v,avg}(T_2 - T_1) = (0.730\text{ kJ/kg}\cdot\text{K})(564.9 - 295)\text{K} = \mathbf{197.0\ kJ/kg}$$

(*b*) Assuming variable specific heats, the final temperature can be determined using the relative pressure data (Table A-17),

$$T_1 = 295\text{ K} \longrightarrow \begin{matrix} P_{r_1} = 1.3068 \\ u_1 = 210.49\text{ kJ/kg} \end{matrix}$$

and

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = \frac{900\text{ kPa}}{90\text{ kPa}}(1.3068) = 13.068 \longrightarrow \begin{matrix} T_2 = 564.9\ \mathbf{K} \\ u_2 = 408.09\text{ kJ/kg} \end{matrix}$$

Then the work input becomes

$$w_{in} = u_2 - u_1 = (408.09 - 210.49)\text{ kJ/kg} = \mathbf{197.6\ kJ/kg}$$

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7-91 Problem 7-90 is reconsidered. The work done and final temperature during the compression process are to be calculated and plotted as functions of the final pressure for the two cases as the final pressure varies from 100 kPa to 1200 kPa.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
Procedure ConstPropSol(P_1,T_1,P_2,Gas$:Work_in_ConstProp,T2_ConstProp)
C_P=SPECHEAT(Gas$,T=27)
MM=MOLARMASS(Gas$)
R_u=8.314 [kJ/kmol-K]
R=R_u/MM
C_V = C_P - R
k = C_P/C_V
T2= (T_1+273)*(P_2/P_1)^((k-1)/k)
T2_ConstProp=T2-273 "[C]"
DELTAu = C_v*(T2-(T_1+273))
Work_in_ConstProp = DELTAu
End

"Knowns:"
P_1 = 90 [kPa]
T_1 = 22 [C]
"P_2 = 900 [kPa]"

"Analysis: "
" Treat the piston-cylinder as a closed system, with no heat transfer in, neglect
changes in KE and PE of the air.  The process is reversible and adiabatic thus isentropic."
"The isentropic work is determined from:"
e_in - e_out = DELTAe_sys
e_out = 0 [kJ/kg]
e_in = Work_in
DELTAE_sys = (u_2 - u_1)

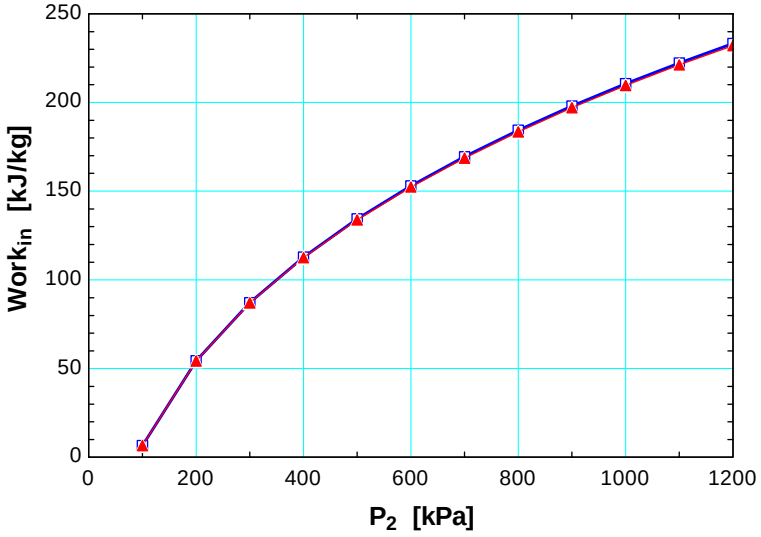
u_1 = INTENERGY(air,T=T_1)
v_1 = volume(air,P=P_1,T=T_1)
s_1 = entropy(air,P=P_1,T=T_1)

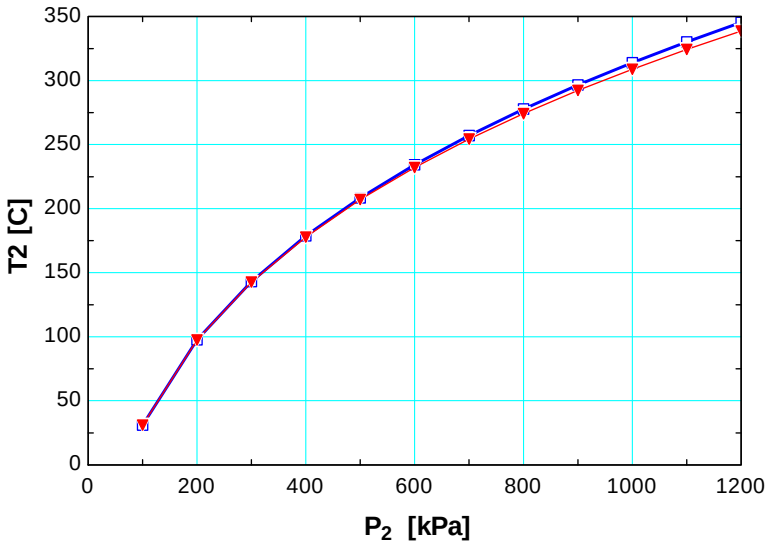
" The process is reversible and adiabatic or isentropic. "
s_2 = s_1
u_2 = INTENERGY(air,P=P_2,s=s_2)
T_2_isen = temperature(air,P=P_2,s=s_2)

Gas$ = 'air'
Call ConstPropSol(P_1,T_1,P_2,Gas$:Work_in_ConstProp,T2_ConstProp)
```

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P_2 [kPa]	$Work_{in}$ [kJ/kg]	$Work_{in,ConstProp}$ [kJ/kg]	$T2_{ConstProp}$ [C]	$T_{2,isen}$ [C]
100	6.467	6.469	31.01	31.01
200	54.29	54.25	97.59	97.42
300	87.09	86.91	143.1	142.6
400	112.8	112.5	178.7	177.8
500	134.3	133.8	208.5	207
600	152.9	152.3	234.2	232.1
700	169.4	168.7	257	254.3
800	184.3	183.5	277.7	274.2
900	198	197	296.5	292.2
1000	210.5	209.5	313.9	308.9
1100	222.3	221.1	330.1	324.2
1200	233.2	232	345.3	338.6



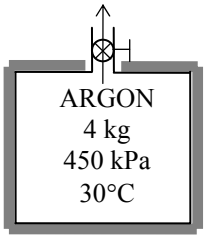


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7-92 An insulated rigid tank contains argon gas at a specified pressure and temperature. A valve is opened, and argon escapes until the pressure drops to a specified value. The final mass in the tank is to be determined.

Assumptions **1** At specified conditions, argon can be treated as an ideal gas. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply.

Properties The specific heat ratio of argon is $k = 1.667$ (Table A-2).



Analysis From the ideal gas isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (303\ \text{K})\left(\frac{200\ \text{kPa}}{450\ \text{kPa}}\right)^{0.667/1.667} = 219.0\ \text{K}$$

The final mass in the tank is determined from the ideal gas relation,

$$\frac{P_1 \cancel{V}}{P_2 \cancel{V}} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow m_2 = \frac{P_2 T_1}{P_1 T_2} m_1 = \frac{(200\ \text{kPa})(303\ \text{K})}{(450\ \text{kPa})(219\ \text{K})}(4\ \text{kg}) = \mathbf{2.46\ kg}$$

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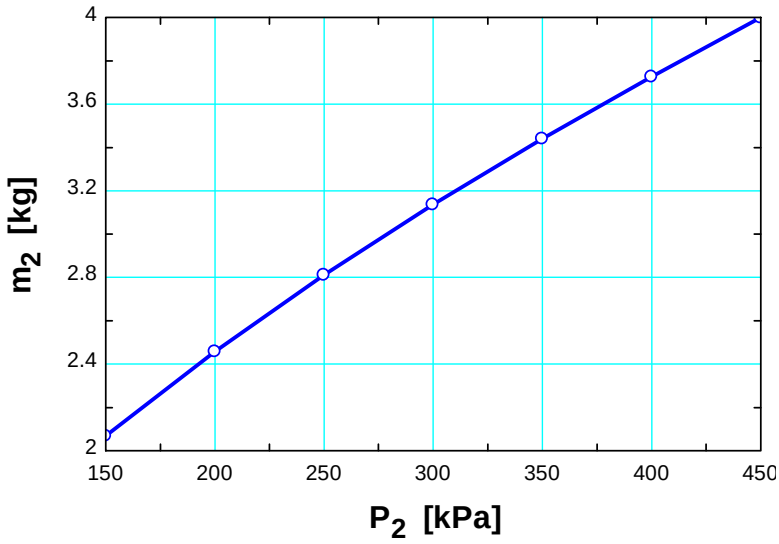
7-93 Problem 7-92 is reconsidered. The effect of the final pressure on the final mass in the tank is to be investigated as the pressure varies from 450 kPa to 150 kPa, and the results are to be plotted.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
c_p = 0.5203 [kJ/kg-K ]
c_v = 0.3122 [kJ/kg-K ]
R=0.2081 [kPa-m^3/kg-K]
P_1= 450 [kPa]
T_1 = 30 [C]
m_1 = 4 [kg]
P_2= 150 [kPa]

"Analysis:
We assume the mass that stays in the tank undergoes an isentropic expansion process. This allows us to determine the final temperature of that gas at the final pressure in the tank by using the isentropic relation:"
k = c_p/c_v
T_2 = ((T_1+273)*(P_2/P_1)^((k-1)/k)-273)
V_2 = V_1
P_1*V_1=m_1*R*(T_1+273)
P_2*V_2=m_2*R*(T_2+273)
```

P_2 [kPa]	m_2 [kg]
150	2.069
200	2.459
250	2.811
300	3.136
350	3.44
400	3.727
450	4



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7-94E Air is accelerated in an adiabatic nozzle. Disregarding irreversibilities, the exit velocity of air is to be determined.

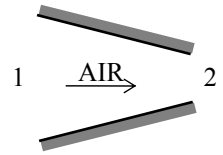
Assumptions **1** Air is an ideal gas with variable specific heats. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply. **2** The nozzle operates steadily.

Analysis Assuming variable specific heats, the inlet and exit properties are determined to be

$$T_1 = 1000 \text{ R} \longrightarrow \begin{matrix} P_{r_1} = 12.30 \\ h_1 = 240.98 \text{ Btu/lbm} \end{matrix}$$

and

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{12 \text{ psia}}{60 \text{ psia}}(12.30) = 2.46 \longrightarrow \begin{matrix} T_2 = 635.9 \text{ R} \\ h_2 = 152.11 \text{ Btu/lbm} \end{matrix}$$



We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = 0$$

Therefore,

$$V_2 = \sqrt{2(h_1 - h_2) + V_1^2} = \sqrt{2(240.98 - 152.11)\text{Btu/lbm}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right) + (200 \text{ ft/s})^2}$$

$$= \mathbf{2119 \text{ ft/s}}$$

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7-95 Air is expanded in an adiabatic nozzle by a polytropic process. The temperature and velocity at the exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).

Analysis For the polytropic process of an ideal gas, $Pv^n = \text{Constant}$, and the exit temperature is given by

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n-1)/n} = (373\text{ K})\left(\frac{200\text{ kPa}}{700\text{ kPa}}\right)^{0.3/1.3} = \mathbf{279\text{ K}}$$

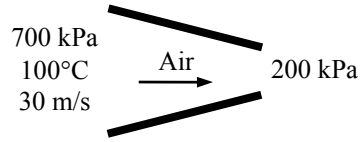
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$



Solving for the exit velocity,

$$\begin{aligned} V_2 &= \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} \\ &= \left[V_1^2 + 2c_p(T_1 - T_2)\right]^{0.5} \\ &= \left[(30\text{ m/s})^2 + 2(1.005\text{ kJ/kg}\cdot\text{K})(373-279)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)\right]^{0.5} \\ &= \mathbf{436\text{ m/s}} \end{aligned}$$

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7-96 Air is expanded in an adiabatic nozzle by a polytropic process. The temperature and velocity at the exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

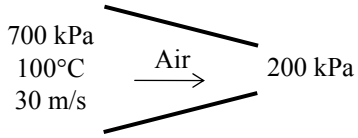
Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).

Analysis For the polytropic process of an ideal gas, $Pv^n = \text{Constant}$, and the exit temperature is given by

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n-1)/n} = (373\text{ K})\left(\frac{200\text{ kPa}}{700\text{ kPa}}\right)^{0.1/1.1} = \mathbf{333\ K}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

Solving for the exit velocity,

$$\begin{aligned} V_2 &= \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} \\ &= \left[V_1^2 + 2c_p(T_1 - T_2)\right]^{0.5} \\ &= \left[(30\text{ m/s})^2 + 2(1.005\text{ kJ/kg}\cdot\text{K})(373-333)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)\right]^{0.5} \\ &= \mathbf{286\ m/s} \end{aligned}$$

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7-97 Air is expanded in a piston-cylinder device until a final pressure. The maximum work input is given. The mass of air in the device is to be determined.

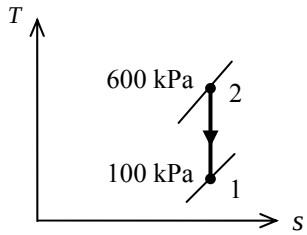
Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at 300 K is $c_v = 0.718$ kJ/kg·K and $k = 1.4$ (Table A-2a).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$



For the minimum work input to the compressor, the process must be reversible as well as adiabatic (i.e., isentropic). This being the case, the exit temperature will be

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (427 + 273\,\text{K})\left(\frac{100\,\text{kPa}}{600\,\text{kPa}}\right)^{0.4/1.4} = 419.5\,\text{K}$$

Substituting into the energy balance equation gives

$$W_{\text{out}} = m(u_1 - u_2) = mc_v(T_1 - T_2) \longrightarrow m = \frac{W_{\text{out}}}{c_v(T_1 - T_2)} = \frac{1000\,\text{kJ}}{(0.718\,\text{kJ/kg}\cdot\text{K})(700 - 419.5)\text{K}} = \mathbf{4.97\ kg}$$

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7-98 Helium is compressed in a steady-flow, isentropic compressor. The exit temperature and velocity are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is adiabatic, and thus there is no heat transfer. **3** Helium is an ideal gas with constant specific heats.

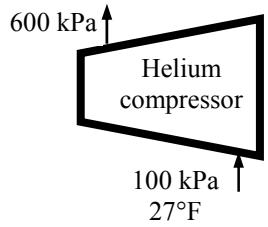
Properties The properties of helium are $c_p = 5.1926$ kJ/kg·K and $k = 1.667$ (Table A-2*b*).

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

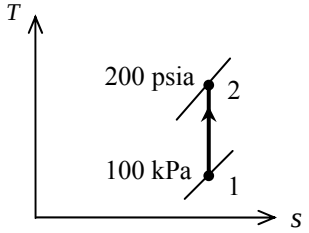
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) + \dot{W}_{\text{in}} = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) \qquad (\text{since } \dot{Q} \cong \Delta \text{pe} \cong 0)$$



The process should be isentropic for minimum work input. This being the case, the exit temperature will be

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (27 + 273\text{ K})\left(\frac{600\text{ kPa}}{100\text{ kPa}}\right)^{0.667/1.667} = \mathbf{614.4\ K}$$



(b) Solving the energy balance for the exit velocity and substituting,

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) + \dot{W}_{\text{in}} = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$(h_1 - h_2) + \frac{\dot{W}_{\text{in}}}{\dot{m}} = \left(\frac{V_2^2 - V_1^2}{2}\right)$$

$$V_2 = \left[V_1^2 + 2\left[(h_1 - h_2) + \frac{\dot{W}_{\text{in}}}{\dot{m}}\right]\right]^{0.5}$$

$$= \left[V_1^2 + 2\left[c_p(T_1 - T_2) + \frac{\dot{W}_{\text{in}}}{\dot{m}}\right]\right]^{0.5}$$

$$= \left[0 + 2\left[(5.1926\text{ kJ/kg}\cdot\text{K})(300 - 614.4)\text{K} + \frac{1000\text{ kW}}{0.6\text{ kg/s}}\right]\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)\right]^{0.5}$$

$$= \mathbf{261.2\ m/s}$$

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7-99 Air is expanded adiabatically in a piston-cylinder device. The entropy change is to be determined and it is to be discussed if this process is realistic.

Assumptions **1** Air is an ideal gas with constant specific heats.

Properties The properties of air at 300 K are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K and k = 1.4. Also, R = 0.287 kJ/kg·K (Table A-2*a*).

Analysis (a) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$-W_{\text{out}} = mc_v(T_2 - T_1)$$

Solving for the final temperature,

$$W_{\text{out}} = mc_v(T_1 - T_2) \longrightarrow T_2 = T_1 - \frac{W_{\text{out}}}{mc_v} = (427 + 273\,\text{K}) - \frac{600\,\text{kJ}}{(5\,\text{kg})(0.718\,\text{kJ/kg}\cdot\text{K})} = 532.9\,\text{K}$$

From the entropy change relation of an ideal gas,

$$\Delta s_{\text{air}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (1.005\,\text{kJ/kg}\cdot\text{K})\ln\frac{532.9\,\text{K}}{700\,\text{K}} - (0.287\,\text{kJ/kg}\cdot\text{K})\ln\frac{100\,\text{kPa}}{600\,\text{kPa}}$$

$$= \mathbf{0.240\,kJ/kg\cdot K}$$

(b) Since the entropy change is positive for this adiabatic process, the process is irreversible and realistic.

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7-100 Air contained in a constant-volume tank s cooled to ambient temperature. The entropy changes of the air and the universe due to this process are to be determined and the process is to be sketched on a T-s diagram.

***Assumptions* 1** Air is an ideal gas with constant specific heats.

Properties The specific heat of air at room temperature is $c_v = 0.718$ kJ/kg.K (Table A-2a).



Analysis (a) The entropy change of air is determined from

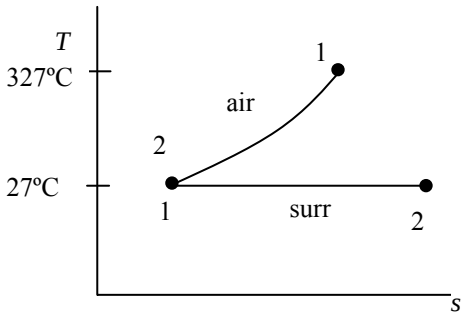
$$\Delta S_{\text{air}} = mc_v \ln\frac{T_2}{T_1}$$
$$= (5\,\text{kg})(0.718\,\text{kJ/kg.K})\ln\frac{(27+273)\,\text{K}}{(327+273)\,\text{K}}$$
$$= \mathbf{-2.488\,kJ/K}$$

(b) An energy balance on the system gives

$$Q_{\text{out}} = mc_v(T_1 - T_2)$$
$$= (5\,\text{kg})(0.718\,\text{kJ/kg.K})(327-27)$$
$$= 1077\,\text{kJ}$$

The entropy change of the surroundings is

$$\Delta s_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{1077\,\text{kJ}}{300\,\text{K}} = 3.59\,\text{kJ/K}$$



The entropy change of universe due to this process is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{air}} + \Delta S_{\text{surr}} = -2.488 + 3.59 = \mathbf{1.10\,kJ/K}$$

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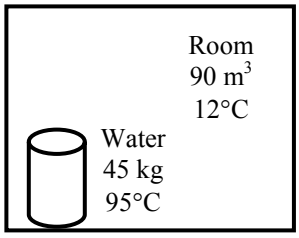
7-101 A container filled with liquid water is placed in a room and heat transfer takes place between the container and the air in the room until the thermal equilibrium is established. The final temperature, the amount of heat transfer between the water and the air, and the entropy generation are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats. **3** The room is well-sealed and there is no heat transfer from the room to the surroundings. **4** Sea level atmospheric pressure is assumed. P = 101.3 kPa.

Properties The properties of air at room temperature are R = 0.287 kPa.m^3/kg.K, c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg.K. The specific heat of water at room temperature is c_w = 4.18 kJ/kg.K (Tables A-2, A-3).

Analysis (*a*) The mass of the air in the room is

$$m_a = \frac{P\boldsymbol{V}}{RT_{a1}} = \frac{(101.3\,\text{kPa})(90\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(12+273\,\text{K})} = 111.5\,\text{kg}$$



An energy balance on the system that consists of the water in the container and the air in the room gives the final equilibrium temperature

$$0 = m_w c_w (T_2 - T_{w1}) + m_a c_v (T_2 - T_{a1})$$

$$0 = (45\,\text{kg})(4.18\,\text{kJ/kg.K})(T_2 - 95) + (111.5\,\text{kg})(0.718\,\text{kJ/kg.K})(T_2 - 12) \longrightarrow T_2 = \mathbf{70.2°C}$$

(*b*) The heat transfer to the air is

$$Q = m_a c_v (T_2 - T_{a1}) = (111.5\,\text{kg})(0.718\,\text{kJ/kg.K})(70.2 - 12) = \mathbf{4660\,kJ}$$

(*c*) The entropy generation associated with this heat transfer process may be obtained by calculating total entropy change, which is the sum of the entropy changes of water and the air.

$$\Delta S_w = m_w c_w \ln\frac{T_2}{T_{w1}} = (45\,\text{kg})(4.18\,\text{kJ/kg.K})\ln\frac{(70.2+273)\,\text{K}}{(95+273)\,\text{K}} = -13.11\,\text{kJ/K}$$

$$P_2 = \frac{m_a R T_2}{\boldsymbol{V}} = \frac{(111.5\,\text{kg})(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(70.2+273\,\text{K})}{(90\,\text{m}^3)} = 122\,\text{kPa}$$

$$\Delta S_a = m_a\left(c_p \ln\frac{T_2}{T_{a1}} - R\ln\frac{P_2}{P_1}\right)$$

$$= (111.5\,\text{kg})\left[(1.005\,\text{kJ/kg.K})\ln\frac{(70.2+273)\,\text{K}}{(12+273)\,\text{K}} - (0.287\,\text{kJ/kg.K})\ln\frac{122\,\text{kPa}}{101.3\,\text{kPa}}\right] = 14.88\,\text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_w + \Delta S_a = -13.11 + 14.88 = \mathbf{1.77\,kJ/K}$$

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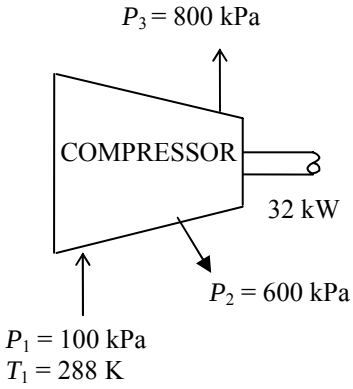
7-102 An ideal gas is compressed in an isentropic compressor. 10% of gas is compressed to 600 kPa and 90% is compressed to 800 kPa. The compression process is to be sketched, and the exit temperatures at the two exits, and the mass flow rate into the compressor are to be determined.

Assumptions **1** The compressor operates steadily. **2** The process is reversible-adiabatic (isentropic)

Properties The properties of ideal gas are given to be c_p = 1.1 kJ/kg.K and c_v = 0.8 kJ/kg.K.

Analysis (b) The specific heat ratio of the gas is

$$k = \frac{c_p}{c_v} = \frac{1.1}{0.8} = 1.375$$



The exit temperatures are determined from ideal gas isentropic relations to be

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (15+273\text{ K})\left(\frac{600\text{ kPa}}{100\text{ kPa}}\right)^{0.375/1.375} = \mathbf{469.5\ K}$$

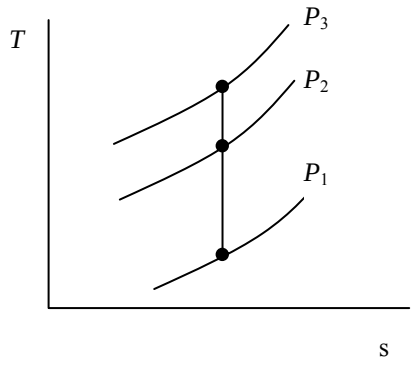
$$T_3 = T_1\left(\frac{P_3}{P_1}\right)^{(k-1)/k} = (15+273\text{ K})\left(\frac{800\text{ kPa}}{100\text{ kPa}}\right)^{0.375/1.375} = \mathbf{507.8\ K}$$

(c) A mass balance on the control volume gives

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3$$

where

$$\dot{m}_2 = 0.1\dot{m}_1$$
$$\dot{m}_3 = 0.9\dot{m}_1$$



We take the compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_1 h_1 + \dot{W}_{\text{in}} = \dot{m}_2 h_2 + \dot{m}_3 h_3$$
$$\dot{m}_1 c_p T_1 + \dot{W}_{\text{in}} = 0.1\dot{m}_1 c_p T_2 + 0.9\dot{m}_1 c_p T_3$$

Solving for the inlet mass flow rate, we obtain

$$\dot{m}_1 = \frac{\dot{W}_{in}}{c_p\left[0.1(T_2 - T_1) + 0.9(T_3 - T_1)\right]}$$
$$= \frac{32\text{ kW}}{(1.1\text{ kJ/kg}\cdot\text{K})\left[0.1(469.5-288) + 0.9(507.8-288)\right]}$$
$$= \mathbf{0.1347\ kg/s}$$

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7-103E Air is charged to an initially evacuated container from a supply line. The minimum temperature of the air in the container after it is filled is to be determined.

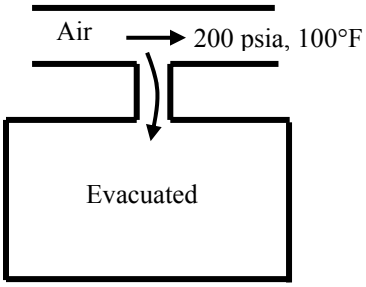
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The specific heat of air at room temperature is c_p = 0.240 Btu/lbm·R (Table A-2Ea).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and entropy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$m_i = m_2 - m_1$$
$$m_i = m_2$$



Entropy balance:

$$m_2 s_2 - m_1 s_1 + m_e s_e - m_i s_i \geq 0$$
$$m_2 s_2 - m_i s_i \geq 0$$

Combining the two balances,

$$m_2 s_2 - m_2 s_i \geq 0$$
$$s_2 - s_i \geq 0$$

The minimum temperature will result when the equal sign applies. Noting that $P_2 = P_i$, we have

$$s_2 - s_i = c_p \ln \frac{T_2}{T_i} - R \ln \frac{P_2}{P_i} = 0 \longrightarrow c_p \ln \frac{T_2}{T_i} = 0$$

Then,

$$T_2 = T_i = \mathbf{100°F}$$

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Reversible Steady-Flow Work

7-104C The work associated with steady-flow devices is proportional to the specific volume of the gas. Cooling a gas during compression will reduce its specific volume, and thus the power consumed by the compressor.

7-105C Cooling the steam as it expands in a turbine will reduce its specific volume, and thus the work output of the turbine. Therefore, this is not a good proposal.

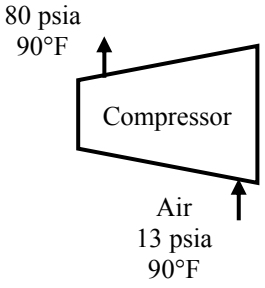
7-106C We would not support this proposal since the steady-flow work input to the pump is proportional to the specific volume of the liquid, and cooling will not affect the specific volume of a liquid significantly.

7-107E Air is compressed isothermally in a reversible steady-flow device. The work required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer associated with the process. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $R = 0.06855$ Btu/lbm·R (Table A-1E).

Analysis Substituting the ideal gas equation of state into the reversible steady-flow work expression gives



$$w_{\text{in}} = \int_1^2 v\,dP = RT\int_1^2 \frac{dP}{P} = RT\ln\frac{P_2}{P_1}$$

$$= (0.06855\ \text{Btu/lbm}\cdot\text{R})(90+460\ \text{K})\ln\left(\frac{80\ \text{psia}}{13\ \text{psia}}\right)$$

$$= \mathbf{68.5\ Btu/lbm}$$

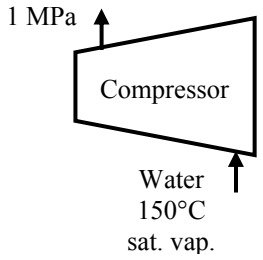
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7-108 Saturated water vapor is compressed in a reversible steady-flow device. The work required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer associated with the process. **3** Kinetic and potential energy changes are negligible.

Analysis The properties of water at the inlet state are

$$\left.\begin{array}{l} T_1 = 150°\text{C} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} P_1 = 476.16\ \text{kPa} \\ \nu_1 = 0.39248\ \text{m}^3/\text{kg} \end{array} \quad \text{(Table A - 4)}$$



Noting that the specific volume remains constant, the reversible steady-flow work expression gives

$$w_{\text{in}} = \int_1^2 \nu dP = \nu_1 (P_2 - P_1)$$

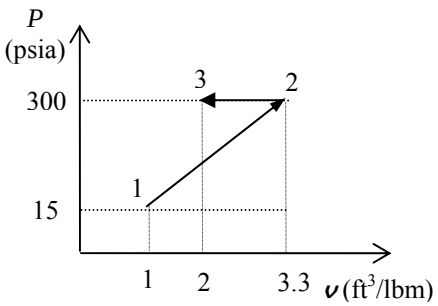
$$= (0.39248\ \text{m}^3/\text{kg})(1000 - 476.16)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa} \cdot \text{m}^3}\right)$$

$$= \mathbf{205.6\ kJ/kg}$$

7-109E The reversible work produced during the process shown in the figure is to be determined.

Assumptions The process is reversible.

Analysis The work produced is equal to the areas to the left of the reversible process line on the *P*-ν diagram. The work done during the process 2-3 is zero. Then,



$$w_{13} = w_{12} + 0 = \int_1^2 \nu dP = \frac{\nu_1 + \nu_2}{2}(P_2 - P_1)$$

$$= \frac{(1 + 3.3)\text{ft}^3/\text{lbm}}{2}(300 - 15)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia} \cdot \text{ft}^3}\right)$$

$$= \mathbf{113.4\ Btu/lbm}$$

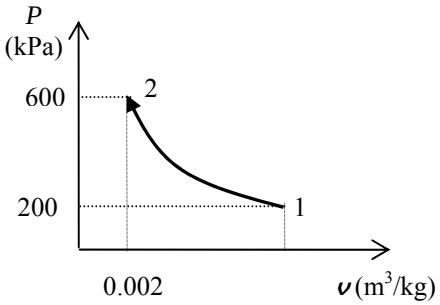
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7-110 The reversible work produced during the process shown in the figure is to be determined.

Assumptions The process is reversible.

Analysis The reversible work relation is

$$w_{12} = \int_1^2 \nu dP$$



When combined with the ideal gas equation of state

$$\nu = \frac{RT}{P}$$

The work expression reduces to

$$w_{12} = \int_1^2 \nu dP = -RT\int_1^2 \frac{dP}{P} = -RT\ln\frac{P_2}{P_1} = -P_2\nu_2\ln\frac{P_2}{P_1}$$

$$= -(600\text{ kPa})(0.002\text{ m}^3/\text{kg})\ln\frac{600\text{ kPa}}{200\text{ kPa}}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{-1.32\text{ kJ/kg}}$$

The negative sign indicates that work is done on the system in the amount of 1.32 kJ/kg.

7-111 Liquid water is to be pumped by a 25-kW pump at a specified rate. The highest pressure the water can be pumped to is to be determined.

Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is assumed to be reversible since we will determine the limiting case.

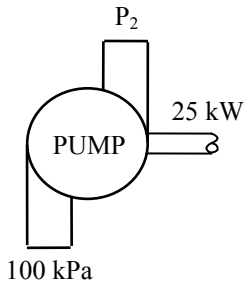
Properties The specific volume of liquid water is given to be $\nu_1 = 0.001\text{ m}^3/\text{kg}$.

Analysis The highest pressure the liquid can have at the pump exit can be determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 \nu dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}\right) = \dot{m}\nu_1(P_2 - P_1)$$

Thus,

$$25\text{ kJ/s} = (5\text{ kg/s})(0.001\text{ m}^3/\text{kg})(P_2 - 100)\text{k Pa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$



It yields

$$P_2 = \mathbf{5100\text{ kPa}}$$

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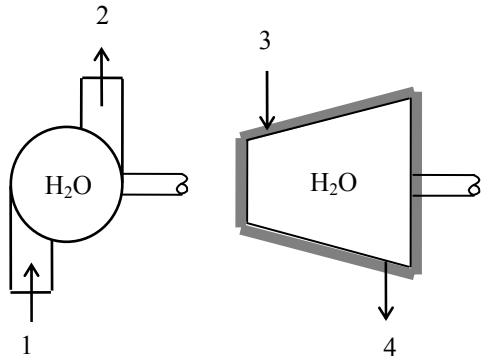
7-112 A steam power plant operates between the pressure limits of 5 MPa and 10 kPa. The ratio of the turbine work to the pump work is to be determined.

Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible. **4** The pump and the turbine are adiabatic.

Analysis Both the compression and expansion processes are reversible and adiabatic, and thus isentropic, $s_1 = s_2$ and $s_3 = s_4$. Then the properties of the steam are

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ \textit{sat.vapor} \end{array}\right\} \begin{array}{l} h_4 = h_{g@10\text{ kPa}} = 2583.9 \text{ kJ/kg} \\ s_4 = s_{g@10\text{ kPa}} = 8.1488 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 5 \text{ MPa} \\ s_3 = s_4 \end{array}\right\} h_3 = 4608.1 \text{ kJ/kg}$$



Also, $\nu_1 = \nu_{f@\,10\text{ kPa}} = 0.00101 \text{ m}^3/\text{kg}$.

The work output to this isentropic turbine is determined from the steady-flow energy balance to be

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_3 = \dot{m}h_4 + \dot{W}_{\text{out}}$$
$$\dot{W}_{\text{out}} = \dot{m}(h_3 - h_4)$$

Substituting,

$$w_{\text{turb,out}} = h_3 - h_4 = 4608.1 - 2583.9 = 2024.2 \text{ kJ/kg}$$

The pump work input is determined from the steady-flow work relation to be

$$w_{\text{pump,in}} = \int_1^2 \nu dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} = \nu_1 (P_2 - P_1)$$
$$= (0.00101 \text{ m}^3/\text{kg})(5000 - 10)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$
$$= 5.041 \text{ kJ/kg}$$

Thus,

$$\frac{w_{\text{turb,out}}}{w_{\text{pump,in}}} = \frac{2024.2}{5.041} = \mathbf{402}$$

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7-113 Problem 7-112 is reconsidered. The effect of the quality of the steam at the turbine exit on the net work output is to be investigated as the quality is varied from 0.5 to 1.0, and the net work output us to be plotted as a function of this quality.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
WorkFluid$ = 'Steam_IAPWS'
P[1] = 10  [kPa]
x[1] = 0
P[2] = 5000 [kPa]
x[4] = 1.0

"Pump Analysis:"
T[1]=temperature(WorkFluid$,P=P[1],x=0)
v[1]=volume(workFluid$,P=P[1],x=0)
h[1]=enthalpy(WorkFluid$,P=P[1],x=0)
s[1]=entropy(WorkFluid$,P=P[1],x=0)
s[2] = s[1]
h[2]=enthalpy(WorkFluid$,P=P[2],s=s[2])
T[2]=temperature(WorkFluid$,P=P[2],s=s[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

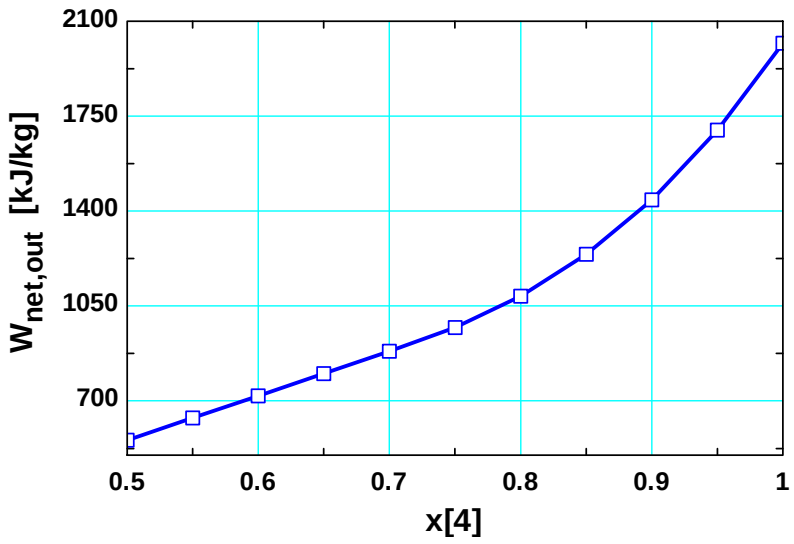
"Conservation of Energy - SSSF energy balance for pump"
" -- neglect the change in potential energy, no  heat transfer:"
h[1]+W_pump = h[2]

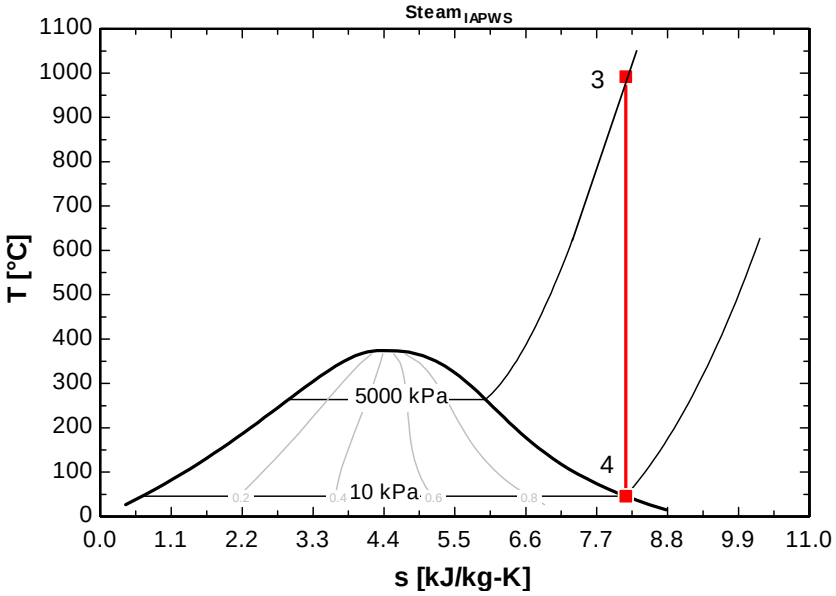
"Also the work of pump can be obtained from the incompressible fluid, steady-flow result:"
W_pump_incomp = v[1]*(P[2] - P[1])

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential energy, no
heat transfer:"
P[4] = P[1]
P[3] = P[2]
h[4]=enthalpy(WorkFluid$,P=P[4],x=x[4])
s[4]=entropy(WorkFluid$,P=P[4],x=x[4])
T[4]=temperature(WorkFluid$,P=P[4],x=x[4])
s[3] = s[4]
h[3]=enthalpy(WorkFluid$,P=P[3],s=s[3])
T[3]=temperature(WorkFluid$,P=P[3],s=s[3])
h[3] = h[4] + W_turb
W_net_out  = W_turb - W_pump
```

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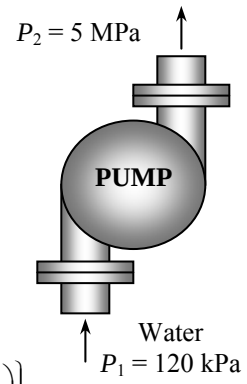
x_4	$W_{net,out}$ [kJ/kg]
0.5	555.6
0.55	637.4
0.6	719.2
0.65	801
0.7	882.8
0.75	971.2
0.8	1087
0.85	1240
0.9	1442
0.95	1699
1	2019





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7-114 Liquid water is pumped by a 70-kW pump to a specified pressure at a specified level. The highest possible mass flow rate of water is to be determined.



Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic energy changes are negligible, but potential energy changes may be significant. **3** The process is assumed to be reversible since we will determine the limiting case.

Properties The specific volume of liquid water is given to be $\nu_1 = 0.001\ \text{m}^3/\text{kg}$.

Analysis The highest mass flow rate will be realized when the entire process is reversible. Thus it is determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 \nu\, dP + \Delta ke^{\nearrow 0} + \Delta pe\right) = \dot{m}\{\nu(P_2 - P_1) + g(z_2 - z_1)\}$$

Thus,

$$7\ \text{kJ/s} = \dot{m}\left\{(0.001\ \text{m}^3/\text{kg})(5000-120)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) + (9.8\ \text{m/s}^2)(10\ \text{m})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right\}$$

It yields

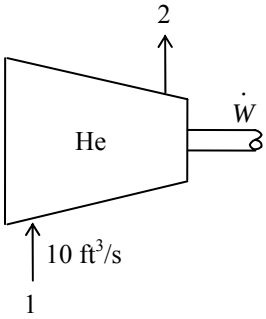
$$\dot{m} = \mathbf{1.41\ kg/s}$$

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7-115E Helium gas is compressed from a specified state to a specified pressure at a specified rate. The power input to the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible.

Properties The gas constant of helium is $R = 2.6805$ psia.ft^3/lbm.R = 0.4961 Btu/lbm.R. The specific heat ratio of helium is $k = 1.667$ (Table A-2E).



Analysis The mass flow rate of helium is

$$\dot{m} = \frac{P_1\dot{V}_1}{RT_1} = \frac{(16\ \text{psia})(10\ \text{ft}^3/\text{s})}{(2.6805\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(545\ \text{R})} = 0.1095\ \text{lbm/s}$$

(*a*) Isentropic compression with $k = 1.667$:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right\}$$

$$= (0.1095\ \text{lbm/s})\frac{(1.667)(0.4961\ \text{Btu/lbm}\cdot\text{R})(545\ \text{R})}{1.667-1}\left\{\left(\frac{120\ \text{psia}}{16\ \text{psia}}\right)^{0.667/1.667} - 1\right\}$$

$$= 91.74\ \text{Btu/s}$$

$$= \mathbf{129.8\ hp} \quad \text{since 1 hp = 0.7068 Btu/s}$$

(*b*) Polytropic compression with $n = 1.2$:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right\}$$

$$= (0.1095\ \text{lbm/s})\frac{(1.2)(0.4961\ \text{Btu/lbm}\cdot\text{R})(545\ \text{R})}{1.2-1}\left\{\left(\frac{120\ \text{psia}}{16\ \text{psia}}\right)^{0.2/1.2} - 1\right\}$$

$$= 70.89\ \text{Btu/s}$$

$$= \mathbf{100.3\ hp} \quad \text{since 1 hp = 0.7068 Btu/s}$$

(*c*) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT\ln\frac{P_2}{P_1} = (0.1095\ \text{lbm/s})(0.4961\ \text{Btu/lbm}\cdot\text{R})(545\ \text{R})\ln\frac{120\ \text{psia}}{16\ \text{psia}} = 59.67\ \text{Btu/s} = \mathbf{84.42\ hp}$$

(*d*) Ideal two-stage compression with intercooling ($n = 1.2$): In this case, the pressure ratio across each stage is the same, and its value is determined from

$$P_x = \sqrt{P_1P_2} = \sqrt{(16\ \text{psia})(120\ \text{psia})} = 43.82\ \text{psia}$$

The compressor work across each stage is also the same, thus total compressor work is twice the compression work for a single stage:

$$\dot{W}_{\text{comp,in}} = 2\dot{m}w_{\text{comp,I}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_x}{P_1}\right)^{(n-1)/n} - 1\right\}$$

$$= 2(0.1095\ \text{lbm/s})\frac{(1.2)(0.4961\ \text{Btu/lbm}\cdot\text{R})(545\ \text{R})}{1.2-1}\left\{\left(\frac{43\,82\ \text{psia}}{14\ \text{psia}}\right)^{0.2/1.2} - 1\right\}$$

$$= 64.97\ \text{Btu/s}$$

$$= \mathbf{91.92\ hp} \quad \text{since 1 hp = 0.7068 Btu/s}$$

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7-116E Problem 7-115E is reconsidered. The work of compression and entropy change of the helium is to be evaluated and plotted as functions of the polytropic exponent as it varies from 1 to 1.667.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Given"
P1=16 [psia]
T1=85+460
V1_dot=10 [ft^3/s]
P2=120 [psia]
n=1.2

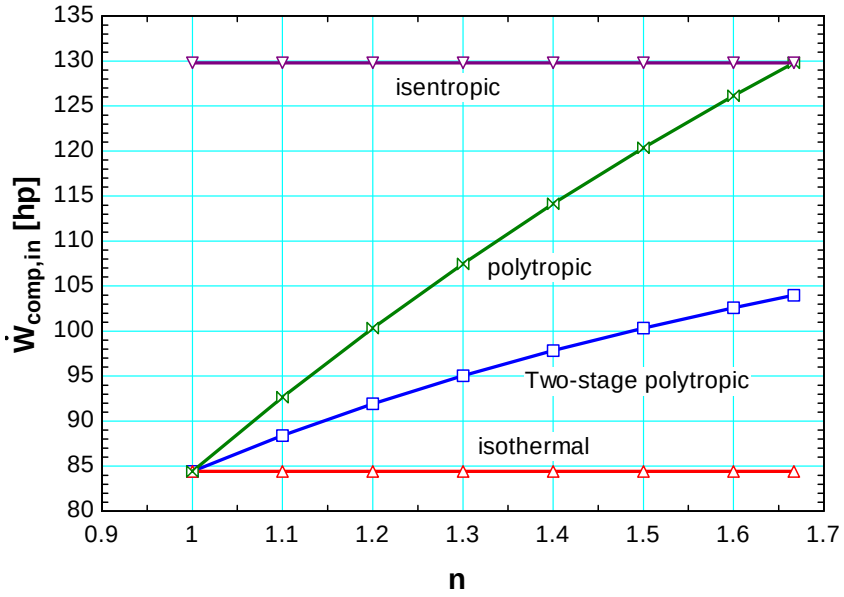
"Properties"
R=0.4961 [Btu/lbm-R]
R1=2.6805 [psia-ft^3/lbm-R]
k=1.667
c_p=1.25 [Btu/lbm-R]

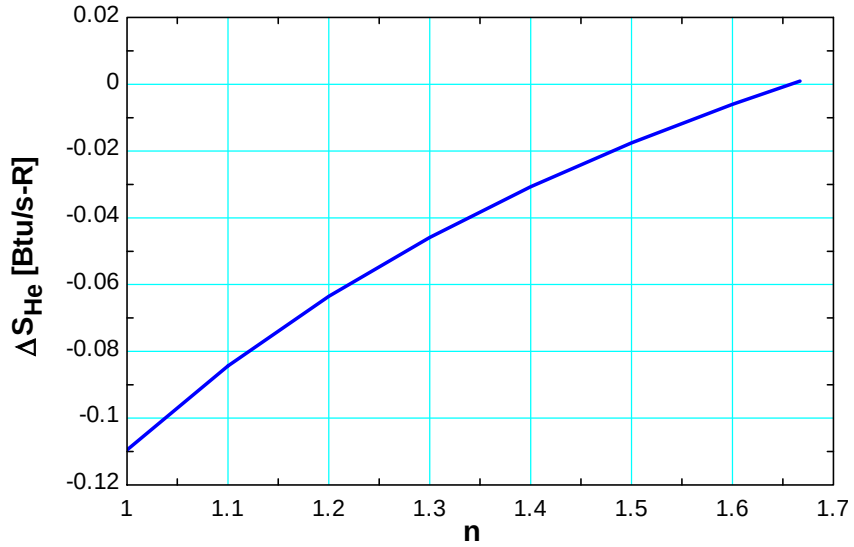
"Analysis"
m_dot=(P1*V1_dot)/(R1*T1)
W_dot_comp_in_a=m_dot*(k*R*T1)/(k-1)*((P2/P1)^((k-1)/k)-1)*Convert(Btu/s, hp)
W_dot_comp_in_b=m_dot*(n*R*T1)/(n-1)*((P2/P1)^((n-1)/n)-1)*Convert(Btu/s, hp)
W_dot_comp_in_c=m_dot*R*T1*ln(P2/P1)*Convert(Btu/s, hp)
P_x=sqrt(P1*P2)
W_dot_comp_in_d=2*m_dot*(n*R*T1)/(n-1)*((P_x/P1)^((n-1)/n)-1)*Convert(Btu/s, hp)

"Entropy change"
T2/T1=(P2/P1)^((n-1)/n)
DELTAS_He=m_dot*(c_p*ln(T2/T1)-R*ln(P2/P1))
```

n	$W_{comp,in,a}$ [hp]	$W_{comp,in,b}$ [hp]	$W_{comp,in,c}$ [hp]	$W_{comp,in,d}$ [hp]	ΔS_{He} [Btu/s-R]
1	129.8	84.42	84.42	84.42	-0.1095
1.1	129.8	92.64	84.42	88.41	-0.0844
1.2	129.8	100.3	84.42	91.92	-0.0635
1.3	129.8	107.5	84.42	95.04	-0.04582
1.4	129.8	114.1	84.42	97.82	-0.03066
1.5	129.8	120.3	84.42	100.3	-0.01753
1.6	129.8	126.1	84.42	102.6	-0.006036
1.667	129.8	129.8	84.42	104	0.0008937

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7-117 Water mist is to be sprayed into the air stream in the compressor to cool the air as the water evaporates and to reduce the compression power. The reduction in the exit temperature of the compressed air and the compressor power saved are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible. **3** Air is compressed isentropically. **4** Water vaporizes completely before leaving the compressor. **4** Air properties can be used for the air-vapor mixture.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1). The specific heat ratio of air is $k = 1.4$. The inlet enthalpies of water and air are (Tables A-4 and A-17)

$$h_{w1} = h_{f@20°C} = 83.29 \text{ kJ/kg}, \; h_{fg@20°C} = 2453.9 \text{ kJ/kg and } h_{a1} = h_{@300\text{ K}} = 300.19 \text{ kJ/kg}$$

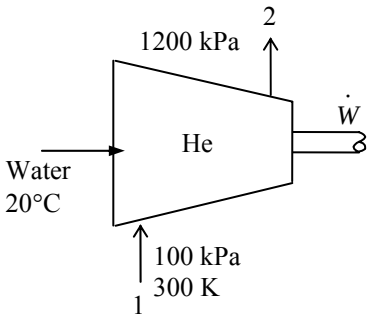
Analysis In the case of isentropic operation (thus no cooling or water spray), the exit temperature and the power input to the compressor are

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \rightarrow T_2 = (300\text{ K})\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(1.4-1)/1.4} = 610.2\text{ K}$$

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{\left(P_2/P_1\right)^{(k-1)/k} - 1\right\}$$

$$= (2.1\text{ kg/s})\frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left\{(1200\text{ kPa}/100\text{ kPa})^{0.4/1.4} - 1\right\} = 654.3\text{ kW}$$

When water is sprayed, we first need to check the accuracy of the assumption that the water vaporizes completely in the compressor. In the limiting case, the compression will be isothermal at the compressor inlet temperature, and the water will be a saturated vapor. To avoid the complexity of dealing with two fluid streams and a gas mixture, we disregard water in the air stream (other than the mass flow rate), and assume air is cooled by an amount equal to the enthalpy change of water.



The rate of heat absorption of water as it evaporates at the inlet temperature completely is

$$\dot{Q}_{\text{cooling,max}} = \dot{m}_w h_{fg\,@\,20°C} = (0.2\text{ kg/s})(2453.9\text{ kJ/kg}) = 490.8\text{ kW}$$

The minimum power input to the compressor is

$$\dot{W}_{\text{comp,in,min}} = \dot{m}RT\ln\frac{P_2}{P_1} = (2.1\text{ kg/s})(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\ln\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right) = 449.3\text{ kW}$$

This corresponds to maximum cooling from the air since, at constant temperature, $\Delta h = 0$ and thus $\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 449.3\text{ kW}$, which is close to 490.8 kW. Therefore, the assumption that all the water vaporizes is approximately valid. Then the reduction in required power input due to water spray becomes

$$\Delta\dot{W}_{\text{comp,in}} = \dot{W}_{\text{comp, isentropic}} - \dot{W}_{\text{comp, isothermal}} = 654.3 - 449.3 = \mathbf{205\ kW}$$

Discussion (can be ignored): At constant temperature, $\Delta h = 0$ and thus $\dot{Q}_{out} = \dot{W}_{in} = 449.3\text{ kW}$ corresponds to maximum cooling from the air, which is less than 490.8 kW. Therefore, the assumption that all the water vaporizes is only roughly valid. As an alternative, we can assume the compression process to be polytropic and the water to be a saturated vapor at the compressor exit temperature, and disregard the remaining liquid. But in this case there is not a unique solution, and we will have to select either the amount of water or the exit temperature or the polytropic exponent to obtain a solution. Of course we can also tabulate the results for different cases, and then make a selection.

Sample Analysis: We take the compressor exit temperature to be $T_2 = 200°C = 473$ K. Then,

$$h_{w2} = h_{g@200°C} = 2792.0 \text{ kJ/kg and } h_{a2} = h_{@473\text{ K}} = 475.3 \text{ kJ/kg}$$

Then,

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$$\frac{T_2}{T_1}=\left(\frac{P_2}{P_1}\right)^{(n-1)/n} \rightarrow \frac{473\,\text{K}}{300\,\text{K}}=\left(\frac{1200\,\text{kPa}}{100\,\text{kPa}}\right)^{(n-1)/n} \rightarrow n=1.224$$

$$\dot{W}_{comp,in}=\dot{m}\frac{nRT_1}{n-1}\left\{\left(P_2/P_1\right)^{(n-1)/n}-1\right\}=\dot{m}\frac{nR}{n-1}(T_2-T_1)$$
$$=(2.1\,\text{kg/s})\frac{(1.224)(0.287\,\text{kJ/kg}\cdot\text{K})}{1.224-1}(473-300)\text{K}=570\,\text{kW}$$

Energy balance:

$$\dot{W}_{\text{comp,in}}-\dot{Q}_{\text{out}}=\dot{m}(h_2-h_1) \rightarrow \dot{Q}_{\text{out}}=\dot{W}_{\text{comp,in}}-\dot{m}(h_2-h_1)$$
$$=569.7\,\text{kW}-(2.1\,\text{kg/s})(475.3-300.19)=202.0\,\text{kW}$$

Noting that this heat is absorbed by water, the rate at which water evaporates in the compressor becomes

$$\dot{Q}_{\text{out,air}}=\dot{Q}_{\text{in,water}}=\dot{m}_w(h_{w2}-h_{w1}) \longrightarrow \dot{m}_w=\frac{\dot{Q}_{\text{in,water}}}{h_{w2}-h_{w1}}=\frac{202.0\,\text{kJ/s}}{(2792.0-83.29)\,\text{kJ/kg}}=0.0746\,\text{kg/s}$$

Then the reductions in the exit temperature and compressor power input become

$$\Delta T_2=T_{2,\text{isentropic}}-T_{2,\text{water cooled}}=610.2-473=\mathbf{137.2^\circ C}$$
$$\Delta\dot{W}_{comp,in}=\dot{W}_{comp,\text{isentropic}}-\dot{W}_{comp,\text{water cooled}}=654.3-570=\mathbf{84.3\ kW}$$

Note that selecting a different compressor exit temperature T_2 will result in different values.

7-118 A water-injected compressor is used in a gas turbine power plant. It is claimed that the power output of a gas turbine will increase when water is injected into the compressor because of the increase in the mass flow rate of the gas (air + water vapor) through the turbine. This, however, is **not necessarily right** since the compressed air in this case enters the combustor at a low temperature, and thus it absorbs much more heat. In fact, the cooling effect will most likely dominate and cause the cyclic efficiency to drop.

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Isentropic Efficiencies of Steady-Flow Devices

7-119C The ideal process for all three devices is the reversible adiabatic (i.e., isentropic) process. The adiabatic efficiencies of these devices are defined as

$$\eta_T = \frac{\text{actual work output}}{\text{insentropic work output}},\ \eta_C = \frac{\text{insentropic work input}}{\text{actual work input}},\ \text{and}\ \eta_N = \frac{\text{actual exit kineticenergy}}{\text{insentropic exit kinetic energy}}$$

7-120C No, because the isentropic process is not the model or ideal process for compressors that are cooled intentionally.

7-121C Yes. Because the entropy of the fluid must increase during an actual adiabatic process as a result of irreversibilities. Therefore, the actual exit state has to be on the right-hand side of the isentropic exit state

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7-122E Steam is compressed in an adiabatic closed system with an isentropic efficiency of 80%. The work produced and the final temperature are to be determined.

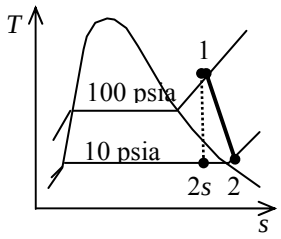
Assumptions **1** Kinetic and potential energy changes are negligible. **2** The device is adiabatic and thus heat transfer is negligible.

Analysis We take the steam as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$w_{\text{out}} = u_1 - u_2$$



From the steam tables (Tables A-5 and A-6),

$$\left.\begin{array}{l} P_1 = 100 \text{ psia} \\ T_1 = 650°\text{F} \end{array}\right\} \begin{array}{l} u_1 = 1233.7 \text{ Btu/lbm} \\ s_1 = 1.7816 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{1.7816 - 0.47427}{1.12888} = 0.9961 \\ u_{2s} = u_f + x_{2s} + u_{fg} = 298.19 + 0.9961 \times 807.29 = 1068.4 \text{ Btu/lbm} \end{array}$$

The work input during the isentropic process is

$$w_{s,\text{out}} = u_1 - u_{2s} = (1233.7 - 1068.4)\text{Btu/lbm} = 165.3 \text{ Btu/lbm}$$

The actual work input is then

$$w_{a,\text{out}} = \eta_{\text{isen}} w_{s,\text{out}} = (0.80)(165.3 \text{ Btu/lbm}) = \mathbf{132.2 \text{ Btu/lbm}}$$

The internal energy at the final state is determined from

$$w_{\text{out}} = u_1 - u_2 \longrightarrow u_2 = u_1 - w_{\text{out}} = (1233.7 - 132.2)\text{Btu/lbm} = 1101.4 \text{ Btu/lbm}$$

Using this internal energy and the pressure at the final state, the temperature is determined from Table A-6 to be

$$\left.\begin{array}{l} P_2 = 10 \text{ psia} \\ u_2 = 1101.4 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{274.6°F}$$

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7-123 Steam is expanded in an adiabatic turbine with an isentropic efficiency of 0.92. The power output of the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

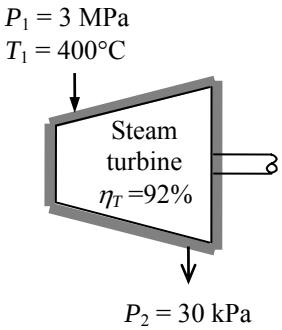
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2)$$



From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3231.7 \text{ kJ/kg} \\ s_1 = 6.9235 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 30 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{6.9235 - 0.9441}{6.8234} = 0.8763 \\ h_{2s} = h_f + x_{2s} h_{fg} = 289.27 + (0.8763)(2335.3) = 2335.7 \text{ kJ/kg} \end{array}$$

The actual power output may be determined by multiplying the isentropic power output with the isentropic efficiency. Then,

$$\begin{aligned} \dot{W}_{a,\text{out}} &= \eta_T \dot{W}_{s,\text{out}} \\ &= \eta_T \dot{m}(h_1 - h_{2s}) \\ &= (0.92)(2 \text{ kg/s})(3231.7 - 2335.7)\text{kJ/kg} \\ &= \mathbf{1649 \text{ kW}} \end{aligned}$$

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7-124 Steam is expanded in an adiabatic turbine with an isentropic efficiency of 0.85. The power output of the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

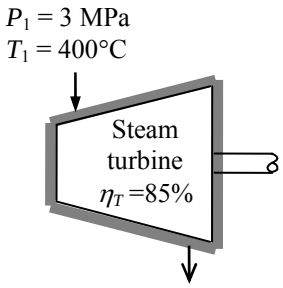
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2)$$



From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3231.7 \text{ kJ/kg} \\ s_1 = 6.9235 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 30 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{6.9235 - 0.9441}{6.8234} = 0.8763 \\ h_{2s} = h_f + x_{2s} h_{fg} = 289.27 + (0.8763)(2335.3) = 2335.7 \text{ kJ/kg} \end{array}$$

The actual power output may be determined by multiplying the isentropic power output with the isentropic efficiency. Then,

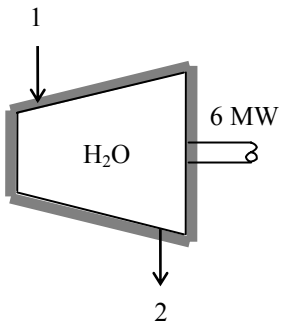
$$\begin{aligned} \dot{W}_{a,\text{out}} &= \eta_T \dot{W}_{s,\text{out}} \\ &= \eta_T \dot{m}(h_1 - h_{2s}) \\ &= (0.85)(2 \text{ kg/s})(3231.7 - 2335.7)\text{kJ/kg} \\ &= \mathbf{1523 \text{ kW}} \end{aligned}$$

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7-125 Steam enters an adiabatic turbine at a specified state, and leaves at a specified state. The mass flow rate of the steam and the isentropic efficiency are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis (*a*) From the steam tables (Tables A-4 and A-6),



$$\left.\begin{array}{l} P_1 = 7\ \text{MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3650.6\ \text{kJ/kg} \\ s_1 = 7.0910\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\ \text{kPa} \\ T_2 = 150°\text{C} \end{array}\right\} h_{2a} = 2780.2\ \text{kJ/kg}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}}^{\not{0}\ \text{(steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{a,out}} + \dot{m}(h_2 + V_1^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate of the steam is determined to be

$$6000\ \text{kJ/s} = -\dot{m}\left(2780.2 - 3650.6 + \frac{(140\ \text{m/s})^2 - (80\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = \mathbf{6.95\ kg/s}$$

(*b*) The isentropic exit enthalpy of the steam and the power output of the isentropic turbine are

$$\left.\begin{array}{l} P_{2s} = 50\ \text{kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{7.0910 - 1.0912}{6.5019} = 0.9228 \\ h_{2s} = h_f + x_{2s} h_{fg} = 340.54 + (0.9228)(2304.7) = 2467.3\ \text{kJ/kg} \end{array}$$

and

$$\dot{W}_{\text{s,out}} = -\dot{m}\left(h_{2s} - h_1 + \left\{\left(V_2^2 - V_1^2\right)/2\right\}\right)$$

$$\dot{W}_{\text{s,out}} = -(6.95\ \text{kg/s})\left(2467.3 - 3650.6 + \frac{(140\ \text{m/s})^2 - (80\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right)$$

$$= 8174\ \text{kW}$$

Then the isentropic efficiency of the turbine becomes

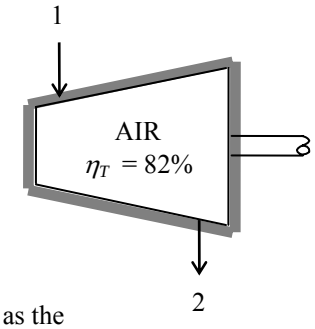
$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{6000\ \text{kW}}{8174\ \text{kW}} = 0.734 = \mathbf{73.4\%}$$

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7-126E Combustion gases enter an adiabatic gas turbine with an isentropic efficiency of 82% at a specified state, and leave at a specified pressure. The work output of the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

Analysis From the air table and isentropic relations,



$$T_1 = 2000 \text{ R} \longrightarrow \begin{matrix} h_1 = 504.71 \text{ Btu/lbm} \\ P_{r_1} = 174.0 \end{matrix}$$

$$P_{r_2} = \left(\frac{P_2}{P_1}\right) P_{r_1} = \left(\frac{60 \text{ psia}}{120 \text{ psia}}\right)(174.0) = 87.0 \longrightarrow h_{2s} = 417.3 \text{ Btu/lbm}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}h_1 = \dot{W}_{\text{a,out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{W}_{\text{a,out}} = \dot{m}(h_1 - h_2)$$

Noting that $w_a = \eta_T w_s$, the work output of the turbine per unit mass is determined from

$$w_a = (0.82)(504.71 - 417.3)\text{Btu/lbm} = \mathbf{71.7\ Btu/lbm}$$

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7-127 Air is compressed by an adiabatic compressor with an isentropic efficiency of 95%. The power input is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at the anticipated average temperature of 400 K are c_p = 1.013 kJ/kg·°C and k = 1.395 (Table A-2b).

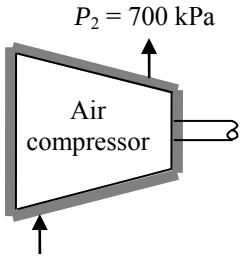
Analysis We take the compressor as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the compressor, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

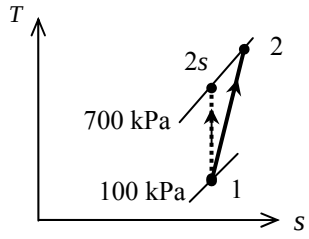


The isentropic exit temperature is

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (293\text{ K})\left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)^{0.395/1.395} = 508.3\text{ K}$$

The power input during isentropic process would be

$$\dot{W}_{\text{s,in}} = \dot{m}c_p(T_{2s} - T_1) = (2\text{ kg/s})(1.013\text{ kJ/kg}\cdot\text{K})(508.3 - 293)\text{K} = 436.3\text{ kW}$$



The power input during the actual process is

$$\dot{W}_{\text{in}} = \frac{\dot{W}_{\text{s,in}}}{\eta_C} = \frac{436.3\text{ kW}}{0.95} = \mathbf{459.3\text{ kW}}$$

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7-128 Steam is expanded in an adiabatic turbine. The isentropic efficiency is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

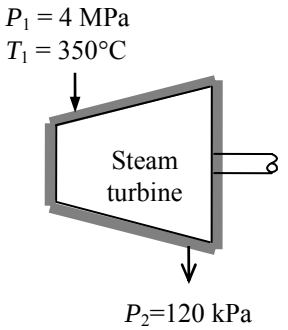
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2)$$

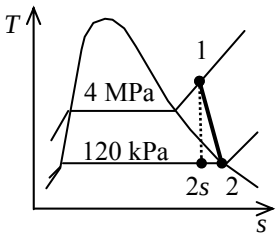


From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 4 \text{ MPa} \\ T_1 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3093.3 \text{ kJ/kg} \\ s_1 = 6.5843 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 120 \text{ kPa} \\ x_2 = 1 \end{array}\right\} h_2 = 2683.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{2s} = 120 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = 0.8798 \\ h_{2s} = 2413.4 \text{ kJ/kg} \end{array}$$



From the definition of the isentropic efficiency,

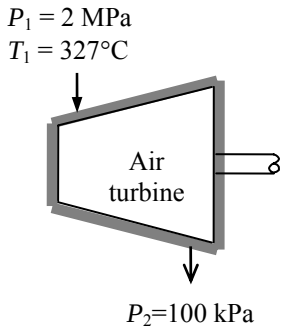
$$\eta_T = \frac{\dot{W}_{a,\text{out}}}{\dot{W}_{s,\text{out}}} = \frac{\dot{m}(h_1 - h_2)}{\dot{m}(h_1 - h_{2s})} = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{3093.3 - 2683.1}{3093.3 - 2413.4} = \mathbf{0.603} = \mathbf{60.3\%}$$

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7-129 Air is expanded by an adiabatic turbine. The isentropic efficiency is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at the anticipated average temperature of 400 K are $c_p = 1.013$ kJ/kg·°C and $k = 1.395$ (Table A-2a).



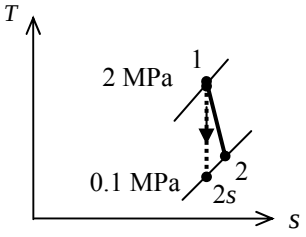
Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the turbine, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$



The isentropic exit temperature is

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (327 + 273\text{ K})\left(\frac{100\text{ kPa}}{2000\text{ kPa}}\right)^{0.395/1.395} = 256.9\text{ K}$$

From the definition of the isentropic efficiency,

$$\eta_T = \frac{w_{a,\text{out}}}{w_{s,\text{out}}} = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{c_p(T_1 - T_2)}{c_p(T_1 - T_{2s})} = \frac{T_1 - T_2}{T_1 - T_{2s}} = \frac{600 - 273}{900 - 256.9} = \mathbf{0.953 = 95.3\%}$$

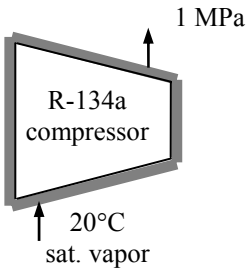
7-130 R-134a is compressed by an adiabatic compressor with an isentropic efficiency of 85%. The power required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis From the R-134a tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} T_1 = 20°\text{C} \\ x = 1\text{ (sat. vap.)} \end{array}\right\} \begin{array}{l} h_1 = 261.59\text{ kJ/kg} \\ s_1 = 0.9223\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1000\text{ kPa} \\ s_{2s} = s_1 = 0.9223\text{ kJ/kg}\cdot\text{K} \end{array}\right\} h_{2s} = 273.11\text{ kJ/kg}$$



The power input during isentropic process would be

$$\dot{W}_{\text{s,in}} = \dot{m}(h_{2s} - h_1) = (0.5\text{ kg/s})(273.11 - 261.59)\text{ kJ/kg} = 5.76\text{ kW}$$

The power input during the actual process is

$$\dot{W}_{\text{in}} = \frac{\dot{W}_{\text{s,in}}}{\eta_C} = \frac{5.76\text{ kW}}{0.85} = \mathbf{6.78\text{ kW}}$$

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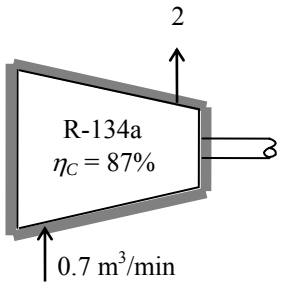
7-131 Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The compressor exit temperature and power input to the compressor are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis (a) From the refrigerant tables (Tables A-11E through A-13E),

$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@100\text{ kPa}} = 234.44 \text{ kJ/kg} \\ s_1 = s_{g@100\text{ kPa}} = 0.95183 \text{ kJ/kg}\cdot\text{K} \\ \nu_1 = \nu_{g@100\text{ kPa}} = 0.19254 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 282.51 \text{ kJ/kg}$$



From the isentropic efficiency relation,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_C = 234.44 + (282.51 - 234.44)/0.87 = 289.69 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 1 \text{ MPa} \\ h_{2a} = 289.69 \text{ kJ/kg} \end{array}\right\} T_{2a} = \mathbf{56.5°C}$$

(b) The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{V}_1}{\nu_1} = \frac{0.7/60 \text{ m}^3/\text{s}}{0.19254 \text{ m}^3/\text{kg}} = 0.06059 \text{ kg/s}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\not{0}\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor becomes,

$$\dot{W}_{\text{a,in}} = (0.06059 \text{ kg/s})(289.69 - 234.44)\text{kJ/kg} = \mathbf{3.35\ kW}$$

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7-132 Problem 7-131 is reconsidered. The problem is to be solved by considering the kinetic energy and by assuming an inlet-to-exit area ratio of 1.5 for the compressor when the compressor exit pipe inside diameter is 2 cm.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data from diagram window"
{P[1] = 100 [kPa]
P[2] = 1000 [kPa]
Vol_dot_1 = 0.7 [m^3/min]
Eta_c = 0.87 "Compressor adiabatic efficiency"
A_ratio = 1.5
d_2 = 0.02 [m]}

"System: Control volume containing the compressor, see the diagram window.
Property Relation:  Use the real fluid properties for R134a.
Process: Steady-state, steady-flow, adiabatic process."
Fluid$='R134a'
"Property Data for state 1"
T[1]=temperature(Fluid$,P=P[1],x=1)"Real fluid equ. at the sat. vapor state"
h[1]=enthalpy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
s[1]=entropy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
v[1]=volume(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"

"Property Data for state 2"
s_s[1]=s[1]; T_s[1]=T[1] "needed for plot"
s_s[2]=s[1] "for the ideal, isentropic process across the compressor"
h_s[2]=ENTHALPY(Fluid$, P=P[2], s=s_s[2])"Enthalpy 2 at the isentropic state 2s and pressure P[2]"
T_s[2]=Temperature(Fluid$, P=P[2], s=s_s[2])"Temperature of ideal state - needed only for plot."
"Steady-state, steady-flow conservation of mass"
 m_dot_1 = m_dot_2
m_dot_1 = Vol_dot_1/(v[1]*60)
Vol_dot_1/v[1]=Vol_dot_2/v[2]
Vel[2]=Vol_dot_2/(A[2]*60)
A[2] = pi*(d_2)^2/4
A_ratio*Vel[1]/v[1] = Vel[2]/v[2] "Mass flow rate: = A*Vel/v, A_ratio = A[1]/A[2]"
A_ratio=A[1]/A[2]

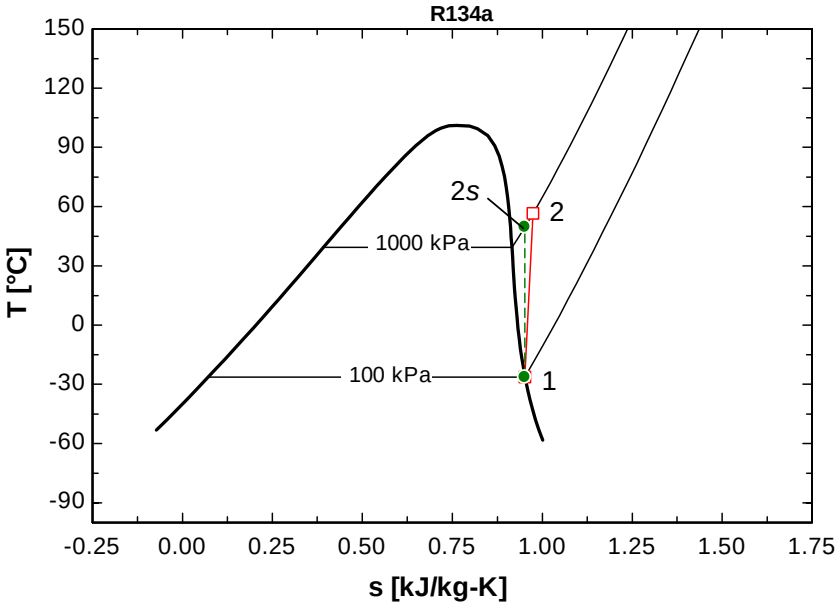
"Steady-state, steady-flow conservation of energy, adiabatic compressor, see diagram window"
m_dot_1*(h[1]+(Vel[1])^2/(2*1000)) + W_dot_c= m_dot_2*(h[2]+(Vel[2])^2/(2*1000))

"Definition of the compressor isentropic efficiency, Eta_c=W_isen/W_act"
Eta_c = (h_s[2]-h[1])/(h[2]-h[1])

"Knowing h[2], the other properties at state 2 can be found."
v[2]=volume(Fluid$, P=P[2], h=h[2])"v[2] is found at the actual state 2, knowing P and h."
T[2]=temperature(Fluid$, P=P[2],h=h[2])"Real fluid equ. for T at the known outlet h and P."
s[2]=entropy(Fluid$, P=P[2], h=h[2])   "Real fluid equ. at the known outlet h and P."
T_exit=T[2]
"Neglecting the kinetic energies, the work is:"
m_dot_1*h[1] + W_dot_c_noke= m_dot_2*h[2]
```

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SOLUTION
A_ratio=1.5
d_2=0.02 [m]
Eta_c=0.87
Fluid$='R134a'
m_dot_1=0.06059 [kg/s]
m_dot_2=0.06059 [kg/s]
T_exit=56.51 [C]
Vol_dot_1=0.7 [m^3 /min]
Vol_dot_2=0.08229 [m^3 /min]
W_dot_c=3.33 [kW]
W_dot_c_noke=3.348 [kW]



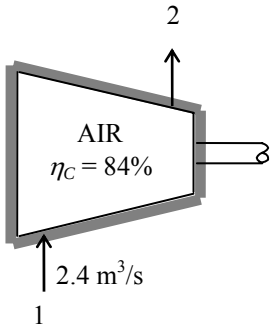
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7-133 Air enters an adiabatic compressor with an isentropic efficiency of 84% at a specified state, and leaves at a specified temperature. The exit pressure of air and the power input to the compressor are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1)

Analysis (*a*) From the air table (Table A-17),



$$T_1 = 290 \text{ K} \longrightarrow h_1 = 290.16 \text{ kJ/kg},\ P_{r1} = 1.2311$$
$$T_2 = 530 \text{ K} \longrightarrow h_{2a} = 533.98 \text{ kJ/kg}$$

From the isentropic efficiency relation $\eta_C = \dfrac{h_{2s} - h_1}{h_{2a} - h_1}$,

$$h_{2s} = h_1 + \eta_C (h_{2a} - h_1)$$
$$= 290.16 + (0.84)(533.98 - 290.16) = 495.0 \text{ kJ/kg} \longrightarrow P_{r_2} = 7.951$$

Then from the isentropic relation ,

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right) P_1 = \left(\frac{7.951}{1.2311}\right)(100 \text{ kPa}) = \mathbf{646 \text{ kPa}}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

where

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(100 \text{ kPa})(2.4 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 2.884 \text{ kg/s}$$

Then the power input to the compressor is determined to be

$$\dot{W}_{\text{a,in}} = (2.884 \text{ kg/s})(533.98 - 290.16) \text{ kJ/kg} = \mathbf{703 \text{ kW}}$$

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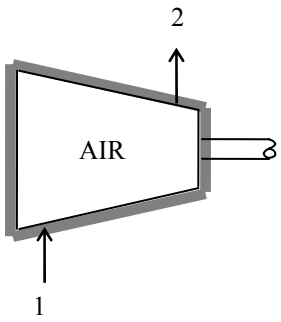
7-134 Air is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor and the exit temperature of air for the isentropic case are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Analysis (a) From the air table (Table A-17),

$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}, \qquad P_{r_1} = 1.386$$

$$T_2 = 550 \text{ K} \longrightarrow h_{2a} = 554.74 \text{ kJ/kg}$$



From the isentropic relation,

$$P_{r_2} = \left(\frac{P_2}{P_1}\right) P_{r_1} = \left(\frac{600 \text{ kPa}}{95 \text{ kPa}}\right)(1.386) = 8.754 \longrightarrow h_{2s} = 508.72 \text{ kJ/kg}$$

Then the isentropic efficiency becomes

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{508.72 - 300.19}{554.74 - 300.19} = 0.819 = \mathbf{81.9\%}$$

(b) If the process were isentropic, the exit temperature would be

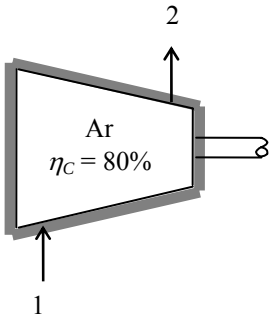
$$h_{2s} = 508.72 \text{ kJ/kg} \longrightarrow T_{2s} = \mathbf{505.5 \text{ K}}$$

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7-135E Argon enters an adiabatic compressor with an isentropic efficiency of 80% at a specified state, and leaves at a specified pressure. The exit temperature of argon and the work input to the compressor are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

Properties The specific heat ratio of argon is $k = 1.667$. The constant pressure specific heat of argon is $c_p = 0.1253$ Btu/lbm.R (Table A-2E).



Analysis (*a*) The isentropic exit temperature T_{2s} is determined from

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (550\ \text{R})\left(\frac{200\ \text{psia}}{20\ \text{psia}}\right)^{0.667/1.667} = 1381.9\ \text{R}$$

The actual kinetic energy change during this process is

$$\Delta ke_a = \frac{V_2^2 - V_1^2}{2} = \frac{(240\ \text{ft/s})^2 - (60\ \text{ft/s})^2}{2}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right) = 1.08\ \text{Btu/lbm}$$

The effect of kinetic energy on isentropic efficiency is very small. Therefore, we can take the kinetic energy changes for the actual and isentropic cases to be same in efficiency calculations. From the isentropic efficiency relation, including the effect of kinetic energy,

$$\eta_C = \frac{w_s}{w_a} = \frac{(h_{2s} - h_1) + \Delta ke}{(h_{2a} - h_1) + \Delta ke} = \frac{c_p(T_{2s} - T_1) + \Delta ke_s}{c_p(T_{2a} - T_1) + \Delta ke_a} \longrightarrow 0.8 = \frac{0.1253(1381.9 - 550) + 1.08}{0.1253(T_{2a} - 550) + 1.08}$$

It yields

$$T_{2a} = \mathbf{1592\ R}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right) \longrightarrow w_{\text{a,in}} = h_2 - h_1 + \Delta\text{ke}$$

Substituting, the work input to the compressor is determined to be

$$w_{\text{a,in}} = (0.1253\ \text{Btu/lbm} \cdot \text{R})(1592 - 550)\text{R} + 1.08\ \text{Btu/lbm} = \mathbf{131.6\ Btu/lbm}$$

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7-136E Air is accelerated in a 85% efficient adiabatic nozzle from low velocity to a specified velocity. The exit temperature and pressure of the air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Analysis From the air table (Table A-17E),

$$T_1 = 1400\ \text{R} \longrightarrow h_1 = 342.90\ \text{Btu/lbm},\ P_{r_1} = 42.88$$

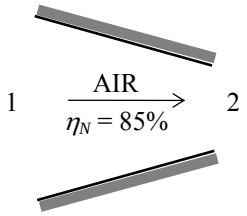
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$h_2 = h_1 - \frac{V_2^2 - V_1^{2\,\nearrow 0}}{2}$$



Substituting, the exit temperature of air is determined to be

$$h_2 = 342.90\ \text{kJ/kg} - \frac{(650\ \text{ft/s})^2 - 0}{2}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right) = 334.46\ \text{Btu/lbm}$$

From the air table we read

$$T_{2a} = \mathbf{1368\ R = 908°F}$$

From the isentropic efficiency relation

$$\eta_N = \frac{h_{2a} - h_1}{h_{2s} - h_1} \longrightarrow$$

$$h_{2s} = h_1 + (h_{2a} - h_1)/\eta_N = 342.90 + (334.46 - 342.90)/(0.85) = 332.97\ \text{Btu/lbm} \longrightarrow P_{r_2} = 38.62$$

Then the exit pressure is determined from the isentropic relation to be

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right)P_1 = \left(\frac{38.62}{42.88}\right)(45\ \text{psia}) = \mathbf{40.5\ psia}$$

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7-137E Problem 7-136E is reconsidered. The effect of varying the nozzle isentropic efficiency from 0.8 to 1.0 on the exit temperature and pressure of the air is to be investigated, and the results are to be plotted.

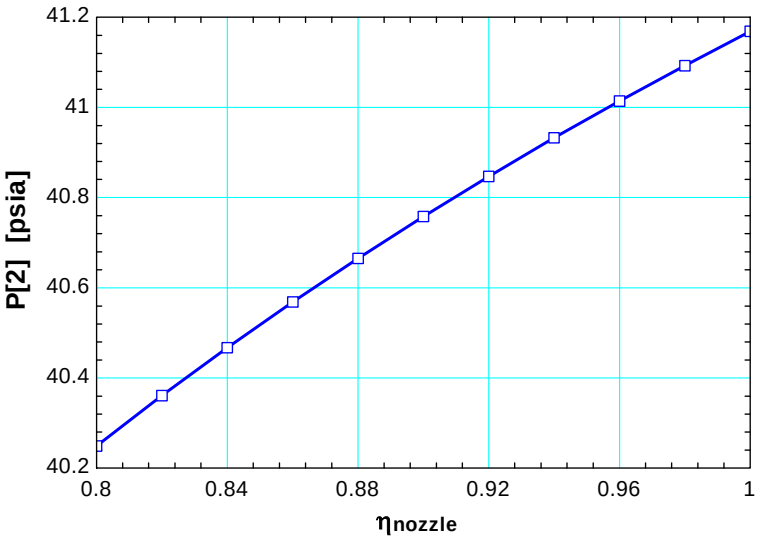
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

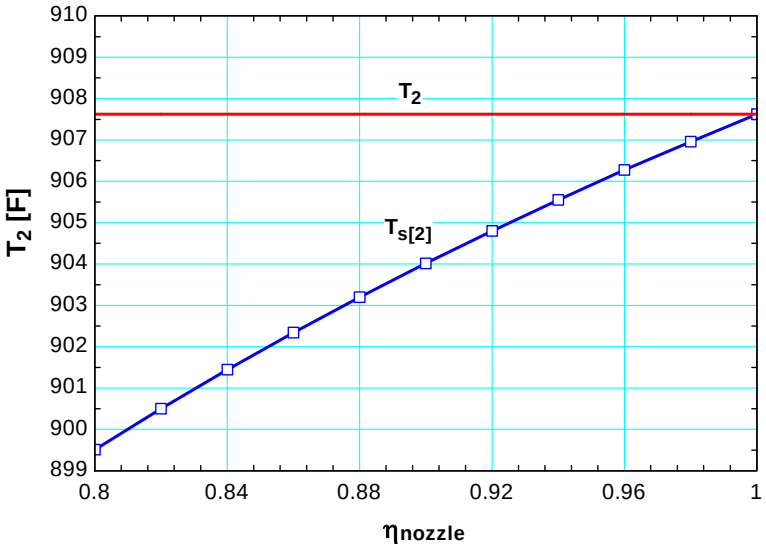
```
"Knowns:"
WorkFluid$ = 'Air'
P[1] = 45  [psia]
T[1] = 940 [F]
Vel[2] = 650 [ft/s]
Vel[1] = 0 [ft/s]
eta_nozzle = 0.85

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential energy, no  heat transfer:"
h[1]=enthalpy(WorkFluid$,T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
T_s[1] = T[1]
s[2] =s[1]
s_s[2] = s[1]
h_s[2]=enthalpy(WorkFluid$,T=T_s[2])
T_s[2]=temperature(WorkFluid$,P=P[2],s=s_s[2])
eta_nozzle = ke[2]/ke_s[2]
ke[1] = Vel[1]^2/2
ke[2]=Vel[2]^2/2
h[1]+ke[1]*convert(ft^2/s^2,Btu/lbm) = h[2] + ke[2]*convert(ft^2/s^2,Btu/lbm)
h[1] +ke[1]*convert(ft^2/s^2,Btu/lbm) = h_s[2] + ke_s[2]*convert(ft^2/s^2,Btu/lbm)
T[2]=temperature(WorkFluid$,h=h[2])
P_2_answer = P[2]
T_2_answer = T[2]
```

η_{nozzle}	P_2 [psia]	T_2 [F	$T_{s,2}$ [F]
0.8	40.25	907.6	899.5
0.82	40.36	907.6	900.5
0.84	40.47	907.6	901.4
0.86	40.57	907.6	902.3
0.88	40.67	907.6	903.2
0.9	40.76	907.6	904
0.92	40.85	907.6	904.8
0.94	40.93	907.6	905.6
0.96	41.01	907.6	906.3
0.98	41.09	907.6	907
1	41.17	907.6	907.6

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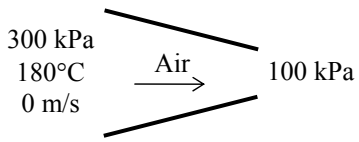
7-138 Air is expanded in an adiabatic nozzle with an isentropic efficiency of 0.96. The air velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).

Analysis For the isentropic process of an ideal gas, the exit temperature is determined from

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (180+273\,\text{K})\left(\frac{100\,\text{kPa}}{300\,\text{kPa}}\right)^{0.4/1.4} = 331.0\,\text{K}$$



There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$h_1 - h_2 = \frac{V_2^2 - V_1^2}{2}$$

$$c_p(T_1 - T_2) = \frac{V_2^2 - V_1^2}{2} = \Delta\text{ke}$$

The kinetic energy change for the isentropic case is

$$\Delta\text{ke}_s = c_p(T_1 - T_{2s}) = (1.005\,\text{kJ/kg}\cdot\text{K})(453-331)\text{K} = 122.6\,\text{kJ/kg}$$

The kinetic energy change for the actual process is

$$\Delta\text{ke}_a = \eta_N \Delta\text{ke}_s = (0.96)(122.6\,\text{kJ/kg}) = 117.7\,\text{kJ/kg}$$

Substituting into the energy balance and solving for the exit velocity gives

$$V_2 = (2\Delta\text{ke}_a)^{0.5} = \left[2(117.7\,\text{kJ/kg})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)\right]^{0.5} = \mathbf{485\ m/s}$$

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7-139E Air is decelerated in an adiabatic diffuser with an isentropic efficiency of 0.82. The air velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** There is no heat transfer or shaft work associated with the process. **3** Air is an ideal gas with constant specific heats.

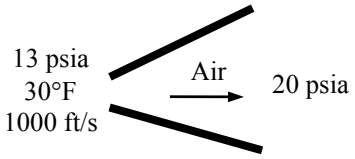
Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis For the isentropic process of an ideal gas, the exit temperature is determined from

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (30+460\text{ R})\left(\frac{20\text{ psia}}{13\text{ psia}}\right)^{0.4/1.4} = 554.2\text{ R}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$h_1 - h_2 = \frac{V_2^2 - V_1^2}{2}$$

$$c_p(T_1 - T_2) = \frac{V_2^2 - V_1^2}{2} = \Delta\text{ke}$$

The kinetic energy change for the isentropic case is

$$\Delta\text{ke}_s = c_p(T_{2s} - T_1) = (0.240\text{ Btu/lbm}\cdot\text{R})(554.2 - 490)\text{R} = 15.41\text{ Btu/lbm}$$

The kinetic energy change for the actual process is

$$\Delta\text{ke}_a = \eta_N \Delta\text{ke}_s = (0.82)(15.41\text{ Btu/lbm}) = 12.63\text{ Btu/lbm}$$

Substituting into the energy balance and solving for the exit velocity gives

$$V_2 = (V_1^2 - 2\Delta\text{ke}_a)^{0.5} = \left[(1000\text{ ft/s})^2 - 2(12.63\text{ Btu/lbm})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)\right]^{0.5} = \mathbf{606\text{ ft/s}}$$

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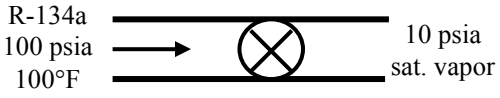
Entropy Balance

7-140E Refrigerant-134a is expanded adiabatically from a specified state to another. The entropy generation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The rate of entropy generation within the expansion device during this process can be determined by applying the rate form of the entropy balance on the system. Noting that the system is adiabatic and thus there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$



$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

$$s_{\text{gen}} = s_2 - s_1$$

The properties of the refrigerant at the inlet and exit states are (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 100 \text{ psia} \\ T_1 = 100°\text{F} \end{array}\right\} \; s_1 = 0.22900 \text{ Btu/lbm} \cdot \text{R}$$

$$\left.\begin{array}{l} P_2 = 10 \text{ psia} \\ x_2 = 1 \end{array}\right\} \; s_2 = 0.22948 \text{ Btu/lbm} \cdot \text{R}$$

Substituting,

$$s_{\text{gen}} = s_2 - s_1 = 0.22948 - 0.22900 = \mathbf{0.00048 \text{ Btu/lbm} \cdot \text{R}}$$

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7-141 Oxygen is cooled as it flows in an insulated pipe. The rate of entropy generation in the pipe is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The pipe is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies are negligible. **4** Oxygen is an ideal gas with constant specific heats.

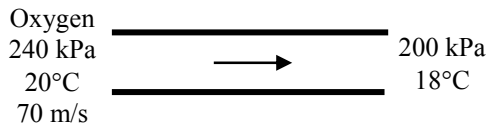
Properties The properties of oxygen at room temperature are $R = 0.2598$ kJ/kg·K, $c_p = 0.918$ kJ/kg·K (Table A-2a).

Analysis The rate of entropy generation in the pipe is determined by applying the rate form of the entropy balance on the pipe:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$



The specific volume of oxygen at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.2598\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{240\,\text{kPa}} = 0.3172\,\text{m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{\pi D^2 V_1}{4\nu_1} = \frac{\pi(0.12\,\text{m})^2(70\,\text{m/s})}{4(0.3172\,\text{m}^3/\text{kg})} = 2.496\,\text{kg/s}$$

Substituting into the entropy balance relation,

$$\begin{aligned}\dot{S}_{\text{gen}} &= \dot{m}(s_2 - s_1)\\ &= \dot{m}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)\\ &= (2.496\,\text{kg/s})\left[(0.918\,\text{kJ/kg}\cdot\text{K})\ln\frac{291\,\text{K}}{293\,\text{K}} - (0.2598\,\text{kJ/kg}\cdot\text{K})\ln\frac{200\,\text{kPa}}{240\,\text{kPa}}\right]\\ &= \mathbf{0.1025\ kW/K}\end{aligned}$$

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7-142 Nitrogen is compressed by an adiabatic compressor. The entropy generation for this process is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The compressor is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies are negligible. **4** Nitrogen is an ideal gas with constant specific heats.

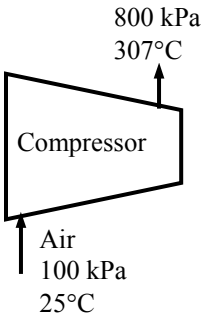
Properties The specific heat of nitrogen at the average temperature of (25+307)/2=166°C = 439 K is c_p = 1.048 kJ/kg·K (Table A-2b). Also, R = 0.2968 kJ/kg·K (Table A-2a).

Analysis The rate of entropy generation in the pipe is determined by applying the rate form of the entropy balance on the compressor:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$



Substituting per unit mass of the oxygen,

$$\begin{aligned} s_{\text{gen}} &= s_2 - s_1 \\ &= c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \\ &= (1.048\,\text{kJ/kg}\cdot\text{K}) \ln \frac{(307+273)\,\text{K}}{(25+273)\,\text{K}} - (0.2968\,\text{kJ/kg}\cdot\text{K}) \ln \frac{800\,\text{kPa}}{100\,\text{kPa}} \\ &= \mathbf{0.0807\ kJ/kg\cdot K} \end{aligned}$$

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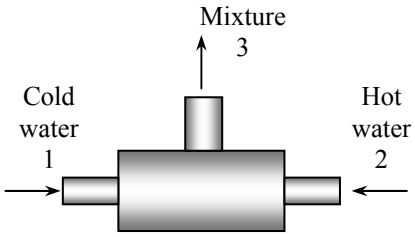
7-143 Each member of a family of four take a 5-min shower every day. The amount of entropy generated by this family per year is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The kinetic and potential energies are negligible. **3** Heat losses from the pipes and the mixing section are negligible and thus $\dot{Q} \cong 0$. **4** Showers operate at maximum flow conditions during the entire shower. **5** Each member of the household takes a 5-min shower every day. **6** Water is an incompressible substance with constant properties at room temperature. **7** The efficiency of the electric water heater is 100%.

Properties The density and specific heat of water at room temperature are $\rho = 1$ kg/L and $c = 4.18$ kJ/kg.°C (Table A-3).

Analysis The mass flow rate of water at the shower head is

$$\dot{m} = \rho \dot{V} = (1\,\text{kg/L})(12\,\text{L/min}) = 12\,\text{kg/min}$$



The mass balance for the mixing chamber can be expressed in the rate form as

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

where the subscript 1 denotes the cold water stream, 2 the hot water stream, and 3 the mixture.

The rate of entropy generation during this process can be determined by applying the rate form of the entropy balance on a system that includes the electric water heater and the mixing chamber (the T-elbow). Noting that there is no entropy transfer associated with work transfer (electricity) and there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0 \text{ and work is entropy free})$$

$$\dot{S}_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

Noting from mass balance that $\dot{m}_1 + \dot{m}_2 = \dot{m}_3$ and $s_2 = s_1$ since hot water enters the system at the same temperature as the cold water, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_3 s_3 - (\dot{m}_1 + \dot{m}_2) s_1 = \dot{m}_3 (s_3 - s_1) = \dot{m}_3 c_p \ln \frac{T_3}{T_1}$$

$$= (12\,\text{kg/min})(4.18\,\text{kJ/kg.K}) \ln \frac{42 + 273}{15 + 273} = 4.495\,\text{kJ/min.K}$$

Noting that 4 people take a 5-min shower every day, the amount of entropy generated per year is

$$S_{\text{gen}} = (\dot{S}_{\text{gen}}) \Delta t (\text{No. of people})(\text{No. of days})$$

$$= (4.495\,\text{kJ/min.K})(5\,\text{min/person} \cdot \text{day})(4\,\text{persons})(365\,\text{days/year})$$

$$= \mathbf{32{,}814\,kJ/K} \quad (\text{per year})$$

Discussion The value above represents the entropy generated within the water heater and the T-elbow in the absence of any heat losses. It does not include the entropy generated as the shower water at 42°C is discarded or cooled to the outdoor temperature. Also, an entropy balance on the mixing chamber alone (hot water entering at 55°C instead of 15°C) will exclude the entropy generated within the water heater.

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7-144 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the rate of entropy generation within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

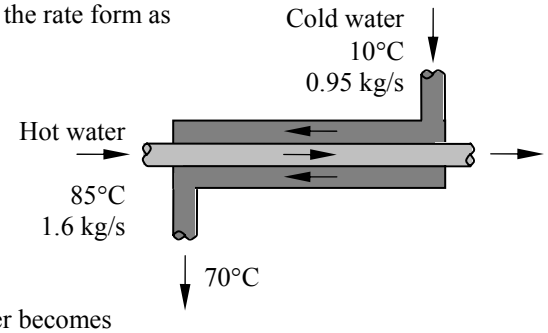
Analysis We take the cold water tubes as the system, which is a control volume.
The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q}_{\text{in}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cold water}} = (0.95 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70°\text{C} - 10°\text{C}) = \mathbf{238.3\ kW}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{hot water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 85°\text{C} - \frac{238.3 \text{ kW}}{(1.6 \text{ kg/s})(4.19 \text{ kJ/kg.°C})} = 49.5°\text{C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\swarrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{cold}} s_1 + \dot{m}_{\text{hot}} s_3 - \dot{m}_{\text{cold}} s_2 - \dot{m}_{\text{hot}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{cold}}(s_2 - s_1) + \dot{m}_{\text{hot}}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{cold}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{hot}} c_p \ln\frac{T_4}{T_3}$$

$$= (0.95 \text{ kg/s})(4.18 \text{ kJ/kg.K})\ln\frac{70 + 273}{10 + 273} + (1.6 \text{ kg/s})(4.19 \text{ kJ/kg.K})\ln\frac{49.5 + 273}{85 + 273}$$

$$= \mathbf{0.06263\ kW/K}$$

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7-145 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer, the outlet temperature of the air, and the rate of entropy generation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively. The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

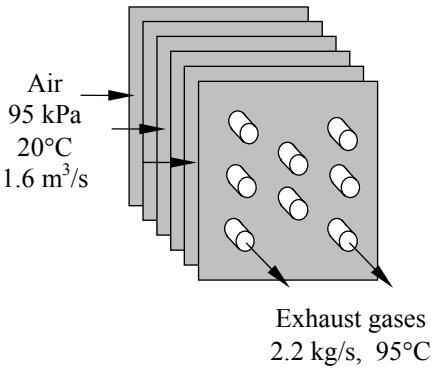
Analysis We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{gas.}} = (2.2 \text{ kg/s})(1.1 \text{ kJ/kg.°C})(180°\text{C} - 95°\text{C}) = \mathbf{205.7\ kW}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95 \text{ kPa})(1.6 \text{ m}^3/\text{s})}{(0.287 \text{ kPa.m}^3/\text{kg.K}) \times (293 \text{ K})} = 1.808 \text{ kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

$$\dot{Q} = \left[\dot{m}c_p(T_{\text{out}} - T_{\text{in}})\right]_{\text{air}} \longrightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{205.7 \text{ kW}}{(1.808 \text{ kg/s})(1.005 \text{ kJ/kg.°C})} = \mathbf{133.2°C}$$

The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not{\searrow} 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{exhaust}} s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_{\text{exhaust}} s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{exhaust}}(s_2 - s_1) + \dot{m}_{\text{air}}(s_4 - s_3)$$

Then the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{exhaust}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{air}} c_p \ln\frac{T_4}{T_3}$$

$$= (2.2 \text{ kg/s})(1.1 \text{ kJ/kg.K})\ln\frac{95 + 273}{180 + 273} + (1.808 \text{ kg/s})(1.005 \text{ kJ/kg.K})\ln\frac{133.2 + 273}{20 + 273}$$

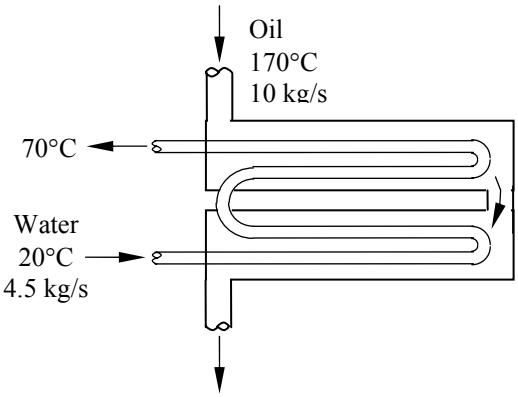
$$= \mathbf{0.091\ kW/K}$$

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7-146 Water is heated by hot oil in a heat exchanger. The outlet temperature of the oil and the rate of entropy generation within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.



Analysis (*a*) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (4.5\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = 940.5\text{ kW}$$

Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} \rightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5\text{ kW}}{(10\text{ kg/s})(2.3\text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{oil}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{oil}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{oil}}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{oil}} c_p \ln\frac{T_4}{T_3}$$

$$= (4.5\text{ kg/s})(4.18\text{ kJ/kg.K})\ln\frac{70+273}{20+273} + (10\text{ kg/s})(2.3\text{ kJ/kg.K})\ln\frac{129.1+273}{170+273}$$

$$= \mathbf{0.736\text{ kW/K}}$$

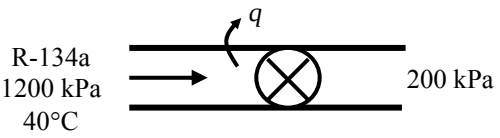
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7-147 Heat is lost from Refrigerant-134a as it is throttled. The exit temperature of the refrigerant and the entropy generation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The properties of the refrigerant at the inlet of the device are (Table A-13)

$$\left.\begin{array}{l} P_1 = 1200 \text{ kPa} \\ T_1 = 40°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 108.23 \text{ kJ/kg} \\ s_1 = 0.39424 \text{ kJ/kg} \end{array}$$



The enthalpy of the refrigerant at the exit of the device is

$$h_2 = h_1 - q_{\text{out}} = 108.23 - 0.5 = 107.73 \text{ kJ/kg}$$

Now, the properties at the exit state may be obtained from the R-134a tables

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ h_2 = 107.73 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{-10.09°C} \\ s_2 = 0.41800 \text{ kJ/kg.K} \end{array}$$

The entropy generation associated with this process may be obtained by adding the entropy change of R-134a as it flows in the device and the entropy change of the surroundings.

$$\Delta s_{\text{R-134a}} = s_2 - s_1 = 0.41800 - 0.39424 = 0.02375 \text{ kJ/kg.K}$$

$$\Delta s_{\text{surr}} = \frac{q_{\text{out}}}{T_{\text{surr}}} = \frac{0.5 \text{ kJ/kg}}{(25 + 273) \text{ K}} = 0.001678 \text{ kJ/kg.K}$$

$$s_{\text{gen}} = \Delta s_{\text{total}} = \Delta s_{\text{R-134a}} + \Delta s_{\text{surr}} = 0.02375 + 0.001678 = \mathbf{0.02543 \text{ kJ/kg.K}}$$

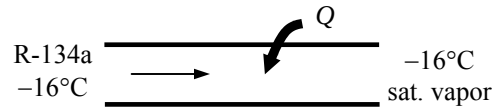
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7-148 In an ice-making plant, water is frozen by evaporating saturated R-134a liquid. The rate of entropy generation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis We take the control volume formed by the R-134a evaporator with a single inlet and single exit as the system. The rate of entropy generation within this evaporator during this process can be determined by applying the rate form of the entropy balance on the system. The entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0 \text{ (steady)}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$



$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{\text{in}}}{T_w} + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_R (s_2 - s_1) - \frac{\dot{Q}_{\text{in}}}{T_w}$$

$$\dot{S}_{\text{gen}} = \dot{m}_R s_{fg} - \frac{\dot{Q}_{\text{in}}}{T_w}$$

The properties of the refrigerant are (Table A-11)

$$h_{fg\,@\,-16°\text{C}} = 210.18\,\text{kJ/kg}$$
$$s_{fg\,@\,-16°\text{C}} = 0.81729\,\text{kJ/kg}\cdot\text{K}$$

The rate of that must be removed from the water in order to freeze it at a rate of 4000 kg/h is

$$\dot{Q}_{\text{in}} = \dot{m}_w h_{if} = (2500 / 3600\ \text{kg/s})(333.7\ \text{kJ/kg}) = 231.7\ \text{kW}$$

where the heat of fusion of water at 1 atm is 333.7 kJ/kg. The mass flow rate of R-134a is

$$\dot{m}_R = \frac{\dot{Q}_{\text{in}}}{h_{fg}} = \frac{231.7\,\text{kJ/s}}{210.18\,\text{kJ/kg}} = 1.103\,\text{kg/s}$$

Substituting,

$$\dot{S}_{\text{gen}} = \dot{m}_R s_{fg} - \frac{\dot{Q}_{\text{in}}}{T_w} = (1.103\,\text{kg/s})(0.81729\,\text{kJ/kg}\cdot\text{K}) - \frac{231.7\,\text{kW}}{273\,\text{K}} = \mathbf{0.0528\ kW/K}$$

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7-149E Water and steam are mixed in a chamber that is losing heat at a specified rate. The rate of entropy generation during this process is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{cv} = 0$, $\Delta E_{cv} = 0$, and $\Delta S_{cv} = 0$. **2** There are no work interactions involved. **3** The kinetic and potential energies are negligible, ke = pe = 0.

Analysis We take the *mixing chamber* as the system. This is a *control volume* since mass crosses the system boundary during the process.

We note that there are two inlets and one exit. Under the stated assumptions and observations, the mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{Q}_{\text{out}}$$

Combining the mass and energy balances gives

$$\dot{Q}_{\text{out}} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3$$

The desired properties at the specified states are determined from the steam tables to be

$$\left.\begin{array}{l} P_1 = 20 \text{ psia} \\ T_1 = 50°\text{F} \end{array}\right\} \begin{array}{l} h_1 = h_{f\,@\,50°\text{F}} = 18.07 \text{ Btu/lbm} \\ s_1 = s_{f\,@\,50°\text{F}} = 0.03609 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 20 \text{ psia} \\ T_2 = 240°\text{F} \end{array}\right\} \begin{array}{l} h_2 = 1162.3 \text{ Btu/lbm} \\ s_2 = 1.7406 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_3 = 20 \text{ psia} \\ T_3 = 130°\text{F} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,130°\text{F}} = 97.99 \text{ Btu/lbm} \\ s_3 = s_{f\,@\,130°\text{F}} = 0.18174 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

Substituting,

$$180 \text{ Btu/min} = \left[300 \times 18.07 + \dot{m}_2 \times 1162.3 - (300 + \dot{m}_2) \times 97.99\right] \text{Btu/min} \longrightarrow \dot{m}_2 = 22.7 \text{ lbm/min}$$

The rate of entropy generation during this process can be determined by applying the rate form of the entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings so that the boundary temperature of the extended system is 70°F = 530 R:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{out}}{T_b} + \dot{S}_{\text{gen}} = 0$$

Substituting, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{out}}{T_b}$$

$$= (322.7 \times 0.18174 - 300 \times 0.03609 - 22.7 \times 1.7406) \text{ Btu/min} \cdot \text{R}) + \frac{180 \text{ Btu/min}}{530 \text{ R}}$$

$$= \mathbf{8.65 \text{ Btu/min} \cdot \text{R}}$$

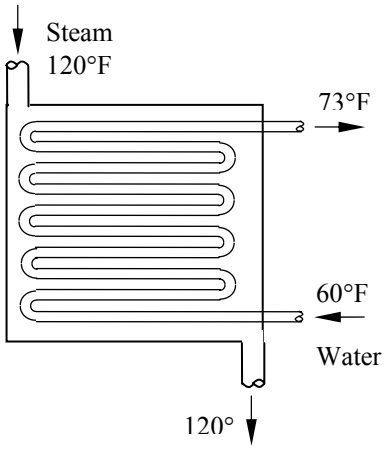
Discussion Note that entropy is generated during this process at a rate of 8.65 Btu/min · R. This entropy generation is caused by the mixing of two fluid streams (an irreversible process) and the heat transfer between the mixing chamber and the surroundings through a finite temperature difference (another irreversible process).

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7-150E Steam is condensed by cooling water in a condenser. The rate of heat transfer and the rate of entropy generation within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy and entropy of vaporization of water at 120°F are 1025.2 Btu/lbm and s_{fg} = 1.7686 Btu/lbm.R (Table A-4E).



Analysis We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (92 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73°\text{F} - 60°\text{F}) = \mathbf{1196\ Btu/s}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1196 \text{ Btu/s}}{1025.2 \text{ Btu/lbm}} = 1.167 \text{ lbm/s}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\swarrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{steam}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{steam}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{steam}}(s_f - s_g) = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} - \dot{m}_{\text{steam}} s_{fg}$$

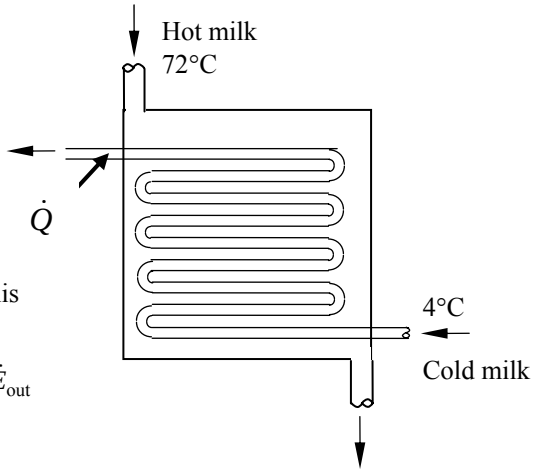
$$= (92 \text{ lbm/s})(1.0 \text{ Btu/lbm.R}) \ln\frac{73 + 460}{60 + 460} - (1.167 \text{ lbm/s})(17686 \text{ Btu/lbm.R})$$

$$= \mathbf{0.209\ Btu/s.R}$$

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7-151 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year and the annual reduction in the rate of entropy generation are to be determined.

Assumptions 1 Steady operating conditions exist. **2** The properties of the milk are constant.



Properties The average density and specific heat of milk can be taken to be $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1$ kg/L and $c_{p,\text{ milk}} = 3.79$ kJ/kg.°C (Table A-3).

Analysis The mass flow rate of the milk is

$$\dot{m}_{\text{milk}} = \rho \dot{V}_{\text{milk}} = (1\text{ kg/L})(12\text{ L/s}) = 12\text{ kg/s} = 43{,}200\text{ kg/h}$$

Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{milk}} c_p (T_2 - T_1)$$

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot{Q}_{\text{current}} = [\dot{m} c_{\text{p}} (T_{\text{pasturization}} - T_{\text{refrigeration}})]_{\text{milk}} = (12\text{ kg/s})(3.79\text{ kJ/kg.°C})(72 - 4)\text{°C} = 3093\text{ kJ/s}$$

The proposed regenerator has an effectiveness of $\varepsilon = 0.82$, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{\text{saved}} = \varepsilon \dot{Q}_{\text{current}} = (0.82)(3093\text{ kJ/s}) = 2536\text{ kJ/s}$$

Noting that the boiler has an efficiency of $\eta_{\text{boiler}} = 0.82$, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot{Q}_{\text{saved}}}{\eta_{\text{boiler}}} = \frac{(2536\text{ kJ/s})}{(0.82)} \frac{(1\text{therm})}{(105{,}500\text{ kJ})} = 0.02931\text{ therm/s}$$

Noting that 1 year = 365×24=8760 h and unit cost of natural gas is \$0.52/therm, the annual fuel and money savings will be

Fuel Saved = (0.02931 therms/s)(8760×3600 s) = **924,450 therms/yr**

Money saved = (Fuel saved)(Unit cost of fuel) = (924,450 therm/yr)(\$1.04/therm) = **\$961,400/yr**

The rate of entropy generation during this process is determined by applying the rate form of the entropy balance on an *extended system* that includes the regenerator and the immediate surroundings so that the boundary temperature is the surroundings temperature, which we take to be the cold water temperature of 18°C.:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} \rightarrow \dot{S}_{\text{gen}} = \dot{S}_{\text{out}} - \dot{S}_{\text{in}}$$

Disregarding entropy transfer associated with fuel flow, the only significant difference between the two cases is the reduction is the entropy transfer to water due to the reduction in heat transfer to water, and is determined to be

$$\dot{S}_{gen,\text{ reduction}} = \dot{S}_{out,\text{ reduction}} = \frac{\dot{Q}_{\text{out,reduction}}}{T_{\text{surr}}} = \frac{\dot{Q}_{\text{saved}}}{T_{\text{surr}}} = \frac{2536\text{ kJ/s}}{18 + 273} = 8.715\text{ kW/K}$$

$$S_{\text{gen, reduction}} = \dot{S}_{\text{gen, reduction}} \Delta t = (8.715\text{ kJ/s.K})(8760 \times 3600\text{ s/year}) = \mathbf{2.75 \times 10^8\ kJ/K}\text{ (per year)}$$

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7-152 Stainless steel ball bearings leaving the oven at a uniform temperature of 900°C at a rate of 1100 /min are exposed to air and are cooled to 850°C before they are dropped into the water for quenching. The rate of heat transfer from the ball to the air and the rate of entropy generation due to this heat transfer are to be determined.

Assumptions **1** The thermal properties of the bearing balls are constant. **2** The kinetic and potential energy changes of the balls are negligible. **3** The balls are at a uniform temperature at the end of the process

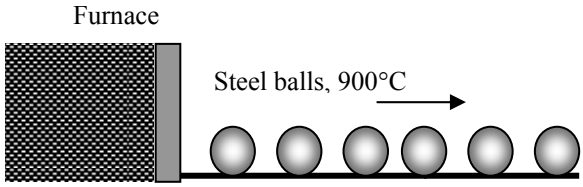
Properties The density and specific heat of the ball bearings are given to be $\rho = 8085$ kg/m^3 and $c_p = 0.480$ kJ/kg.°C.

Analysis (*a*) We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (8085 \text{ kg/m}^3)\frac{\pi (0.018 \text{ m})^3}{6} = 0.02469 \text{ kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.02469 \text{ kg})(0.480 \text{ kJ/kg.°C})(900 - 850)°\text{C} = 0.5925 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{out (per ball)}} = (1100 \text{ balls/min}) \times (0.5925 \text{ kJ/ball}) = 651.8 \text{ kJ/min} = \mathbf{10.86 \text{ kW}}$$

Therefore, heat is lost to the air at a rate of 10.86 kW.

(*b*) We again take a single bearing ball as the system. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the ball and its immediate surroundings so that the boundary temperature of the extended system is at 20°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln \frac{T_2}{T_1} = (0.02469 \text{ kg})(0.480 \text{ kJ/kg.K}) \ln \frac{850 + 273}{900 + 273} = -0.0005162 \text{ kJ/K}$$

Substituting,

$$S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}} = \frac{0.5925 \text{ kJ}}{293 \text{ K}} - 0.0005162 \text{ kJ/K} = 0.001506 \text{ kJ/K} \quad \text{(per ball)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = S_{\text{gen}} \dot{n}_{\text{ball}} = (0.001506 \text{ kJ/K} \cdot \text{ball})(1100 \text{ balls/min}) = 1.657 \text{ kJ/min.K} = \mathbf{0.02761 \text{ kW/K}}$$

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7-153 An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked and the amount of entropy generation associated with this heat transfer process are to be determined.

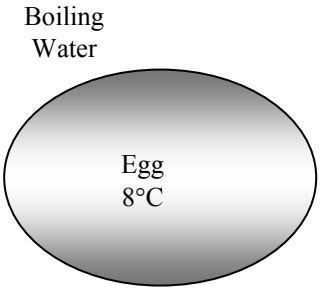
Assumptions **1** The egg is spherical in shape with a radius of $r_0 = 2.75$ cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies.

Properties The density and specific heat of the egg are given to be $\rho = 1020$ kg/m^3 and $c_p = 3.32$ kJ/kg.°C.

Analysis We take the egg as the system. This is a closes system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{egg}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020 \text{ kg/m}^3)\frac{\pi (0.055 \text{ m})^3}{6} = 0.0889 \text{ kg}$$

$$Q_{\text{in}} = mc_p(T_2 - T_1) = (0.0889 \text{ kg})(3.32 \text{ kJ/kg.°C})(70 - 8)°\text{C} = \mathbf{18.3\ kJ}$$

We again take a single egg as the system The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the egg and its immediate surroundings so that the boundary temperature of the extended system is at 97°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \rightarrow S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln \frac{T_2}{T_1} = (0.0889 \text{ kg})(3.32 \text{ kJ/kg.K}) \ln \frac{70 + 273}{8 + 273} = 0.0588 \text{ kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{18.3 \text{ kJ}}{370 \text{ K}} + 0.0588 \text{ kJ/K} = \mathbf{0.00934\ kJ/K} \quad \text{(per egg)}$$

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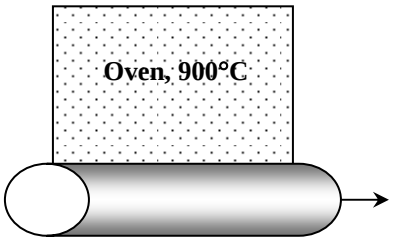
7-154 Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven and the rate of entropy generation associated with this heat transfer process are to be determined.

Assumptions **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the steel rods are given to be $\rho = 7833$ kg/m^3 and $c_p = 0.465$ kJ/kg.°C.

Analysis (*a*) Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho V = \rho LA = \rho L(\pi D^2/4)$$
$$= (7833\,\text{kg/m}^3)(3\,\text{m})[\pi(0.1\,\text{m})^2/4] = 184.6\,\text{kg}$$



We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (184.6\,\text{kg})(0.465\,\text{kJ/kg.°C})(700 - 30)°\text{C} = 57{,}512\,\text{kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}}/\Delta t = (57{,}512\,\text{kJ})/(1\,\text{min}) = 57{,}512\,\text{kJ/min} = \mathbf{958.5\,kW}$$

(*b*) We again take the 3-m long section of the rod as the system. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the rod and its immediate surroundings so that the boundary temperature of the extended system is at 900°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (184.6\,\text{kg})(0.465\,\text{kJ/kg.K})\ln\frac{700 + 273}{30 + 273} = 100.1\,\text{kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{57{,}512\,\text{kJ}}{900 + 273\,\text{K}} + 100.1\,\text{kJ/K} = 51.1\,\text{kJ/K}$$

Noting that this much entropy is generated in 1 min, the rate of entropy generation becomes

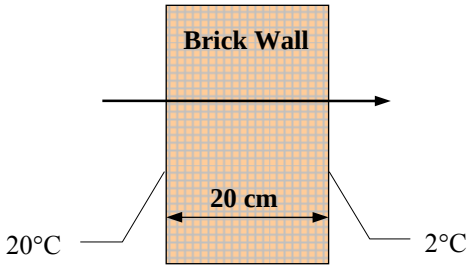
$$\dot{S}_{\text{gen}} = \frac{S_{\text{gen}}}{\Delta t} = \frac{51.1\,\text{kJ/K}}{1\,\text{min}} = 51.1\,\text{kJ/min.K} = \mathbf{0.85\ kW/K}$$

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7-155 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of entropy generation within the wall is to be determined.

Assumptions Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values.

Analysis We take the wall to be the system, which is a closed system. Under steady conditions, the rate form of the entropy balance for the wall simplifies to



$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}{}^{\nearrow 0} = 0$$

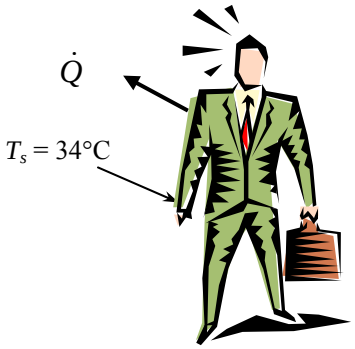
$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,wall}} = 0$$

$$\frac{1550\ \text{W}}{293\ \text{K}} - \frac{1550\ \text{W}}{275\ \text{K}} + \dot{S}_{\text{gen,wall}} = 0$$

$$\dot{S}_{\text{gen,wall}} = \mathbf{0.346\ W/K}$$

Therefore, the rate of entropy generation in the wall is 0.346 W/K.

7-156 A person is standing in a room at a specified temperature. The rate of entropy transfer from the body with heat is to be determined.



Assumptions Steady operating conditions exist.

Analysis Noting that Q/T represents entropy transfer with heat, the rate of entropy transfer from the body of the person accompanying heat transfer is

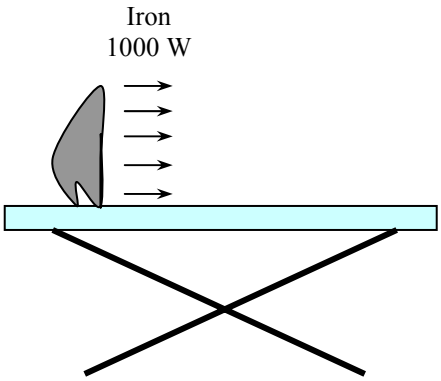
$$\dot{S}_{\text{transfer}} = \frac{\dot{Q}}{T} = \frac{336\ \text{W}}{307\ \text{K}} = \mathbf{1.094\ W/K}$$

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7-157 A 1000-W iron is left on the iron board with its base exposed to the air at 20°C. The rate of entropy generation is to be determined in steady operation.

Assumptions Steady operating conditions exist.

Analysis We take the iron to be the system, which is a closed system. Considering that the iron experiences no change in its properties in steady operation, including its entropy, the rate form of the entropy balance for the iron simplifies to



$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}{}^{\nearrow 0} = 0$$

$$-\frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,iron}} = 0$$

Therefore,

$$\dot{S}_{\text{gen,iron}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} = \frac{1000\ \text{W}}{673\ \text{K}} = \mathbf{1.486\ W/K}$$

The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the iron and its immediate surroundings so that the boundary temperature of the extended system is at 20°C at all times. It gives

$$\dot{S}_{\text{gen,total}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} = \frac{\dot{Q}}{T_{\text{surr}}} = \frac{1000\ \text{W}}{293\ \text{K}} = \mathbf{3.413\ W/K}$$

Discussion Note that only about one-third of the entropy generation occurs within the iron. The rest occurs in the air surrounding the iron as the temperature drops from 400°C to 20°C without serving any useful purpose.

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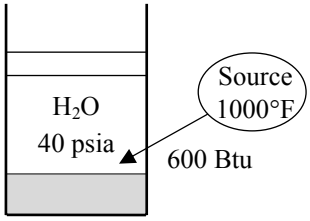
7-158E A cylinder contains saturated liquid water at a specified pressure. Heat is transferred to liquid from a source and some liquid evaporates. The total entropy generation during this process is to be determined.

Assumptions **1** No heat loss occurs from the water to the surroundings during the process. **2** The pressure inside the cylinder and thus the water temperature remains constant during the process. **3** No irreversibilities occur within the cylinder during the process.

Analysis The pressure of the steam is maintained constant. Therefore, the temperature of the steam remains constant also at

$$T = T_{\text{sat@40 psia}} = 267.2°\text{F} = 727.2\ \text{R} \qquad \text{(Table A-5E)}$$

Taking the contents of the cylinder as the system and noting that the temperature of water remains constant, the entropy change of the system during this isothermal, internally reversible process becomes



$$\Delta S_{\text{system}} = \frac{Q_{\text{sys,in}}}{T_{\text{sys}}} = \frac{600\ \text{Btu}}{727.2\ \text{R}} = 0.8251\ \text{Btu/R}$$

Similarly, the entropy change of the heat source is determined from

$$\Delta S_{\text{source}} = -\frac{Q_{\text{source,out}}}{T_{\text{source}}} = -\frac{600\ \text{Btu}}{1000 + 460\ \text{R}} = -0.4110\ \text{Btu/R}$$

Now consider a combined system that includes the cylinder and the source. Noting that no heat or mass crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen,total}} = \Delta S_{\text{water}} + \Delta S_{\text{source}}$$

Therefore, the total entropy generated during this process is

$$S_{\text{gen,total}} = \Delta S_{\text{water}} + \Delta S_{\text{source}} = 0.8251 - 0.4110 = \mathbf{0.414\ Btu/R}$$

Discussion The entropy generation in this case is entirely due to the irreversible heat transfer through a finite temperature difference. We could also determine the total entropy generation by writing an energy balance on an extended system that includes the system and its immediate surroundings so that part of the boundary of the extended system, where heat transfer occurs, is at the source temperature.

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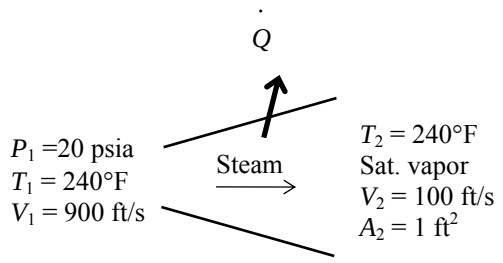
7-159E Steam is decelerated in a diffuser from a velocity of 900 ft/s to 100 ft/s. The mass flow rate of steam and the rate of entropy generation are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

Properties The properties of steam at the inlet and the exit of the diffuser are (Tables A-4E through A-6E)

$$\left.\begin{matrix} P_1 = 20 \text{ psia} \\ T_1 = 240°\text{F} \end{matrix}\right\} \begin{matrix} h_1 = 1162.3 \text{ Btu/lbm} \\ s_1 = 1.7406 \text{ Btu/lbm} \cdot \text{R} \end{matrix}$$

$$\left.\begin{matrix} T_2 = 240°\text{F} \\ \text{sat. vapor} \end{matrix}\right\} \begin{matrix} h_2 = 1160.5 \text{ Btu/lbm} \\ s_2 = 1.7141 \text{ Btu/lbm} \cdot \text{R} \\ \nu_2 = 16.316 \text{ ft}^3\text{/lbm} \end{matrix}$$



Analysis (a) The mass flow rate of the steam can be determined from its definition to be

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 = \frac{1}{16.316 \text{ ft}^3\text{/lbm}} (1 \text{ ft}^2)(100 \text{ ft/s}) = \mathbf{6.129 \text{ lbm/s}}$$

(*b*) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) - \dot{Q}_{\text{out}} = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

Substituting, the rate of heat loss from the diffuser is determined to be

$$\dot{Q}_{\text{out}} = -(6.129 \text{ lbm/s})\left(1160.5 - 1162.3 + \frac{(100 \text{ ft/s})^2 - (900 \text{ ft/s})^2}{2} \left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2} \right) \right) = 108.42 \text{ Btu/s}$$

The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the diffuser and its immediate surroundings so that the boundary temperature of the extended system is 77°F at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} \overset{\nearrow 0}{} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the total rate of entropy generation during this process becomes

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} = (6.129 \text{ lbm/s})(1.7141 - 1.7406)\text{Btu/lbm} \cdot \text{R} + \frac{108.42 \text{ Btu/s}}{537 \text{ R}} = \mathbf{0.0395 \ Btu/s \cdot R}$$

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7-160 Steam is accelerated in a nozzle from a velocity of 55 m/s to 390 m/s. The exit temperature and the rate of entropy generation are to be determined.

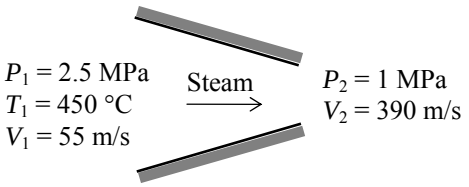
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-6),

$$\left.\begin{matrix} P_1 = 2.5 \text{ MPa} \\ T_1 = 450\,^\circ\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3351.6 \text{ kJ/kg} \\ s_1 = 7.1768 \text{ kJ/kg} \cdot \text{K} \\ \nu_1 = 0.13015 \text{ m}^3\text{/kg} \end{matrix}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta \text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,,

$$\text{or,} \qquad h_2 = 3351.6 \text{ kJ/kg} - \frac{(390 \text{ m/s})^2 - (55 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3277.0 \text{ kJ/kg}$$

Thus,

$$\left.\begin{matrix} P_2 = 1 \text{ MPa} \\ h_{2a} = 3277.0 \text{ kJ/kg} \end{matrix}\right\} \begin{matrix} T_2 = \mathbf{405.9\,^\circ C} \\ s_2 = 7.4855 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$

The mass flow rate of steam is

$$\dot{m} = \frac{1}{\nu_1} A_1 V_1 = \frac{1}{0.13015 \text{ m}^3\text{/kg}}\left(6 \times 10^{-4} \text{ m}^2\right)(55 \text{ m/s}) = 0.2535 \text{ kg/s}$$

(*b*) Again we take the nozzle to be the system. Noting that no heat crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}^{\not\nearrow 0} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

Substituting, the rate of entropy generation during this process is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) = (0.2535 \text{ kg/s})(7.4855 - 7.1768)\text{kJ/kg} \cdot \text{K} = \mathbf{0.07829 \text{ kW/K}}$$

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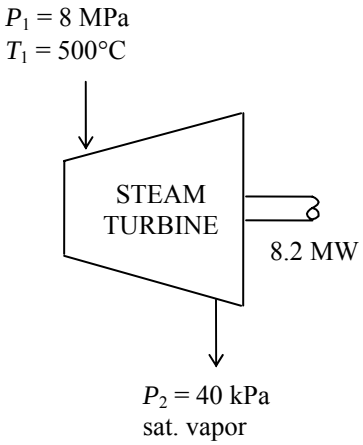
7-161 Steam expands in a turbine from a specified state to another specified state. The rate of entropy generation during this process is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 8\ \text{MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3399.5\ \text{kJ/kg} \\ s_1 = 6.7266\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 40\ \text{kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_2 = 2636.1\ \text{kJ/kg} \\ s_2 = 7.6691\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \dot{m}h_2$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2) - \dot{W}_{\text{out}}$$

Substituting,

$$\dot{Q}_{\text{out}} = (40{,}000/3600\ \text{kg/s})(3399.5 - 2636.1)\text{kJ/kg} - 8200\ \text{kJ/s} = 282.6\ \text{kJ/s}$$

The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the turbine and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the rate of entropy generation during this process is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} = (40{,}000/3600\ \text{kg/s})(7.6691 - 6.7266)\text{kJ/kg}\cdot\text{K} + \frac{282.6\ \text{kW}}{298\ \text{K}} = \mathbf{11.4\ kW/K}$$

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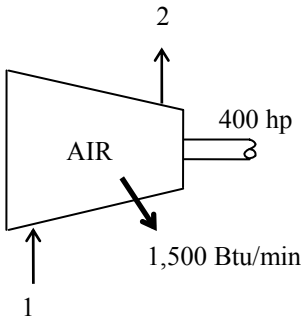
7-162E Air is compressed steadily by a compressor. The mass flow rate of air through the compressor and the rate of entropy generation are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is 0.06855 Btu/lbm.R (Table A-1E). The inlet and exit enthalpies of air are (Table A-17E)

$$\left.\begin{array}{l} T_1 = 520\ \text{R} \\ P_1 = 15\ \text{psia} \end{array}\right\} \begin{array}{l} h_1 = 124.27\ \text{Btu/lbm} \\ s_1^\circ = 0.59173\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} T_2 = 1080\ \text{R} \\ P_2 = 150\ \text{psia} \end{array}\right\} \begin{array}{l} h_2 = 260.97\ \text{Btu/lbm} \\ s_2^\circ = 0.76964\ \text{Btu/lbm}\cdot\text{R} \end{array}$$



Analysis (*a*) We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate is determined to be

$$\text{Thus,} \quad (400\ \text{hp})\left(\frac{0.7068\ \text{Btu/s}}{1\ \text{hp}}\right) - \frac{1500\ \text{Btu}}{60\ \text{s}} = \dot{m}\left(260.97 - 124.27 + \frac{(350\ \text{ft/s})^2}{2}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right)\right)$$

It yields $\dot{m} = \mathbf{1.852\ lbm/s}$

(*b*) Again we take the compressor to be the system. Noting that no heat or mass crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}}$$

where

$$\Delta \dot{S}_{\text{air}} = \dot{m}(s_2 - s_1) = \dot{m}\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$

$$= (1.852\ \text{lbm/s})\left(0.76964 - 0.59173 - (0.06855\ \text{Btu/lbm}\cdot\text{R})\ln\frac{150\ \text{psia}}{15\ \text{psia}}\right) = 0.0372\ \text{Btu/s}\cdot\text{R}$$

Substituting, the rate of entropy generation during this process is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} = 0.0372\ \text{Btu/s.R} + \frac{1500/60\ \text{Btu/s}}{520\ \text{R}} = \mathbf{0.0853\ Btu/s.R}$$

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7-163 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water stream and the rate of entropy generation are to be determined.

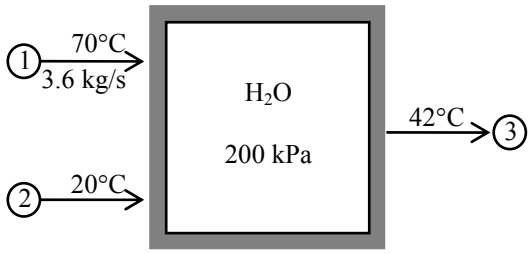
Assumptions **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.21°\text{C}$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus from Table A-4,

$$\left.\begin{matrix} P_1 = 200\ \text{kPa} \\ T_1 = 70°\text{C} \end{matrix}\right\} \begin{matrix} h_1 \cong h_{f@70°\text{C}} = 293.07\ \text{kJ/kg} \\ s_1 \cong s_{f@70°\text{C}} = 0.9551\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 200\ \text{kPa} \\ T_2 = 20°\text{C} \end{matrix}\right\} \begin{matrix} h_2 \cong h_{f@20°\text{C}} = 83.91\ \text{kJ/kg} \\ s_2 \cong s_{f@20°\text{C}} = 0.2965\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_3 = 200\ \text{kPa} \\ T_3 = 42°\text{C} \end{matrix}\right\} \begin{matrix} h_3 \cong h_{f@42°\text{C}} = 175.90\ \text{kJ/kg} \\ s_3 \cong s_{f@42°\text{C}} = 0.5990\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$



Analysis (*a*) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two relations gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2}\dot{m}_1 = \frac{(293.07 - 175.90)\text{kJ/kg}}{(175.90 - 83.91)\text{kJ/kg}}(3.6\ \text{kg/s}) = \mathbf{4.586\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 3.6 + 4.586 = 8.186\ \text{kg/s}$

(*b*) Noting that the mixing chamber is adiabatic and thus there is no heat transfer to the surroundings, the entropy balance of the steady-flow system (the mixing chamber) can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\not\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} = 0$$

Substituting, the total rate of entropy generation during this process becomes

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 \\ &= (8.186\ \text{kg/s})(0.5990\ \text{kJ/kg}\cdot\text{K}) - (4.586\ \text{kg/s})(0.2965\ \text{kJ/kg}\cdot\text{K}) - (3.6\ \text{kg/s})(0.9551\ \text{kJ/kg}\cdot\text{K}) \\ &= \mathbf{0.1054\ kW/K} \end{aligned}$$

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7-164 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of entropy generation are to be determined.

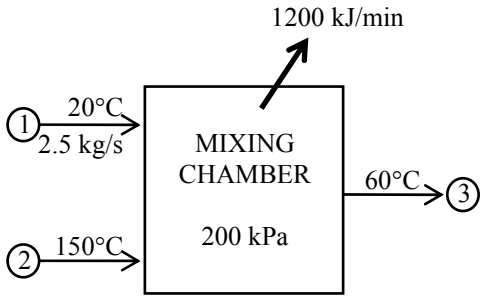
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.21°\text{C}$, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From Tables A-4 through A-6,

$$\left.\begin{matrix} P_1 = 200\ \text{kPa} \\ T_1 = 20°\text{C} \end{matrix}\right\} \begin{matrix} h_1 \cong h_{f@20°\text{C}} = 83.91\ \text{kJ/kg} \\ s_1 \cong s_{f@20°\text{C}} = 0.2965\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 200\ \text{kPa} \\ T_2 = 150°\text{C} \end{matrix}\right\} \begin{matrix} h_2 = 2769.1\ \text{kJ/kg} \\ s_2 = 7.2810\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_3 = 200\ \text{kPa} \\ T_3 = 60°\text{C} \end{matrix}\right\} \begin{matrix} h_3 \cong h_{f@60°\text{C}} = 251.18\ \text{kJ/kg} \\ s_3 \cong s_{f@60°\text{C}} = 0.8313\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$



Analysis (a) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\not\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{Q}_{\text{out}} + \dot{m}_3 h_3$$

Combining the two relations gives $\dot{Q}_{\text{out}} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3 = \dot{m}_1 (h_1 - h_3) + \dot{m}_2 (h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1 (h_1 - h_3)}{h_2 - h_3} = \frac{(1200/60\text{kJ/s}) - (2.5\ \text{kg/s})(83.91 - 251.18)\text{kJ/kg}}{(2769.1 - 251.18)\text{kJ/kg}} = \mathbf{0.166\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 2.5 + 0.166 = 2.666\ \text{kg/s}$

(*b*) The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\not\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the rate of entropy generation during this process is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} \\ &= (2.666\ \text{kg/s})(0.8313\ \text{kJ/kg}\cdot\text{K}) - (0.166\ \text{kg/s})(7.2810\ \text{kJ/kg}\cdot\text{K}) \\ &\quad - (2.5\ \text{kg/s})(0.2965\ \text{kJ/kg}\cdot\text{K}) + \frac{(1200/60\ \text{kJ/s})}{298\ \text{K}} \\ &= \mathbf{0.333\ kW/K} \end{aligned}$$

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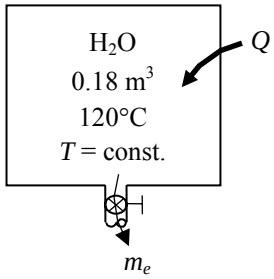
7-165 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer and the entropy generation during this process are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 120°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@120°\text{C}} = 0.001060\ \text{m}^3/\text{kg} \\ u_1 = u_{f@120°\text{C}} = 503.60\ \text{kJ/kg} \\ s_1 = s_{f@120°\text{C}} = 1.5279\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_e = 120°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_e = h_{f@120°\text{C}} = 503.81\ \text{kJ/kg} \\ s_e = s_{f@120°\text{C}} = 1.5279\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{0.18\ \text{m}^3}{0.001060\ \text{m}^3/\text{kg}} = 169.76\ \text{kg}$$

$$m_2 = \frac{1}{2} m_1 = \frac{1}{2}(169.76\ \text{kg}) = 84.88\ \text{kg} = m_e$$

Now we determine the final internal energy and entropy,

$$v_2 = \frac{V}{m_2} = \frac{0.18\ \text{m}^3}{84.88\ \text{kg}} = 0.002121\ \text{m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002121 - 0.001060}{0.8913 - 0.001060} = 0.001191$$

$$\left.\begin{array}{l} T_2 = 120°\text{C} \\ x_2 = 0.001191 \end{array}\right\} \begin{array}{l} u_2 = u_f + x_2 u_{fg} = 503.60 + (0.001191)(2025.3) = 506.01\ \text{kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 1.5279 + (0.001191)(5.6013) = 1.5346\ \text{kJ/kg}\cdot\text{K} \end{array}$$

The heat transfer during this process is determined by substituting these values into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (84.88\ \text{kg})(503.81\ \text{kJ/kg}) + (84.88\ \text{kg})(506.01\ \text{kJ/kg}) - (169.76\ \text{kg})(503.60\ \text{kJ/kg}) \\ &= \mathbf{222.6\ kJ} \end{aligned}$$

(*b*) The total entropy generation is determined by considering a combined system that includes the tank and the heat source. Noting that no heat crosses the boundaries of this combined system and no mass enters, the entropy balance for it can be expressed as

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$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}} \longrightarrow -m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} + \Delta S_{\text{source}}$$

Therefore, the total entropy generated during this process is

$$\begin{aligned} S_{\text{gen}} &= m_e s_e + \Delta S_{\text{tank}} + \Delta S_{\text{source}} = m_e s_e + (m_2 s_2 - m_1 s_1) - \frac{Q_{\text{source,out}}}{T_{\text{source}}} \\ &= (84.88\ \text{kg})(1.5279\ \text{kJ/kg}\cdot\text{K}) + (84.88\ \text{kg})(1.5346\ \text{kJ/kg}\cdot\text{K}) \\ &\quad - (169.76\ \text{kg})(1.5279\ \text{kJ/kg}\cdot\text{K}) - \frac{222.6\ \text{kJ}}{(230+273)\ \text{K}} \\ &= \mathbf{0.1237\ kJ/K} \end{aligned}$$

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7-166E An unknown mass of iron is dropped into water in an insulated tank while being stirred by a 200-W paddle wheel. Thermal equilibrium is established after 10 min. The mass of the iron block and the entropy generated during this process are to be determined.

Assumptions **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energy changes are zero. **3** The system is well-insulated and thus there is no heat transfer.

Properties The specific heats of water and the iron block at room temperature are $c_{p,\,\text{water}} = 1.00$ Btu/lbm.°F and $c_{p,\,\text{iron}} = 0.107$ Btu/lbm.°F (Table A-3E). The density of water at room temperature is 62.1 lbm/ft³.

Analysis (*a*) We take the entire contents of the tank, water + iron block, as the system. This is a closed system since no mass crosses the system boundary during the process. The energy balance on the system can be expressed as

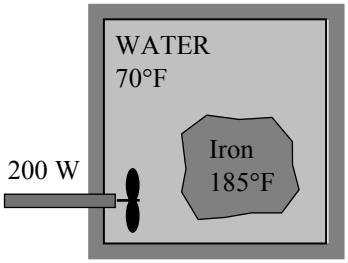
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} = \Delta U$$

or,

$$W_{\text{pw,in}} = \Delta U_{\text{iron}} + \Delta U_{\text{water}}$$

$$W_{\text{pw,in}} = [mc(T_2 - T_1)]_{\text{iron}} + [mc(T_2 - T_1)]_{\text{water}}$$



where

$$m_{\text{water}} = \rho V = (62.1\ \text{lbm/ft}^3)(0.8\ \text{ft}^3) = 49.7\ \text{lbm}$$

$$W_{\text{pw}} = \dot{W}_{\text{pw}}\Delta t = (0.2\ \text{kJ/s})(10 \times 60\ \text{s})\left(\frac{1\ \text{Btu}}{1.055\ \text{kJ}}\right) = 113.7\ \text{Btu}$$

Using specific heat values for iron and liquid water and substituting,

$$113.7\ \text{Btu} = m_{\text{iron}}(0.107\ \text{Btu/lbm}\cdot{}^\circ\text{F})(75 - 185)^\circ\text{F} + (49.7\ \text{lbm})(1.00\ \text{Btu/lbm}\cdot{}^\circ\text{F})(75 - 70)^\circ\text{F}$$

$$m_{\text{iron}} = \mathbf{11.4\ lbm}$$

(*b*) Again we take the iron + water in the tank to be the system. Noting that no heat or mass crosses the boundaries of this combined system, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen,total}} = \Delta S_{\text{iron}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (11.4\ \text{lbm})(0.107\ \text{Btu/lbm}\cdot\text{R})\ln\left(\frac{535\ \text{R}}{645\ \text{R}}\right) = -0.228\ \text{Btu/R}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (49.6\ \text{lbm})(1.0\ \text{Btu/lbm}\cdot\text{R})\ln\left(\frac{535\ \text{R}}{530\ \text{R}}\right) = 0.466\ \text{Btu/R}$$

Therefore, the entropy generated during this process is

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{iron}} + \Delta S_{\text{water}} = -0.228 + 0.466 = \mathbf{0.238\ Btu/R}$$

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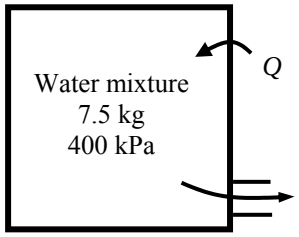
7-167 Liquid water is withdrawn from a rigid tank that initially contains saturated water mixture until no liquid is left in the tank. The quality of steam in the tank at the initial state, the amount of mass that has escaped, and the entropy generation during this process are to be determined.

Assumptions **1** Kinetic and potential energy changes are zero. **2** There are no work interactions.

Analysis (*a*) The properties of the steam in the tank at the final state and the properties of exiting steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_2 = 400\ \text{kPa} \\ x_2 = 1\ (\text{sat. vap.}) \end{array}\right\} \begin{array}{l} u_2 = 2553.1\ \text{kJ/kg} \\ \nu_2 = 0.46242\ \text{m}^3/\text{kg} \\ s_2 = 6.8955\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_{2e} = 400\ \text{kPa} \\ x_e = 0\ (\text{sat. liq.}) \end{array}\right\} \begin{array}{l} h_e = 604.66\ \text{kJ/kg} \\ s_e = 1.7765\ \text{kJ/kg.K} \end{array}$$



The relations for the volume of the tank and the final mass in the tank are

$$V = m_1\nu_1 = (7.5\ \text{kg})\nu_1$$

$$m_2 = \frac{V}{\nu_2} = \frac{(7.5\ \text{kg})\nu_1}{0.46242\ \text{m}^3/\text{kg}} = 16.219\nu_1$$

The mass, energy, and entropy balances may be written as

$$m_e = m_1 - m_2$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$\frac{Q_{\text{in}}}{T_{\text{source}}} - m_e s_e + S_{gen} = m_2 s_2 - m_1 s_1$$

Substituting,

$$m_e = 7.5 - 16.219\nu_1 \qquad (1)$$

$$5 - (7.5 - 16.219\nu_1)(604.66) = 16.219\nu_1(2553.1) - 7.5u_1 \qquad (2)$$

$$\frac{5}{500+273} - (7.5 - 16.219\nu_1)(1.7765) + S_{\text{gen}} = 16.219\nu_1(6.8955) - 7.5s_1 \qquad (3)$$

Eq. (2) may be solved by a trial-error approach by trying different qualities at the inlet state. Or, we can use EES to solve the equations to find

$$x_1 = \mathbf{0.8666}$$

Other properties at the initial state are

$$\left.\begin{array}{l} P_1 = 400\ \text{kPa} \\ x_1 = 0.8666 \end{array}\right\} \begin{array}{l} u_1 = 2293.2\ \text{kJ/kg} \\ \nu_1 = 0.40089\ \text{m}^3/\text{kg} \\ s_1 = 6.2129\ \text{kJ/kg.K} \end{array}$$

Substituting into Eqs (1) and (3),

(*b*) $$m_e = 7.5 - 16.219(0.40089) = \mathbf{0.998\ kg}$$

(*c*)

$$\frac{5}{500+273} - \left[7.5 - 16.219(0.40089)\right](1.7765) + S_{\text{gen}} = 16.219(0.40089)(6.8955) - 7.5(6.2129)$$

$$S_{\text{gen}} = \mathbf{0.00553\ kJ/K}$$

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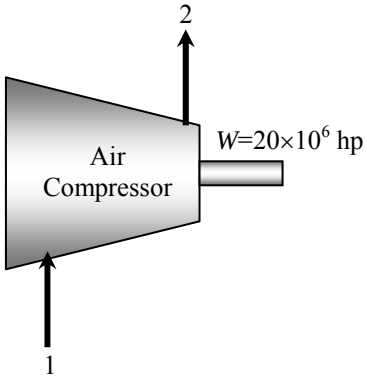
Special Topic: Reducing the Cost of Compressed Air

7-168 The total installed power of compressed air systems in the US is estimated to be about 20 million horsepower. The amount of energy and money that will be saved per year if the energy consumed by compressors is reduced by 5 percent is to be determined.

Assumptions **1** The compressors operate at full load during one-third of the time on average, and are shut down the rest of the time. **2** The average motor efficiency is 85 percent.

Analysis The electrical energy consumed by compressors per year is

$$\begin{aligned}\text{Energy consumed} &= (\text{Power rating})(\text{Load factor})(\text{Annual Operating Hours})/\text{Motor efficiency}\\ &= (20\times10^6\ \text{hp})(0.746\ \text{kW/hp})(1/3)(365\times24\ \text{hours/year})/0.85\\ &= 5.125\times10^{10}\ \text{kWh/year}\end{aligned}$$



Then the energy and cost savings corresponding to a 5% reduction in energy use for compressed air become

$$\begin{aligned}\text{Energy Savings} &= (\text{Energy consumed})(\text{Fraction saved})\\ &= (5.125\times10^{10}\ \text{kWh})(0.05)\\ &= \mathbf{2.563\times10^9\ kWh/year}\\ \text{Cost Savings} &= (\text{Energy savings})(\text{Unit cost of energy})\\ &= (2.563\times10^9\ \text{kWh/year})(\$0.07/\text{kWh})\\ &= \mathbf{\$0.179\times10^9\ /year}\end{aligned}$$

Therefore, reducing the energy usage of compressors by 5% will save $179 million a year.

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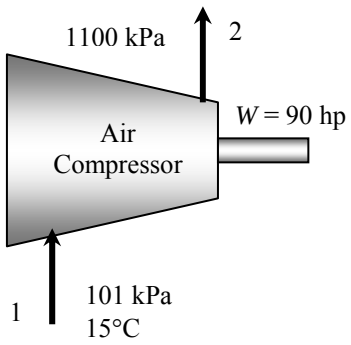
7-169 The compressed air requirements of a plant is being met by a 90 hp compressor that compresses air from 101.3 kPa to 1100 kPa. The amount of energy and money saved by reducing the pressure setting of compressed air to 750 kPa is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible. **3** The load factor of the compressor is given to be 0.75. **4** The pressures given are absolute pressure rather than gage pressure.

Properties The specific heat ratio of air is $k = 1.4$ (Table A-2).

Analysis The electrical energy consumed by this compressor per year is

$$\begin{aligned}\text{Energy consumed} &= (\text{Power rating})(\text{Load factor})(\text{Annual Operating Hours})/\text{Motor efficiency}\\ &= (90\text{ hp})(0.746\text{ kW/hp})(0.75)(3500\text{ hours/year})/0.94\\ &= 187{,}420\text{ kWh/year}\end{aligned}$$



The fraction of energy saved as a result of reducing the pressure setting of the compressor is

$$\begin{aligned}\text{Power Reduction Factor} &= 1-\frac{(P_{2,\text{reduced}}/P_1)^{(k-1)/k}-1}{(P_2/P_1)^{(k-1)/k}-1}\\ &= 1-\frac{(750/101.3)^{(1.4-1)/1,4}-1}{(1100/101.3)^{(1.4-1)/1,4}-1}\\ &= 0.2098\end{aligned}$$

That is, reducing the pressure setting will result in about 11 percent savings from the energy consumed by the compressor and the associated cost. Therefore, the energy and cost savings in this case become

$$\begin{aligned}\text{Energy Savings} &= (\text{Energy consumed})(\text{Power reduction factor})\\ &= (187{,}420\text{ kWh/year})(0.2098)\\ &= \mathbf{39{,}320\text{ kWh/year}}\\ \text{Cost Savings} &= (\text{Energy savings})(\text{Unit cost of energy})\\ &= (39{,}320\text{ kWh/year})(\$0.085\text{/kWh})\\ &= \mathbf{\$3342/year}\end{aligned}$$

Therefore, reducing the pressure setting by 250 kPa will result in annual savings of 39,320 kWh that is worth $3342 in this case.

Discussion Some applications require very low pressure compressed air. In such cases the need can be met by a blower instead of a compressor. Considerable energy can be saved in this manner, since a blower requires a small fraction of the power needed by a compressor for a specified mass flow rate.

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7-170 A 150 hp compressor in an industrial facility is housed inside the production area where the average temperature during operating hours is 25°C. The amounts of energy and money saved as a result of drawing cooler outside air to the compressor instead of using the inside air are to be determined.

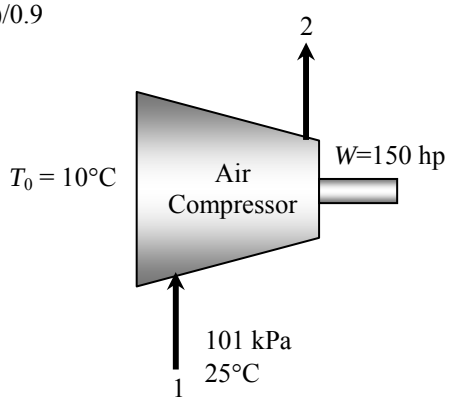
Assumptions **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible.

Analysis The electrical energy consumed by this compressor per year is

Energy consumed = (Power rating)(Load factor)(Annual Operating Hours)/Motor efficiency

= (150 hp)(0.746 kW/hp)(0.85)(4500 hours/year)/0.9

= 475,384 kWh/year



Also,

Cost of Energy = (Energy consumed)(Unit cost of energy)

= (475,384 kWh/year)($0.07/kWh)

= $33,277/year

The fraction of energy saved as a result of drawing in cooler outside air is

$$\text{Power Reduction Factor} = 1 - \frac{T_{\text{outside}}}{T_{\text{inside}}} = 1 - \frac{10+273}{25+273} = 0.0503$$

That is, drawing in air which is 15°C cooler will result in 5.03 percent savings from the energy consumed by the compressor and the associated cost. Therefore, the energy and cost savings in this case become

Energy Savings = (Energy consumed)(Power reduction factor)

= (475,384 kWh/year)(0.0503)

= **23,929 kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

= (23,929 kWh/year)($0.07/kWh)

= $1675/year

Therefore, drawing air in from the outside will result in annual savings of 23,929 kWh, which is worth $1675 in this case.

Discussion The price of a typical 150 hp compressor is much lower than $50,000. Therefore, it is interesting to note that the cost of energy a compressor uses a year may be more than the cost of the compressor itself.

The implementation of this measure requires the installation of an ordinary sheet metal or PVC duct from the compressor intake to the outside. The installation cost associated with this measure is relatively low, and the pressure drop in the duct in most cases is negligible. About half of the manufacturing facilities we have visited, especially the newer ones, have the duct from the compressor intake to the outside in place, and they are already taking advantage of the savings associated with this measure.

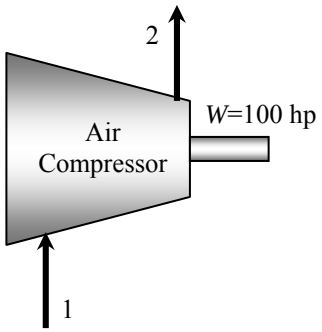
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7-171 The compressed air requirements of the facility during 60 percent of the time can be met by a 25 hp reciprocating compressor instead of the existing 100 hp compressor. The amounts of energy and money saved as a result of switching to the 25 hp compressor during 60 percent of the time are to be determined.

Analysis Noting that 1 hp = 0.746 kW, the electrical energy consumed by each compressor per year is determined from

$$(\text{Energy consumed})_{\text{Large}} = (\text{Power})(\text{Hours})[(\text{LFxTF}/\eta_{\text{motor}})_{\text{Unloaded}} + (\text{LFxTF}/\eta_{\text{motor}})_{\text{Loaded}}]$$
$$= (100\text{ hp})(0.746\text{ kW/hp})(3800\text{ hours/year})[0.35\times0.6/0.82+0.90\times0.4/0.9]$$
$$= 185{,}990\text{ kWh/year}$$

$$(\text{Energy consumed})_{\text{Small}} = (\text{Power})(\text{Hours})[(\text{LFxTF}/\eta_{\text{motor}})_{\text{Unloaded}} + (\text{LFxTF}/\eta_{\text{motor}})_{\text{Loaded}}]$$
$$= (25\text{ hp})(0.746\text{ kW/hp})(3800\text{ hours/year})[0.0\times0.15+0.95\times0.85]/0.88$$
$$= 65{,}031\text{ kWh/year}$$



Therefore, the energy and cost savings in this case become

$$\text{Energy Savings} = (\text{Energy consumed})_{\text{Large}} - (\text{Energy consumed})_{\text{Small}}$$
$$= 185{,}990 - 65{,}031\text{ kWh/year}$$
$$= \mathbf{120{,}959\text{ kWh/year}}$$

$$\text{Cost Savings} = (\text{Energy savings})(\text{Unit cost of energy})$$
$$= (120{,}959\text{ kWh/year})(\$0.075/\text{kWh})$$
$$= \mathbf{\$9{,}072/year}$$

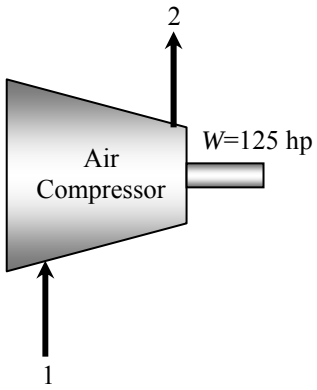
Discussion Note that utilizing a small compressor during the times of reduced compressed air requirements and shutting down the large compressor will result in annual savings of 120,959 kWh, which is worth $9,072 in this case.

7-172 A facility stops production for one hour every day, including weekends, for lunch break, but the 125 hp compressor is kept operating. If the compressor consumes 35 percent of the rated power when idling, the amounts of energy and money saved per year as a result of turning the compressor off during lunch break are to be determined.

Analysis It seems like the compressor in this facility is kept on unnecessarily for one hour a day and thus 365 hours a year, and the idle factor is 0.35. Then the energy and cost savings associated with turning the compressor off during lunch break are determined to be

$$\text{Energy Savings} = (\text{Power Rating})(\text{Turned Off Hours})(\text{Idle Factor})/\eta_{\text{motor}}$$
$$= (125\text{ hp})(0.746\text{ kW/hp})(365\text{ hours/year})(0.35)/0.84$$
$$= \mathbf{14{,}182\text{ kWh/year}}$$

$$\text{Cost Savings} = (\text{Energy savings})(\text{Unit cost of energy})$$
$$= (14{,}182\text{ kWh/year})(\$0.09/\text{kWh})$$
$$= \mathbf{\$1{,}276/year}$$



Discussion Note that the simple practice of turning the compressor off during lunch break will save this facility $1,276 a year in energy costs. There are also side benefits such as extending the life of the motor and the compressor, and reducing the maintenance costs.

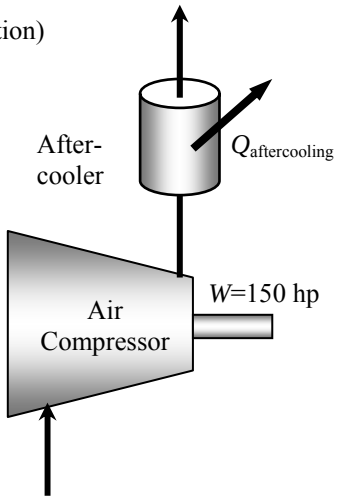
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7-173 It is determined that 25 percent of the energy input to the compressor is removed from the compressed air as heat in the aftercooler with a refrigeration unit whose COP is 2.5. The amounts of the energy and money saved per year as a result of cooling the compressed air before it enters the refrigerated dryer are to be determined.

Assumptions The compressor operates at full load when operating.

Analysis Noting that 25 percent of the energy input to the compressor is removed by the aftercooler, the rate of heat removal from the compressed air in the aftercooler under full load conditions is

$$\dot{Q}_{\text{aftercooling}} = (\text{Rated Power of Compressor})(\text{Load Factor})(\text{Aftercooling Fraction})$$
$$= (150\ \text{hp})(0.746\ \text{kW/hp})(1.0)(0.25) = 27.96\ \text{kW}$$



The compressor is said to operate at full load for 2100 hours a year, and the COP of the refrigeration unit is 2.5. Then the energy and cost savings associated with this measure become

$$\text{Energy Savings} = (\dot{Q}_{\text{aftercooling}})(\text{Annual Operating Hours})/\text{COP}$$
$$= (27.96\ \text{kW})(2100\ \text{hours/year})/2.5$$
$$= \mathbf{23{,}490\ kWh/year}$$
$$\text{Cost Savings} = (\text{Energy savings})(\text{Unit cost of energy saved})$$
$$= (23{,}490\ \text{kWh/year})(\$0.095/\text{kWh})$$
$$= \mathbf{\$2232/year}$$

Discussion Note that the aftercooler will save this facility 23,490 kWh of electrical energy worth $2232 per year. The actual savings will be less than indicated above since we have not considered the power consumed by the fans and/or pumps of the aftercooler. However, if the heat removed by the aftercooler is utilized for some useful purpose such as space heating or process heating, then the actual savings will be much more.

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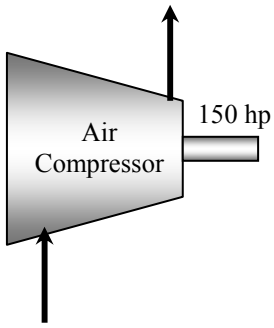
7-174 The motor of a 150 hp compressor is burned out and is to be replaced by either a 93% efficient standard motor or a 96.2% efficient high efficiency motor. The amount of energy and money the facility will save by purchasing the high-efficiency motor instead of standard motor are to be determined. It is also to be determined if the savings from the high efficiency motor justify the price differential.

Assumptions **1** The compressor operates at full load when operating. **2** The life of the motors is 10 years. **3** There are no rebates involved. **4** The price of electricity remains constant.

Analysis The energy and cost savings associated with the installation of the high efficiency motor in this case are determined to be

$$\text{Energy Savings} = (\text{Power Rating})(\text{Operating Hours})(\text{Load Factor})(1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}})$$
$$= (150\text{ hp})(0.746\text{ kW/hp})(4{,}368\text{ hours/year})(1.0)(1/0.930 - 1/0.962)$$
$$= \mathbf{17{,}483\text{ kWh/year}}$$

$$\text{Cost Savings} = (\text{Energy savings})(\text{Unit cost of energy})$$
$$= (17{,}483\text{ kWh/year})(\$0.075/\text{kWh})$$
$$= \mathbf{\$1311/year}$$



The additional cost of the energy efficient motor is

$$\text{Cost Differential} = \$10{,}942 - \$9{,}031 = \$1{,}911$$

Discussion The money saved by the high efficiency motor will pay for this cost difference in \$1,911/\$1311 = 1.5 years, and will continue saving the facility money for the rest of the 10 years of its lifetime. Therefore, the use of the high efficiency motor is recommended in this case even in the absence of any incentives from the local utility company.

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7-175 The compressor of a facility is being cooled by air in a heat-exchanger. This air is to be used to heat the facility in winter. The amount of money that will be saved by diverting the compressor waste heat into the facility during the heating season is to be determined.

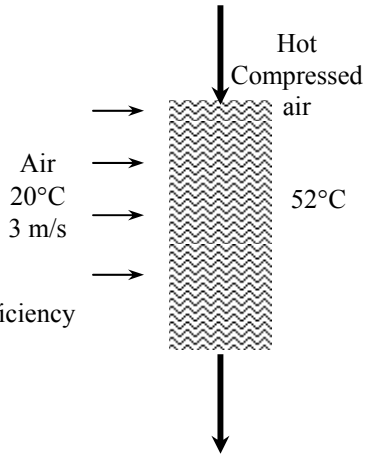
Assumptions The compressor operates at full load when operating.

Analysis Assuming c_p = 1.0 kJ/kg.°C and operation at sea level and taking the density of air to be 1.2 kg/m^3, the mass flow rate of air through the liquid-to-air heat exchanger is determined to be

$$\begin{aligned}\text{Mass flow rate of air} &= (\text{Density of air})(\text{Average velocity})(\text{Flow area})\\ &= (1.2\ \text{kg/m}^3)(3\ \text{m/s})(1.0\ \text{m}^2)\\ &= 3.6\ \text{kg/s} = 12{,}960\ \text{kg/h}\end{aligned}$$

Noting that the temperature rise of air is 32°C, the rate at which heat can be recovered (or the rate at which heat is transferred to air) is

$$\begin{aligned}\text{Rate of Heat Recovery} &= (\text{Mass flow rate of air})(\text{Specific heat of air})(\text{Temperature rise})\\ &= (12{,}960\ \text{kg/h})(1.0\ \text{kJ/kg.°C})(32°\text{C})\\ &= 414{,}720\ \text{kJ/h}\end{aligned}$$



The number of operating hours of this compressor during the heating season is

$$\begin{aligned}\text{Operating hours} &= (20\ \text{hours/day})(5\ \text{days/week})(26\ \text{weeks/year})\\ &= 2600\ \text{hours/year}\end{aligned}$$

Then the annual energy and cost savings become

$$\begin{aligned}\text{Energy Savings} &= (\text{Rate of Heat Recovery})(\text{Annual Operating Hours})/\text{Efficiency}\\ &= (414{,}720\ \text{kJ/h})(2600\ \text{hours/year})/0.8\\ &= 1{,}347{,}840{,}000\ \text{kJ/year}\\ &= 12{,}776\ \text{therms/year}\end{aligned}$$

$$\begin{aligned}\text{Cost Savings} &= (\text{Energy savings})(\text{Unit cost of energy saved})\\ &= (12{,}776\ \text{therms/year})(\$1.0/\text{therm})\\ &= \mathbf{\$12{,}776/year}\end{aligned}$$

Therefore, utilizing the waste heat from the compressor will save $12,776 per year from the heating costs.

Discussion The implementation of this measure requires the installation of an ordinary sheet metal duct from the outlet of the heat exchanger into the building. The installation cost associated with this measure is relatively low. A few of the manufacturing facilities we have visited already have this conservation system in place. A damper is used to direct the air into the building in winter and to the ambient in summer.

Combined compressor/heat-recovery systems are available in the market for both air-cooled (greater than 50 hp) and water cooled (greater than 125 hp) systems.

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7-176 The compressed air lines in a facility are maintained at a gage pressure of 700 kPa at a location where the atmospheric pressure is 85.6 kPa. There is a 3-mm diameter hole on the compressed air line. The energy and money saved per year by sealing the hole on the compressed air line.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K. The specific heat ratio of air is $k = 1.4$ (Table A-2).

Analysis Disregarding any pressure losses and noting that the absolute pressure is the sum of the gage pressure and the atmospheric pressure, the work needed to compress a unit mass of air at 15°C from the atmospheric pressure of 85.6 kPa to 700+85.6 = 785.6 kPa is determined to be

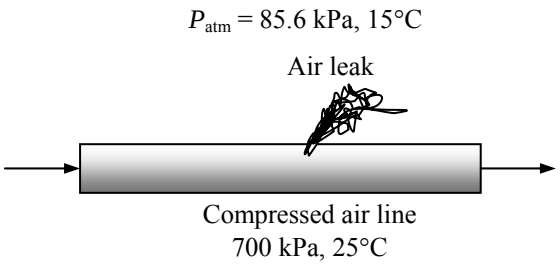
$$w_{\text{comp, in}} = \frac{kRT_1}{\eta_{\text{comp}}(k-1)}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]$$

$$= \frac{(1.4)(0.287\ \text{kJ/kg.K})(288\ \text{K})}{(0.8)(1.4-1)}\left[\left(\frac{785.6\ \text{kPa}}{85.6\ \text{kPa}}\right)^{(1.4-1)/1.4} - 1\right]$$

$$= 319.6\ \text{kJ/kg}$$

The cross-sectional area of the 5-mm diameter hole is

$$A = \pi D^2/4 = \pi(3\times10^{-3}\ \text{m})^2/4 = 7.069\times10^{-6}\ \text{m}^2$$



Noting that the line conditions are $T_0 = 298$ K and $P_0 = 785.6$ kPa, the mass flow rate of the air leaking through the hole is determined to be

$$\dot{m}_{\text{air}} = C_{\text{loss}}\left(\frac{2}{k+1}\right)^{1/(k-1)}\frac{P_0}{RT_0}A\sqrt{kR\left(\frac{2}{k+1}\right)T_0}$$

$$= (0.65)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)}\frac{785.6\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(298\ \text{K})}(7.069\times10^{-6}\ \text{m}^2)$$

$$\times\sqrt{(1.4)(0.287\ \text{kJ/kg.K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)\left(\frac{2}{1.4+1}\right)(298\ \text{K})}$$

$$= 0.008451\ \text{kg/s}$$

Then the power wasted by the leaking compressed air becomes

$$\text{Power wasted} = \dot{m}_{\text{air}}w_{\text{comp,in}} = (0.008451\ \text{kg/s})(319.6\ \text{kJ/kg}) = 2.701\ \text{kW}$$

Noting that the compressor operates 4200 hours a year and the motor efficiency is 0.93, the annual energy and cost savings resulting from repairing this leak are determined to be

Energy Savings = (Power wasted)(Annual operating hours)/Motor efficiency

= (2.701 kW)(4200 hours/year)/0.93

= 12,200 kWh/year

Cost Savings = (Energy savings)(Unit cost of energy)

= (12,200 kWh/year)($0.07/kWh)

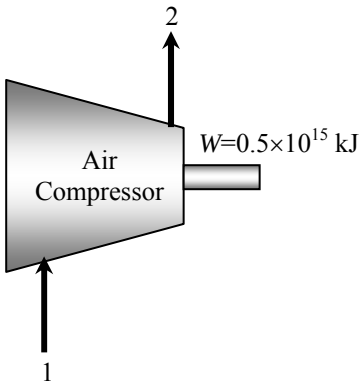
= $854/year

Therefore, the facility will save 12,200 kWh of electricity that is worth $854 a year when this air leak is sealed.

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7-177 The total energy used to compress air in the US is estimated to be 0.5×10^{15} kJ per year. About 20% of the compressed air is estimated to be lost by air leaks. The amount and cost of electricity wasted per year due to air leaks is to be determined.

Assumptions About 20% of the compressed air is lost by air leaks.



Analysis The electrical energy and money wasted by air leaks are

Energy wasted = (Energy consumed)(Fraction wasted)

$= (0.5\times10^{15}\text{ kJ})(1\text{ kWh}/3600\text{ kJ})(0.20)$

$= \mathbf{27.78\times10^9\ kWh/year}$

Money wasted = (Energy wasted)(Unit cost of energy)

$= (27.78\times10^9\text{ kWh/year})(\$0.07/\text{kWh})$

$= \mathbf{\$1.945\times10^9\ /year}$

Therefore, air leaks are costing almost $2 billion a year in electricity costs. The environment also suffers from this because of the pollution associated with the generation of this much electricity.

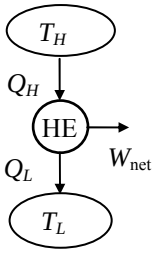
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Review Problems

7-178E The source and sink temperatures and the thermal efficiency of a heat engine are given. The entropy change of the two reservoirs is to be calculated and it is to be determined if this engine satisfies the increase of entropy principle.

Assumptions The heat engine operates steadily.

Analysis According to the first law and the definition of the thermal efficiency,



$$Q_L = (1-\eta)Q_H = (1-0.4)(1\,\text{Btu}) = 0.6\,\text{Btu}$$

when the thermal efficiency is 40%. The entropy change of everything involved in this process is then

$$\Delta S_{\text{total}} = \Delta S_H + \Delta S_L$$
$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{-1\,\text{Btu}}{1300\,\text{R}} + \frac{0.6\,\text{Btu}}{500\,\text{R}} = \mathbf{0.000431\,Btu/R}$$

Since the entropy of everything has increased, this engine is possible. When the thermal efficiency of the engine is 70%,

$$Q_L = (1-\eta)Q_H = (1-0.7)(1\,\text{Btu}) = 0.3\,\text{Btu}$$

The total entropy change is then

$$\Delta S_{\text{total}} = \Delta S_H + \Delta S_L$$
$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{-1\,\text{Btu}}{1300\,\text{R}} + \frac{0.3\,\text{Btu}}{500\,\text{R}} = \mathbf{-0.000169\,Btu/R}$$

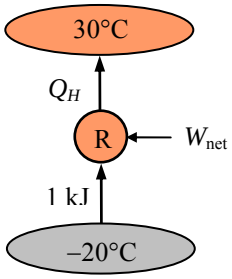
which is a decrease in the entropy of everything involved with this engine. Therefore, this engine is now impossible.

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7-179 The source and sink temperatures and the COP of a refrigerator are given. The total entropy change of the two reservoirs is to be calculated and it is to be determined if this refrigerator satisfies the second law.

Assumptions The refrigerator operates steadily.

Analysis Combining the first law and the definition of the coefficient of performance produces



$$Q_H = Q_L\left(1+\frac{1}{\text{COP}_\text{R}}\right) = (1\,\text{kJ})\left(1+\frac{1}{4}\right) = 1.25\,\text{kJ}$$

when COP = 4. The entropy change of everything is then

$$\Delta S_{\text{total}} = \Delta S_H + \Delta S_L$$
$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{1.25\,\text{kJ}}{303\,\text{K}} + \frac{-1\,\text{kJ}}{253\,\text{K}} = \mathbf{0.000173\ kJ/K}$$

Since the entropy increases, a refrigerator with COP = 4 is possible. When the coefficient of performance is increased to 6,

$$Q_H = Q_L\left(1+\frac{1}{\text{COP}_\text{R}}\right) = (1\,\text{kJ})\left(1+\frac{1}{6}\right) = 1.167\,\text{kJ}$$

and the net entropy change is

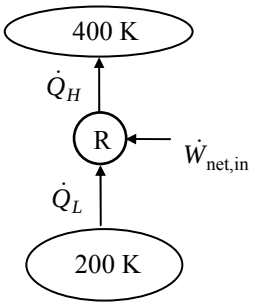
$$\Delta S_{\text{total}} = \Delta S_H + \Delta S_L$$
$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L} = \frac{1.167\,\text{kJ}}{303\,\text{K}} + \frac{-1\,\text{kJ}}{253\,\text{K}} = \mathbf{-0.000101\,kJ/K}$$

and the refrigerator can no longer be possible.

7-180 The operating conditions of a refrigerator are given. The rate of entropy changes of all components and the rate of cooling are to be calculated and it is to be determined if this refrigerator is reversible.

Assumptions The refrigerator operates steadily.

Analysis Applying the first law to the refrigerator below, the rate of cooling is



$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 14\,\text{kW} - 10\,\text{kW} = \mathbf{4\ kW}$$

The rate of entropy change for the low-temperature reservoir, according to the definition of the entropy, is

$$\Delta\dot{S}_L = \frac{\dot{Q}_L}{T_L} = \frac{-4\,\text{kW}}{200\,\text{K}} = -0.02\,\text{kW/K}$$

The rate at which the entropy of the high-temperature energy reservoir is changing is

$$\Delta\dot{S}_H = \frac{\dot{Q}_H}{T_H} = \frac{14\,\text{kW}}{400\,\text{K}} = 0.035\,\text{kW/K}$$

Since the working fluid inside the refrigerator is constantly returning to its original state, the entropy of the device does not change. Summing the rates at which the entropy of everything involved with this device changes, produces

$$\Delta\dot{S}_{\text{total}} = \Delta\dot{S}_H + \Delta\dot{S}_L + \Delta\dot{S}_{\text{device}} = 0.035 - 0.020 + 0 = \mathbf{0.015\,kW/K}$$

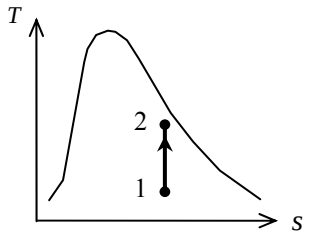
Hence, the increase in entropy principle is satisfied, and this refrigerator is possible, but not completely reversible.

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7-181 R-134a is compressed in a compressor adiabatically. The minimum internal energy at the final state is to be determined.

Analysis The initial state is saturated mixture and the properties are (Table A-12)

$$\begin{aligned} s_1 &= s_{f\,@\,200\,\text{kPa}} + x s_{fg\,@\,200\,\text{kPa}} \\ &= 0.15457 + 0.85(0.78316) \\ &= 0.8203\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$



For minimum internal energy at the final state, the process should be isentropic. Then,

$$\left.\begin{aligned} P_2 &= 800\,\text{kPa} \\ s_2 &= s_1 = 0.8203\,\text{kJ/kg}\cdot\text{K} \end{aligned}\right\} \begin{aligned} x_2 &= \frac{s_2 - s_f}{s_{fg}} = \frac{0.8203 - 0.35404}{0.56431} = 0.8262 \\ u_2 &= u_f + x_2 u_{fg} = 94.79 + (0.8262)(152.00) = \mathbf{220.4\ kJ/kg} \end{aligned}$$

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7-182 R-134a is condensed in a piston-cylinder device in an isobaric and reversible process. It is to be determined if the process described is possible.

Analysis We take the R-134a as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

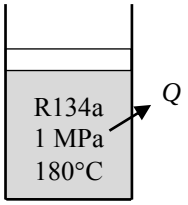
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$-Q_{\text{out}} = W_{\text{b,out}} + m(u_2 - u_1)$$

$$-Q_{\text{out}} = m(h_2 - h_1)$$

$$Q_{\text{out}} = m(h_1 - h_2)$$

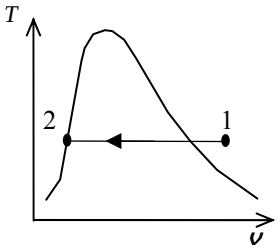


since $\Delta U + W_{\text{b}} = \Delta H$ during a constant pressure quasi-equilibrium process.

The initial and final state properties are (Tables A-12 and A-13)

$$\left.\begin{array}{l} P_1 = 1000\ \text{kPa} \\ T_1 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 421.36\ \text{kJ/kg} \\ s_1 = 1.3124\ \text{kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1000\ \text{kPa} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} h_2 = 107.32\ \text{kJ/kg} \\ s_2 = 0.3919\ \text{kJ/kg} \cdot \text{K} \end{array}$$



Substituting,

$$q_{\text{out}} = h_1 - h_2 = 421.36 - 107.32 = 314.0\ \text{kJ/kg}$$

The entropy change of the energy reservoir as it undergoes a reversible, isothermal process is

$$\Delta s_{\text{surr}} = \frac{q_{\text{out}}}{T_{\text{res}}} = \frac{314.0\ \text{kJ/kg}}{(100 + 273)\ \text{K}} = 0.8419\ \text{kJ/kg} \cdot \text{K}$$

where the sign of heat transfer is taken positive as the reservoir receives heat. The entropy change of R-134a during the process is

$$\Delta s_{\text{R134a}} = s_2 - s_1 = 0.3919 - 1.3124 = -0.9205\ \text{kJ/kg} \cdot \text{K}$$

The total entropy change is then,

$$\Delta s_{\text{total}} = \Delta s_{\text{R134a}} + \Delta s_{\text{surr}} = -0.9205 + 0.8419 = \mathbf{-0.0786\ kJ/kg \cdot K}$$

Since the total entropy change (i.e., entropy generation) is negative, this process is impossible.

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7-183 Air is first compressed adiabatically and then expanded adiabatically to the initial pressure. It is to be determined if the air can be cooled by this process.

Analysis From the entropy change relation of an ideal gas,

$$\Delta s_{\text{air}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

Since the initial and final pressures are the same, the equation reduces to

$$\Delta s_{\text{air}} = c_p \ln\frac{T_2}{T_1}$$

As there are no heat transfer, the total entropy change (i.e., entropy generation) for this process is equal to the entropy change of air. Therefore, we must have

$$\Delta s_{\text{air}} = c_p \ln\frac{T_2}{T_1} \ge 0$$

The only way this result can be satisfied is if

$$T_2 \ge T_1$$

It is therefore impossible to create a cooling effect ($T_2 < T_1$) in the manner proposed.

7-184E Air is compressed adiabatically in a closed system. It is to be determined if this process is possible.

Assumptions **1** Changes in the kinetic and potential energies are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $R = 0.3704$ psia·ft^3/lbm·R, $c_p = 0.240$ Btu/lbm·R, $k = 1.4$ (Table A-2Ea).

Analysis The specific volume of air at the initial state is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(560\,\text{R})}{20\,\text{psia}} = 10.37\,\text{ft}^3/\text{lbm}$$

The volume at the final state will be minimum if the process is isentropic. The specific volume for this case is determined from the isentropic relation of an ideal gas to be

$$v_{2,\min} = v_1\left(\frac{P_1}{P_2}\right)^{1/k} = (10.37\,\text{ft}^3/\text{lbm})\left(\frac{20\,\text{psia}}{120\,\text{psia}}\right)^{1/1.4} = 2.884\,\text{ft}^3/\text{lbm}$$

and the minimum volume is

$$V_2 = mv_2 = (1\,\text{lbm})(2.884\,\text{ft}^3/\text{lbm}) = \mathbf{2.88\ ft^3}$$

which is smaller than the proposed volume 3 ft^3/lbm. Hence, it is possible to compress this air into 3 ft^3/lbm.

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7-185E A solid block is heated with saturated water vapor. The final temperature of the block and water, and the entropy changes of the block, water, and the entire system are to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved. **3** There is no heat transfer between the system and the surroundings.

Analysis (a) As the block is heated, some of the water vapor will be condensed. We will assume (will be checked later) that the water is a mixture of liquid and vapor at the end of the process. Based upon this assumption, the final temperature of the water and solid block is **212°F** (The saturation temperature at 14.7 psia). The heat picked up by the block is

$$Q_{\text{block}} = mc(T_2 - T_1) = (100\ \text{lbm})(0.5\ \text{Btu/lbm}\cdot\text{R})(212 - 70)\text{R} = 7100\ \text{Btu}$$

The water properties at the initial state are

$$\left.\begin{array}{l} P_1 = 14.7\ \text{psia} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} T_1 = 212°\text{F} \\ h_1 = 1150.3\ \text{Btu/lbm} \\ s_1 = 1.7566\ \text{Btu/lbm}\cdot\text{R} \end{array} \qquad \text{(Table A-5E)}$$

The heat released by the water is equal to the heat picked up by the block. Also noting that the pressure of water remains constant, the enthalpy of water at the end of the heat exchange process is determined from

$$h_2 = h_1 - \frac{Q_{\text{water}}}{m_w} = 1150.3\ \text{Btu/lbm} - \frac{7100\ \text{Btu}}{10\ \text{lbm}} = 440.3\ \text{Btu/lbm}$$

The state of water at the final state is saturated mixture. Thus, our initial assumption was correct. The properties of water at the final state are

$$\left.\begin{array}{l} P_2 = 14.7\ \text{psia} \\ h_2 = 440.3\ \text{Btu/lbm} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{h_2 - h_f}{h_{fg}} = \dfrac{440.3 - 180.16}{970.12} = 0.2682 \\ s_2 = s_f + x_2 s_{fg} = 0.31215 + (0.2682)(1.44441) = 0.69947\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

The entropy change of the water is then

$$\Delta S_{\text{water}} = m_w(s_2 - s_1) = (10\ \text{lbm})(0.69947 - 1.7566)\text{Btu/lbm} = \mathbf{-10.57\ Btu/R}$$

(b) The entropy change of the block is

$$\Delta S_{\text{block}} = mc \ln \frac{T_2}{T_1} = (100\ \text{lbm})(0.5\ \text{Btu/lbm}\cdot\text{R}) \ln \frac{(212 + 460)\text{R}}{(70 + 460)\text{R}} = \mathbf{11.87\ Btu/R}$$

(c) The total entropy change is

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{block}} = -10.57 + 11.87 = \mathbf{1.30\ Btu/R}$$

The positive result for the total entropy change (i.e., entropy generation) indicates that this process is possible.

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7-186 Air is compressed in a piston-cylinder device. It is to be determined if this process is possible.

Assumptions **1** Changes in the kinetic and potential energies are negligible. **4** Air is an ideal gas with constant specific heats. **3** The compression process is reversible.

Properties The properties of air at room temperature are $R = 0.287$ kPa·m^3/kg·K, $c_p = 1.005$ kJ/kg·K (Table A-2a).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

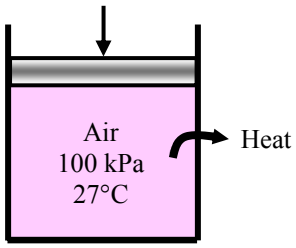
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$W_{b,\text{in}} - Q_{\text{out}} = mc_p(T_2 - T_1)$$

$$W_{b,\text{in}} - Q_{\text{out}} = 0 \quad (\text{since } T_2 = T_1)$$

$$Q_{\text{out}} = W_{b,\text{in}}$$



The work input for this isothermal, reversible process is

$$w_{\text{in}} = RT \ln \frac{P_2}{P_1} = (0.287 \text{ kJ/kg} \cdot \text{K})(300 \text{ K}) \ln \frac{250 \text{ kPa}}{100 \text{ kPa}} = 78.89 \text{ kJ/kg}$$

That is,

$$q_{\text{out}} = w_{\text{in}} = 78.89 \text{ kJ/kg}$$

The entropy change of air during this isothermal process is

$$\Delta s_{\text{air}} = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} = -R \ln \frac{P_2}{P_1} = -(0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{250 \text{ kPa}}{100 \text{ kPa}} = -0.2630 \text{ kJ/kg} \cdot \text{K}$$

The entropy change of the reservoir is

$$\Delta s_{\text{R}} = \frac{q_{\text{R}}}{T_R} = \frac{78.89 \text{ kJ/kg}}{300 \text{ K}} = 0.2630 \text{ kJ/kg} \cdot \text{K}$$

Note that the sign of heat transfer is taken with respect to the reservoir. The total entropy change (i.e., entropy generation) is the sum of the entropy changes of air and the reservoir:

$$\Delta s_{\text{total}} = \Delta s_{\text{air}} + \Delta s_{\text{R}} = -0.2630 + 0.2630 = \mathbf{0 \text{ kJ/kg} \cdot \text{K}}$$

Not only this process is possible but also completely reversible.

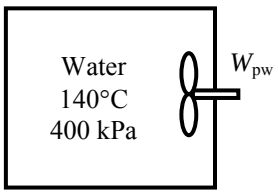
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7-187 A paddle wheel does work on the water contained in a rigid tank. For a zero entropy change of water, the final pressure in the tank, the amount of heat transfer between the tank and the surroundings, and the entropy generation during the process are to be determined.

Assumptions The tank is stationary and the kinetic and potential energy changes are negligible.

Analysis (*a*) Using saturated liquid properties for the compressed liquid at the initial state (Table A-4)

$$\left.\begin{array}{l} T_1 = 140°\text{C} \\ x_1 = 0\ (\text{sat. liq.}) \end{array}\right\} \begin{array}{l} u_1 = 588.76\ \text{kJ/kg} \\ s_1 = 1.7392\ \text{kJ/kg.K} \end{array}$$



The entropy change of water is zero, and thus at the final state we have

$$\left.\begin{array}{l} T_2 = 80°\text{C} \\ s_2 = s_1 = 1.7392\ \text{kJ/kg.K} \end{array}\right\} \begin{array}{l} P_2 = \mathbf{47.4\ kPa} \\ u_2 = 552.93\ \text{kJ/kg} \end{array}$$

(*b*) The heat transfer can be determined from an energy balance on the tank

$$Q_{\text{out}} = W_{\text{Pw,in}} - m(u_2 - u_1) = 48\ \text{kJ} - (3.2\ \text{kg})(552.93 - 588.76)\text{kJ/kg} = \mathbf{163\ kJ}$$

(*c*) Since the entropy change of water is zero, the entropy generation is only due to the entropy increase of the surroundings, which is determined from

$$S_{\text{gen}} = \Delta S_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{163\ \text{kJ}}{(15 + 273)\ \text{K}} = \mathbf{0.565\ kJ/K}$$

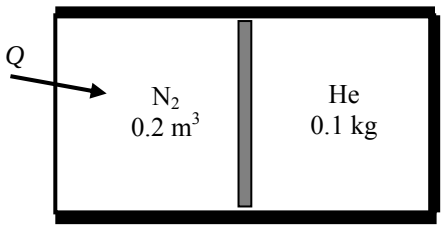
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7-188 A horizontal cylinder is separated into two compartments by a piston, one side containing nitrogen and the other side containing helium. Heat is added to the nitrogen side. The final temperature of the helium, the final volume of the nitrogen, the heat transferred to the nitrogen, and the entropy generation during this process are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Nitrogen and helium are ideal gases with constant specific heats at room temperature. **3** The piston is adiabatic and frictionless.

Properties The properties of nitrogen at room temperature are $R = 0.2968$ kPa.m^3/kg.K, $c_p = 1.039$ kJ/kg.K, $c_v = 0.743$ kJ/kg.K, $k = 1.4$. The properties for helium are $R = 2.0769$ kPa.m^3/kg.K, $c_p = 5.1926$ kJ/kg.K, $c_v = 3.1156$ kJ/kg.K, $k = 1.667$ (Table A-2).

Analysis (*a*) Helium undergoes an isentropic compression process, and thus the final helium temperature is determined from



$$T_{\text{He},2} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (20+273)\text{K}\left(\frac{120\text{ kPa}}{95\text{ kPa}}\right)^{(1.667-1)/1.667}$$
$$= \mathbf{321.7\ K}$$

(*b*) The initial and final volumes of the helium are

$$V_{\text{He},1} = \frac{mRT_1}{P_1} = \frac{(0.1\text{ kg})(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})}{95\text{ kPa}} = 0.6406\text{ m}^3$$

$$V_{\text{He},2} = \frac{mRT_2}{P_2} = \frac{(0.1\text{ kg})(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(321.7\text{ K})}{120\text{ kPa}} = 0.5568\text{ m}^3$$

Then, the final volume of nitrogen becomes

$$V_{\text{N2},2} = V_{\text{N2},1} + V_{\text{He},1} - V_{\text{He},2} = 0.2 + 0.6406 - 0.5568 = \mathbf{0.2838\ m^3}$$

(*c*) The mass and final temperature of nitrogen are

$$m_{\text{N2}} = \frac{P_1V_1}{RT_1} = \frac{(95\text{ kPa})(0.2\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})} = 0.2185\text{ kg}$$

$$T_{\text{N2},2} = \frac{P_2V_2}{mR} = \frac{(120\text{ kPa})(0.2838\text{ m}^3)}{(0.2185\text{ kg})(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 525.1\text{ K}$$

The heat transferred to the nitrogen is determined from an energy balance

$$Q_{\text{in}} = \Delta U_{\text{N2}} + \Delta U_{\text{He}}$$
$$= [mc_v(T_2 - T_1)]_{\text{N2}} + [mc_v(T_2 - T_1)]_{\text{He}}$$
$$= (0.2185\text{ kg})(0.743\text{ kJ/kg.K})(525.1-293) + (0.1\text{ kg})(3.1156\text{ kJ/kg.K})(321.7-293)$$
$$= \mathbf{46.6\ kJ}$$

(*d*) Noting that helium undergoes an isentropic process, the entropy generation is determined to be

$$S_{\text{gen}} = \Delta S_{\text{N2}} + \Delta S_{\text{surr}} = m_{\text{N2}}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + \frac{-Q_{\text{in}}}{T_{\text{R}}}$$
$$= (0.2185\text{ kg})\left[(1.039\text{ kJ/kg.K})\ln\frac{525.1\text{ K}}{293\text{ K}} - (0.2968\text{ kJ/kg.K})\ln\frac{120\text{ kPa}}{95\text{ kPa}}\right] + \frac{-46.6\text{ kJ}}{(500+273)\text{ K}}$$
$$= \mathbf{0.057\ kJ/K}$$

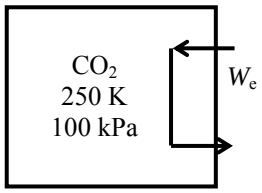
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7-189 An electric resistance heater is doing work on carbon dioxide contained an a rigid tank. The final temperature in the tank, the amount of heat transfer, and the entropy generation are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Carbon dioxide is ideal gas with constant specific heats at room temperature.

Properties The properties of CO_2 at an anticipated average temperature of 350 K are $R = 0.1889$ kPa.m^3/kg.K, $c_p = 0.895$ kJ/kg.K, $c_v = 0.706$ kJ/kg.K (Table A-2b).

Analysis (*a*) The mass and the final temperature of CO_2 may be determined from ideal gas equation



$$m = \frac{P_1 V}{RT_1} = \frac{(100\ \text{kPa})(0.8\ \text{m}^3)}{(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(250\ \text{K})} = 1.694\ \text{kg}$$

$$T_2 = \frac{P_2 V}{mR} = \frac{(175\ \text{kPa})(0.8\ \text{m}^3)}{(1.694\ \text{kg})(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{437.5\ K}$$

(*b*) The amount of heat transfer may be determined from an energy balance on the system

$$Q_{\text{out}} = \dot{E}_{\text{e,in}}\Delta t - mc_v(T_2 - T_1)$$
$$= (0.5\ \text{kW})(40 \times 60\ \text{s}) - (1.694\ \text{kg})(0.706\ \text{kJ/kg.K})(437.5 - 250)\text{K} = \mathbf{975.8\ kJ}$$

(*c*) The entropy generation associated with this process may be obtained by calculating total entropy change, which is the sum of the entropy changes of CO_2 and the surroundings

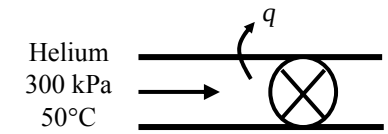
$$S_{\text{gen}} = \Delta S_{\text{CO2}} + \Delta S_{\text{surr}} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$
$$= (1.694\ \text{kg})\left[(0.895\ \text{kJ/kg.K})\ln\frac{437.5\ \text{K}}{250\ \text{K}} - (0.1889\ \text{kJ/kg.K})\ln\frac{175\ \text{kPa}}{100\ \text{kPa}}\right] + \frac{975.8\ \text{kJ}}{300\ \text{K}}$$
$$= \mathbf{3.92\ kJ/K}$$

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7-190 Heat is lost from the helium as it is throttled in a throttling valve. The exit pressure and temperature of helium and the entropy generation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The properties of helium are $R = 2.0769$ kPa.m^3/kg.K, $c_p = 5.1926$ kJ/kg.K (Table A-2a).



Analysis (*a*) The final temperature of helium may be determined from an energy balance on the control volume

$$q_{\text{out}} = c_p(T_1 - T_2) \longrightarrow T_2 = T_1 - \frac{q_{\text{out}}}{c_p} = 50°\text{C} - \frac{1.75\ \text{kJ/kg}}{5.1926\ \text{kJ/kg.°C}} = 322.7\ \text{K} = \mathbf{49.7°C}$$

The final pressure may be determined from the relation for the entropy change of helium

$$\Delta s_{\text{He}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$0.25\ \text{kJ/kg.K} = (5.1926\ \text{kJ/kg.K})\ln\frac{322.7\ \text{K}}{323\ \text{K}} - (2.0769\ \text{kJ/kg.K})\ln\frac{P_2}{300\ \text{kPa}}$$

$$P_2 = \mathbf{265\ kPa}$$

(*b*) The entropy generation associated with this process may be obtained by adding the entropy change of helium as it flows in the valve and the entropy change of the surroundings

$$s_{\text{gen}} = \Delta s_{\text{He}} + \Delta s_{\text{surr}} = \Delta s_{\text{He}} + \frac{q_{\text{out}}}{T_{\text{surr}}} = 0.25\ \text{kJ/kg.K} + \frac{1.75\ \text{kJ/kg}}{(25+273)\ \text{K}} = \mathbf{0.256\ kJ/kg.K}$$

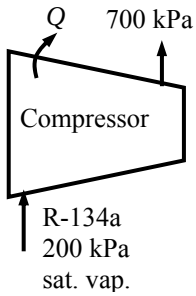
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7-191 Refrigerant-134a is compressed in a compressor. The rate of heat loss from the compressor, the exit temperature of R-134a, and the rate of entropy generation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The properties of R-134a at the inlet of the compressor are (Table A-12)

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} v_1 = 0.09987\ \text{m}^3/\text{kg} \\ h_1 = 244.46\ \text{kJ/kg} \\ s_1 = 0.93773\ \text{kJ/kg.K} \end{array}$$



The mass flow rate of the refrigerant is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.03\ \text{m}^3/\text{s}}{0.09987\ \text{m}^3/\text{kg}} = 0.3004\ \text{kg/s}$$

Given the entropy increase of the surroundings, the heat lost from the compressor is

$$\Delta \dot{S}_{\text{surr}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}} \longrightarrow \dot{Q}_{\text{out}} = T_{\text{surr}} \Delta \dot{S}_{\text{surr}} = (20 + 273\ \text{K})(0.008\ \text{kW/K}) = \mathbf{2.344\ kW}$$

(*b*) An energy balance on the compressor gives

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1)$$

$$10\ \text{kW} - 2.344\ \text{kW} = (0.3004\ \text{kg/s})(h_2 - 244.46)\ \text{kJ/kg} \longrightarrow h_2 = 269.94\ \text{kJ/kg}$$

The exit state is now fixed. Then,

$$\left.\begin{array}{l} P_2 = 700\ \text{kPa} \\ h_2 = 269.94\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{31.5°C} \\ s_2 = 0.93620\ \text{kJ/kg.K} \end{array}$$

(*c*) The entropy generation associated with this process may be obtained by adding the entropy change of R-134a as it flows in the compressor and the entropy change of the surroundings

$$\begin{aligned} \dot{S}_{\text{gen}} &= \Delta \dot{S}_{\text{R}} + \Delta \dot{S}_{\text{surr}} = \dot{m}(s_2 - s_1) + \Delta \dot{S}_{\text{surr}} \\ &= (0.3004\ \text{kg/s})(0.93620 - 0.93773)\ \text{kJ/kg.K} + 0.008\ \text{kW/K} \\ &= \mathbf{0.00754\ kJ/K} \end{aligned}$$

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7-192 Air flows in an adiabatic nozzle. The isentropic efficiency, the exit velocity, and the entropy generation are to be determined.

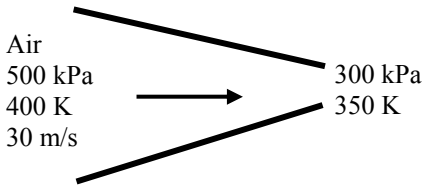
Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

Assumptions **1** Steady operating conditions exist. **2** Potential energy changes are negligible.

Analysis (*a*) (*b*) Using variable specific heats, the properties can be determined from air table as follows

$$T_1 = 400\ \text{K} \longrightarrow \begin{matrix} h_1 = 400.98\ \text{kJ/kg} \\ s_1^0 = 1.99194\ \text{kJ/kg.K} \\ P_{r1} = 3.806 \end{matrix}$$

$$T_2 = 350\ \text{K} \longrightarrow \begin{matrix} h_2 = 350.49\ \text{kJ/kg} \\ s_2^0 = 1.85708\ \text{kJ/kg.K} \end{matrix}$$



$$P_{r2} = \frac{P_2}{P_1} P_{r1} = \frac{300\ \text{kPa}}{500\ \text{kPa}}(3.806) = 2.2836 \longrightarrow h_{2s} = 346.31\ \text{kJ/kg}$$

Energy balances on the control volume for the actual and isentropic processes give

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$400.98\ \text{kJ/kg} + \frac{(30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 350.49\ \text{kJ/kg} + \frac{V_2^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$V_2 = \mathbf{319.1\ m/s}$$

$$h_1 + \frac{V_1^2}{2} = h_{2s} + \frac{V_{2s}^2}{2}$$

$$400.98\ \text{kJ/kg} + \frac{(30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 346.31\ \text{kJ/kg} + \frac{V_{2s}^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$V_{2s} = 331.8\ \text{m/s}$$

The isentropic efficiency is determined from its definition,

$$\eta_N = \frac{V_2^2}{V_{2s}^2} = \frac{(319.1\ \text{m/s})^2}{(331.8\ \text{m/s})^2} = \mathbf{0.925}$$

(*c*) Since the nozzle is adiabatic, the entropy generation is equal to the entropy increase of the air as it flows in the nozzle

$$\begin{aligned} s_{\text{gen}} = \Delta s_{\text{air}} &= s_2^0 - s_1^0 - R\ln\frac{P_2}{P_1} \\ &= (1.85708 - 1.99194)\text{kJ/kg.K} - (0.287\ \text{kJ/kg.K})\ln\frac{300\ \text{kPa}}{500\ \text{kPa}} \\ &= \mathbf{0.0118\ kJ/kg.K} \end{aligned}$$

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7-193 An insulated rigid tank is connected to a piston-cylinder device with zero clearance that is maintained at constant pressure. A valve is opened, and some steam in the tank is allowed to flow into the cylinder. The final temperatures in the tank and the cylinder are to be determined.

Assumptions **1** Both the tank and cylinder are well-insulated and thus heat transfer is negligible. **2** The water that remains in the tank underwent a reversible adiabatic process. **3** The thermal energy stored in the tank and cylinder themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible.

Analysis (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 350 \text{ kPa} \\ \textit{sat.vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@350\text{ kPa}} = 0.52422 \text{ m}^3/\text{kg} \\ u_1 = u_{g@350\text{ kPa}} = 2548.5 \text{ kJ/kg} \\ s_1 = s_{g@350\text{ kPa}} = 6.9402 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_{2,A} = T_{sat@200\text{ kPa}} = \mathbf{120.2^\circ C}$$

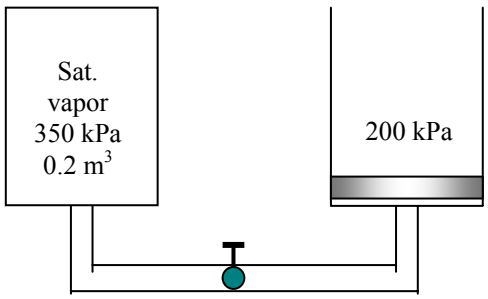
$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ s_2 = s_1 \\ (\textit{sat.mixture}) \end{array}\right\} \begin{array}{l} x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{6.9402 - 1.5302}{5.5968} = 0.9666 \\ v_{2,A} = v_f + x_{2,A} v_{fg} = 0.001061 + (0.9666)(0.88578 - 0.001061) = 0.85626 \text{ m}^3/\text{kg} \\ u_{2,A} = u_f + x_{2,A} u_{fg} = 504.50 + (0.9666)(2024.6 \text{ kJ/kg}) = 2461.5 \text{ kJ/kg} \end{array}$$

The initial and the final masses are

$$m_{1,A} = \frac{V_A}{v_{1,A}} = \frac{0.2 \text{ m}^3}{0.52422 \text{ m}^3/\text{kg}} = 0.3815 \text{ kg}$$

$$m_{2,A} = \frac{V_A}{v_{2,A}} = \frac{0.2 \text{ m}^3}{0.85626 \text{ m}^3/\text{kg}} = 0.2336 \text{ kg}$$

$$m_{2,B} = m_{1,A} - m_{2,A} = 0.3815 - 0.2336 = 0.1479 \text{ kg}$$



(*b*) The boundary work done during this process is

$$W_{b,out} = \int_1^2 P dV = P_B (V_{2,B} - 0) = P_B m_{2,B} v_{2,B}$$

Taking the contents of both the tank and the cylinder to be the system, the energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U = (\Delta U)_A + (\Delta U)_B$$

$$W_{\text{b,out}} + (\Delta U)_A + (\Delta U)_B = 0$$

or,
$$P_B m_{2,B} v_{2,B} + (m_2 u_2 - m_1 u_1)_A + (m_2 u_2)_B = 0$$
$$m_{2,B} h_{2,B} + (m_2 u_2 - m_1 u_1)_A = 0$$

Thus,

$$h_{2,B} = \frac{(m_1 u_1 - m_2 u_2)_A}{m_{2,B}} = \frac{(0.3815)(2548.5) - (0.2336)(2461.5)}{0.1479} = 2685.8 \text{ kJ/kg}$$

At 200 kPa, $h_f = 504.71$ and $h_g = 2706.3$ kJ/kg. Thus at the final state, the cylinder will contain a saturated liquid-vapor mixture since $h_f < h_2 < h_g$. Therefore,

$$T_{2,B} = T_{\text{sat@200 kPa}} = \mathbf{120.25^\circ C}$$

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7-194 Helium gas is compressed in an adiabatic closed system with an isentropic efficiency of 80%. The work input and the final temperature are to be determined.

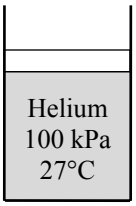
Assumptions **1** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Helium is an ideal gas.

Properties The properties of helium are $c_v = 3.1156$ kJ/kg·K and $k = 1.667$ (Table A-2*b*).

Analysis Analysis We take the helium as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{in}} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1)$$



The isentropic exit temperature is

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{900\text{ kPa}}{100\text{ kPa}}\right)^{0.667/1.667} = 722.7\text{ K}$$

The work input during isentropic process would be

$$W_{\text{s,in}} = mc_v(T_{2s} - T_1) = (3\text{ kg})(3.1156\text{ kJ/kg}\cdot\text{K})(722.7 - 300)\text{K} = 3950\text{ kJ}$$

The work input during the actual process is

$$W_{\text{in}} = \frac{W_{\text{s,in}}}{\eta} = \frac{3950\text{ kJ}}{0.80} = \mathbf{4938\text{ kJ}}$$

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7-195 R-134a undergoes a reversible, isothermal expansion in a steady-flow process. The power produced and the rate of heat transfer are to be determined.

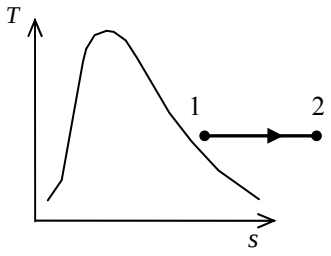
Analysis We take the steady-flow device as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the turbine, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} - \dot{W}_{\text{out}} = \dot{m}(h_2 - h_1)$$



The initial and final state properties are (Table A-13)

$$\left.\begin{array}{l} P_1 = 600 \text{ kPa} \\ T_1 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 339.47 \text{ kJ/kg} \\ s_1 = 1.1536 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ T_2 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 343.60 \text{ kJ/kg} \\ s_2 = 1.2512 \text{ kJ/kg} \cdot \text{K} \end{array}$$

Applying the entropy definition to an isothermal process gives

$$\dot{Q}_{\text{in}} = \dot{m}\int_1^2 T ds = \dot{m}T(s_2 - s_1) = (1 \text{ kg/s})(100 + 273 \text{ K})(1.2512 - 1.1536) = 36.42 \text{ kW}$$

Substituting into energy balance equation,

$$\dot{W}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{m}(h_2 - h_1) = (36.42 \text{ kW}) - (1 \text{ kg/s})(343.60 - 339.47) \text{ kJ/kg} = \mathbf{32.3 \text{ kW}}$$

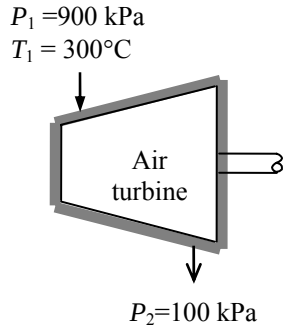
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7-196 The claim of an inventor that an adiabatic steady-flow device produces a specified amount of power is to be evaluated.

Assumptions **1** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at the anticipated average temperature of 400 K are c_p = 1.013 kJ/kg·K and k = 1.395 (Table A-2*b*).

Analysis Analysis We take the steady-flow device as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the turbine, the energy balance for this steady-flow system can be expressed in the rate form as



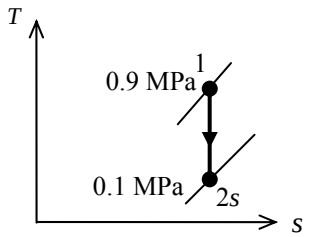
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2) \cong \dot{m}c_p(T_1 - T_2)$$

The adiabatic device would produce the maximum power if the process is isentropic. The isentropic exit temperature is



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (573\text{ K})\left(\frac{100\text{ kPa}}{900\text{ kPa}}\right)^{0.395/1.395} = 307.6\text{ K}$$

The maximum power this device can produce is then

$$\dot{W}_{\text{s,out}} = \dot{m}c_p(T_1 - T_{2s}) = (1\text{ kg/s})(1.013\text{ kJ/kg}\cdot\text{K})(573 - 307.6)\text{K} = \mathbf{269\text{ kW}}$$

This is greater than the power claim of the inventor, and thus the claim is valid.

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7-197 A gas is adiabatically expanded in a piston-cylinder device with a specified isentropic efficiency. It is to be determined if air or neon will produce more work.

Assumptions **1** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air and helium are ideal gases.

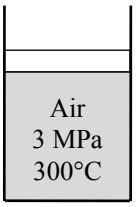
Properties The properties of air at room temperature are c_v = 0.718 kJ/kg·K and k = 1.4. The properties of neon at room temperature are c_v = 0.6179 kJ/kg·K and k = 1.667 (Table A-2*a*).

Analysis We take the gas as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1)$$

$$W_{\text{out}} = mc_v(T_1 - T_2)$$



The isentropic exit temperature is

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300 + 273\ \text{K})\left(\frac{80\ \text{kPa}}{3000\ \text{kPa}}\right)^{0.4/1.4} = 203.4\ \text{K}$$

The work output during the actual process is

$$w_{\text{in}} = \eta c_v(T_1 - T_{2s}) = (0.90)(0.718\ \text{kJ/kg}\cdot\text{K})(573 - 203.4)\text{K} = \mathbf{239\ kJ/kg}$$

Repeating the same calculations for neon,

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (300 + 273\ \text{K})\left(\frac{80\ \text{kPa}}{3000\ \text{kPa}}\right)^{0.667/1.667} = 134.4\ \text{K}$$

$$w_{\text{in}} = \eta c_v(T_1 - T_{2s}) = (0.80)(0.6179\ \text{kJ/kg}\cdot\text{K})(573 - 134.4)\text{K} = \mathbf{217\ kJ/kg}$$

Air will produce more work.

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7-198 Refrigerant-134a is expanded adiabatically in a capillary tube. The rate of entropy generation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

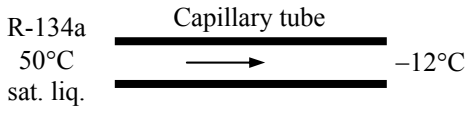
Analysis The rate of entropy generation within the expansion device during this process can be determined by applying the rate form of the entropy balance on the system. Noting that the system is adiabatic and thus there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{gen}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{system}}^{\nearrow 0 \text{ (steady)}}_{\text{Rate of change of entropy}}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1)$$

$$s_{gen} = s_2 - s_1$$



It may be easily shown with an energy balance that the enthalpy remains constant during the throttling process. The properties of the refrigerant at the inlet and exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} T_1 = 50°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 123.50 \text{ kJ/kg} \cdot \text{K} \\ s_1 = 0.44193 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = -12°\text{C} \\ h_2 = h_1 = 123.50 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{h_2 - h_f}{h_{fg}} = \dfrac{123.50 - 35.92}{207.38} = 0.4223 \\ s_2 = s_f + x_2 s_{fg} = 0.14504 + (0.4223)(0.79406) = 0.48038 \text{ kJ/kg} \cdot \text{K} \end{array}$$

Substituting,

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) = (0.2 \text{ kg/s})(0.48038 - 0.44193) \text{ kJ/kg} \cdot \text{K} = \mathbf{0.00769 \text{ kW/K}}$$

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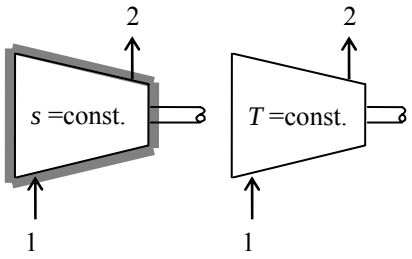
7-199 Air is compressed steadily by a compressor from a specified state to a specified pressure. The minimum power input required is to be determined for the cases of adiabatic and isothermal operation.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats. **4** The process is reversible since the work input to the compressor will be minimum when the compression process is reversible.

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1).

Analysis (*a*) For the adiabatic case, the process will be reversible and adiabatic (i.e., isentropic), thus the isentropic relations are applicable.

$$T_1 = 290\ \text{K} \longrightarrow P_{r_1} = 1.2311 \text{ and } h_1 = 290.16\ \text{kJ/kg}$$



and

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{700\ \text{kPa}}{100\ \text{kPa}}(1.2311) = 8.6177 \quad \rightarrow \quad \begin{matrix} T_2 = 503.3\ \text{K} \\ h_2 = 506.45\ \text{kJ/kg} \end{matrix}$$

The energy balance for the compressor, which is a steady-flow system, can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \ \rightarrow \ \dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor is determined to be

$$\dot{W}_{\text{in}} = (5/60\ \text{kg/s})(506.45 - 290.16)\text{kJ/kg} = \mathbf{18.0\ kW}$$

(*b*) In the case of the reversible isothermal process, the steady-flow energy balance becomes

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \ \rightarrow \dot{W}_{\text{in}} + \dot{m}h_1 - \dot{Q}_{\text{out}} = \dot{m}h_2 \ \rightarrow \ \dot{W}_{\text{in}} = \dot{Q}_{\text{out}} + \dot{m}(h_2 - h_1)^{\nearrow 0} = \dot{Q}_{\text{out}}$$

since $h = h(T)$ for ideal gases, and thus the enthalpy change in this case is zero. Also, for a reversible isothermal process,

$$\dot{Q}_{\text{out}} = \dot{m}T(s_1 - s_2) = -\dot{m}T(s_2 - s_1)$$

where

$$s_2 - s_1 = \left(s_2^\circ - s_1^\circ\right)^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -R\ln\frac{P_2}{P_1} = -(0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{700\ \text{kPa}}{100\ \text{kPa}} = -0.5585\ \text{kJ/kg}\cdot\text{K}$$

Substituting, the power input for the reversible isothermal case becomes

$$\dot{W}_{\text{in}} = -(5/60\ \text{kg/s})(290\ \text{K})(-0.5585\ \text{kJ/kg}\cdot\text{K}) = \mathbf{13.5\ kW}$$

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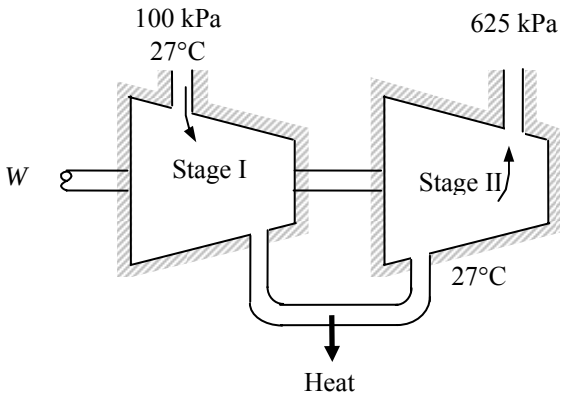
7-200 Air is compressed in a two-stage ideal compressor with intercooling. For a specified mass flow rate of air, the power input to the compressor is to be determined, and it is to be compared to the power input to a single-stage compressor.

Assumptions **1** The compressor operates steadily. **2** Kinetic and potential energies are negligible. **3** The compression process is reversible adiabatic, and thus isentropic. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). The specific heat ratio of air is $k = 1.4$ (Table A-2).

Analysis The intermediate pressure between the two stages is

$$P_x = \sqrt{P_1 P_2} = \sqrt{(100\ \text{kPa})(625\ \text{kPa})} = 250\ \text{kPa}$$



The compressor work across each stage is the same, thus total compressor work is twice the compression work for a single stage:

$$\begin{aligned} w_{\text{comp,in}} &= (2)(w_{\text{comp,in,I}}) = 2\frac{kRT_1}{k-1}\left((P_x/P_1)^{(k-1)/k} - 1\right) \\ &= 2\frac{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(300\ \text{K})}{1.4-1}\left(\left(\frac{250\ \text{kPa}}{100\ \text{kPa}}\right)^{0.4/1.4} - 1\right) \\ &= 180.4\ \text{kJ/kg} \end{aligned}$$

and

$$\dot{W}_{\text{in}} = \dot{m} w_{\text{comp,in}} = (0.15\ \text{kg/s})(180.4\ \text{kJ/kg}) = \mathbf{27.1\ kW}$$

The work input to a single-stage compressor operating between the same pressure limits would be

$$w_{\text{comp,in}} = \frac{kRT_1}{k-1}\left((P_2/P_1)^{(k-1)/k} - 1\right) = \frac{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(300\ \text{K})}{1.4-1}\left(\left(\frac{625\ \text{kPa}}{100\ \text{kPa}}\right)^{0.4/1.4} - 1\right) = 207.4\ \text{kJ/kg}$$

and

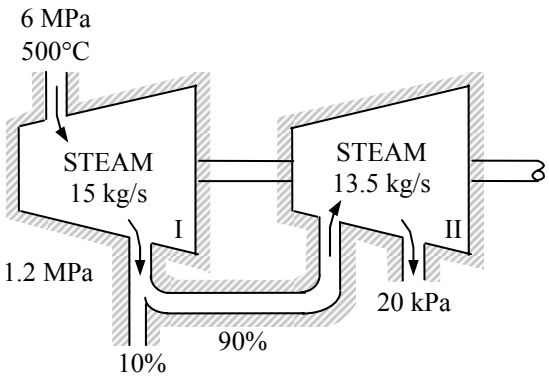
$$\dot{W}_{\text{in}} = \dot{m} w_{\text{comp,in}} = (0.15\ \text{kg/s})(207.4\ \text{kJ/kg}) = \mathbf{31.1\ kW}$$

Discussion Note that the power consumption of the compressor decreases significantly by using 2-stage compression with intercooling.

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7-201 Steam expands in a two-stage adiabatic turbine from a specified state to specified pressure. Some steam is extracted at the end of the first stage. The power output of the turbine is to be determined for the cases of 100% and 88% isentropic efficiencies.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible.



Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{matrix} P_1 = 6\ \text{MPa} \\ T_1 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3423.1\ \text{kJ/kg} \\ s_1 = 6.8826\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 1.2\ \text{MPa} \\ s_2 = s_1 \end{matrix}\right\} h_2 = 2962.8\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 20\ \text{kPa} \\ s_3 = s_1 \end{matrix}\right\} \begin{matrix} x_3 = \dfrac{s_3 - s_f}{s_{fg}} = \dfrac{6.8826 - 0.8320}{7.0752} = 0.8552 \\ h_3 = h_f + x_3 h_{fg} = 251.42 + (0.8552)(2357.5) = 2267.5\ \text{kJ/kg} \end{matrix}$$

Analysis (*a*) The mass flow rate through the second stage is

$$\dot{m}_3 = 0.9\dot{m}_1 = (0.9)(15\ \text{kg/s}) = 13.5\ \text{kg/s}$$

We take the entire turbine, including the connection part between the two stages, as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters the turbine and two fluid streams leave, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\cancelto{0}{}\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 = (\dot{m}_1 - \dot{m}_3)h_2 + \dot{W}_{\text{out}} + \dot{m}_3 h_3$$

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m}_1 h_1 - (\dot{m}_1 - \dot{m}_3)h_2 - \dot{m}_3 h_3 \\ &= \dot{m}_1(h_1 - h_2) + \dot{m}_3(h_2 - h_3) \end{aligned}$$

Substituting, the power output of the turbine is

$$\dot{W}_{\text{out}} = (15\ \text{kg/s})(3423.1 - 2962.8)\text{kJ/kg} + (13.5\ \text{kg})(2962.8 - 2267.5)\text{kJ/kg} = \mathbf{16{,}291\ kW}$$

(*b*) If the turbine has an isentropic efficiency of 88%, then the power output becomes

$$\dot{W}_a = \eta_T \dot{W}_s = (0.88)(16{,}291\ \text{kW}) = \mathbf{14{,}336\ kW}$$

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7-202 Steam expands in an 84% efficient two-stage adiabatic turbine from a specified state to a specified pressure. Steam is reheated between the stages. For a given power output, the mass flow rate of steam through the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible.

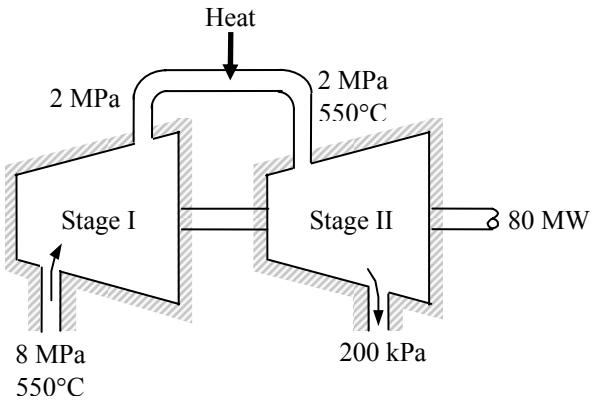
Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{matrix} P_1 = 8\ \text{MPa} \\ T_1 = 550°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3521.8\ \text{kJ/kg} \\ s_1 = 6.8800\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_{2s} = 2\ \text{MPa} \\ s_{2s} = s_1 \end{matrix}\right\} h_{2s} = 3089.7\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 2\ \text{MPa} \\ T_3 = 550°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3579.0\ \text{kJ/kg} \\ s_3 = 7.5725\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_{4s} = 200\ \text{kPa} \\ s_{4s} = s_3 \end{matrix}\right\} h_{4s} = 2901.7\ \text{kJ/kg}$$



Analysis The power output of the actual turbine is given to be 80 MW. Then the power output for the isentropic operation becomes

$$\dot{W}_{s,\text{out}} = \dot{W}_{a,\text{out}} / \eta_T = (80{,}000\ \text{kW}) / 0.84 = 95{,}240\ \text{kW}$$

We take the entire turbine, excluding the reheat section, as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system in isentropic operation can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\begin{matrix}\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\begin{matrix}\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}\end{matrix}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{m}h_3 = \dot{m}h_{2s} + \dot{m}h_{4s} + \dot{W}_{s,\text{out}}$$

$$\dot{W}_{s,\text{out}} = \dot{m}[(h_1 - h_{2s}) + (h_3 - h_{4s})]$$

Substituting,

$$95{,}240\ \text{kJ/s} = \dot{m}\left[(3521.8 - 3089.7)\text{kJ/kg} + (3579.0 - 2901.7)\text{kJ/kg}\right]$$

which gives

$$\dot{m} = \mathbf{85.8\ kg/s}$$

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7-203 Refrigerant-134a is compressed by a 1.3-kW adiabatic compressor from a specified state to another specified state. The isentropic efficiency, the volume flow rate at the inlet, and the maximum flow rate at the compressor inlet are to be determined.

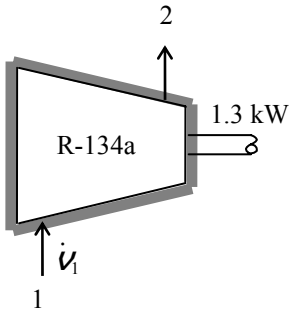
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the R-134a tables (Tables A-11 through A-13)

$$\left.\begin{matrix} P_1 = 100 \text{ kPa} \\ T_1 = -20°\text{C} \end{matrix}\right\} \begin{matrix} \nu_1 = 0.19841 \text{ m}^3/\text{kg} \\ h_1 = 239.50 \text{ kJ/kg} \\ s_1 = 0.9721 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 800 \text{ kPa} \\ T_2 = 60°\text{C} \end{matrix}\right\} h_2 = 296.81 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 800 \text{ kPa} \\ s_{2s} = s_1 \end{matrix}\right\} h_{2s} = 284.07 \text{ kJ/kg}$$



Analysis (a) The isentropic efficiency is determined from its definition,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{284.07 - 239.50}{296.81 - 239.50} = 0.778 = \mathbf{77.8\%}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

Then the mass and volume flow rates of the refrigerant are determined to be

$$\dot{m} = \frac{\dot{W}_{\text{a,in}}}{h_{2a} - h_1} = \frac{1.3 \text{ kJ/s}}{(296.81 - 239.50)\text{kJ/kg}} = 0.02269 \text{ kg/s}$$

$$\dot{V}_1 = \dot{m}\nu_1 = (0.02269 \text{ kg/s})(0.19841 \text{ m}^3/\text{kg}) = 0.004502 \text{ m}^3/\text{s} = \mathbf{270 \text{ L/min}}$$

(*c*) The volume flow rate will be a maximum when the process is isentropic, and it is determined similarly from the steady-flow energy equation applied to the isentropic process. It gives

$$\dot{m}_{\max} = \frac{\dot{W}_{\text{s,in}}}{h_{2s} - h_1} = \frac{1.3 \text{ kJ/s}}{(284.07 - 239.50)\text{kJ/kg}} = 0.02917 \text{ kg/s}$$

$$\dot{V}_{1,\max} = \dot{m}_{\max}\nu_1 = (0.02917 \text{ kg/s})(0.19841 \text{ m}^3/\text{kg}) = 0.005788 \text{ m}^3/\text{s} = \mathbf{347 \text{ L/min}}$$

Discussion Note that the raising the isentropic efficiency of the compressor to 100% would increase the volumetric flow rate by more than 25%.

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7-204 An adiabatic compressor is powered by a direct-coupled steam turbine, which also drives a generator. The net power delivered to the generator and the rate of entropy generation are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The devices are adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

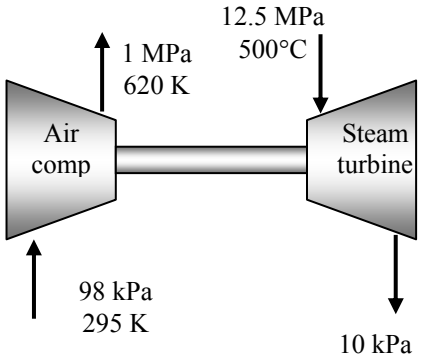
Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1). From the steam tables (Tables A-4 through 6) and air table (Table A-17),

$$T_1 = 295 \text{ K} \longrightarrow h_1 = 295.17 \text{ kJ/kg},\ s_1^\circ = 1.68515 \text{ kJ/kg}\cdot\text{K}$$

$$T_2 = 620 \text{ K} \longrightarrow h_1 = 628.07 \text{ kJ/kg},\ s_2^\circ = 2.44356 \text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{matrix} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3343.6 \text{ kJ/kg} \\ s_3 = 6.4651 \text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{matrix}\right\} \begin{matrix} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 0.6492 + (0.92)(7.4996) = 7.5489 \text{ kJ/kg}\cdot\text{K} \end{matrix}$$



Analysis There is only one inlet and one exit for either device, and thus $\dot{m}_{in} = \dot{m}_{out} = \dot{m}$. We take either the turbine or the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for either steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\begin{matrix}\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\begin{matrix}\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}\end{matrix}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

For the turbine and the compressor it becomes

Compressor: $\dot{W}_{comp,in} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2 \rightarrow \dot{W}_{comp,in} = \dot{m}_{air}(h_2 - h_1)$

Turbine: $\dot{m}_{steam} h_3 = \dot{W}_{turb,out} + \dot{m}_{steam} h_4 \ \rightarrow \ \dot{W}_{turb,out} = \dot{m}_{steam}(h_3 - h_4)$

Substituting,

$$\dot{W}_{comp,in} = (10 \text{ kg/s})(628.07 - 295.17)\text{kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{turb,out} = (25 \text{ kg/s})(3343.6 - 2392.5)\text{kJ/kg} = 23{,}777 \text{ kW}$$

Therefore,

$$\dot{W}_{net,out} = \dot{W}_{turb,out} - \dot{W}_{comp,in} = 23{,}777 - 3329 = \mathbf{20{,}448\ kW}$$

Noting that the system is adiabatic, the total rate of entropy change (or generation) during this process is the sum of the entropy changes of both fluids,

$$\dot{S}_{gen} = \dot{m}_{air}(s_2 - s_1) + \dot{m}_{steam}(s_4 - s_3)$$

where

$$\dot{m}_{air}(s_2 - s_1) = \dot{m}\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right) = (10 \text{ kg/s})\left(2.44356 - 1.68515 - 0.287\ln\frac{1000 \text{ kPa}}{98 \text{ kPa}}\right)\text{kJ/kg}\cdot\text{K} = 0.92 \text{ kW/K}$$

$$\dot{m}_{steam}(s_4 - s_3) = (25 \text{ kg/s})(7.5489 - 6.4651)\text{kJ/kg}\cdot\text{K} = 27.1 \text{ kW/K}$$

Substituting, the total rate of entropy generation is determined to be

$$\dot{S}_{gen,total} = \dot{S}_{gen,comp} + \dot{S}_{gen,turb} = 0.92 + 27.1 = \mathbf{28.02\ kW/K}$$

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7-205 Problem 7-204 is reconsidered. The isentropic efficiencies for the compressor and turbine are to be determined, and then the effect of varying the compressor efficiency over the range 0.6 to 0.8 and the turbine efficiency over the range 0.7 to 0.95 on the net work for the cycle and the entropy generated for the process is to be investigated. The net work is to be plotted as a function of the compressor efficiency for turbine efficiencies of 0.7, 0.8, and 0.9.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

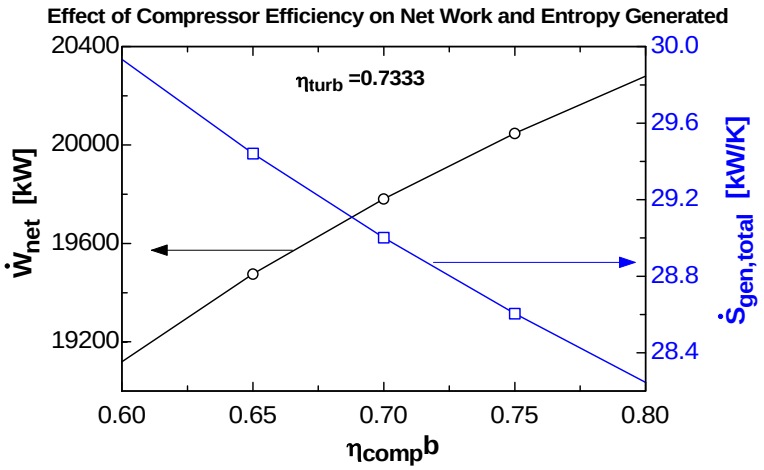
```
"Input Data"
m_dot_air = 10 [kg/s] "air compressor (air) data"
T_air[1]=(295-273) "[C]" "We will input  temperature in C"
P_air[1]=98 [kPa]
T_air[2]=(700-273) "[C]"
P_air[2]=1000 [kPa]
m_dot_st=25 [kg/s] "steam turbine (st) data"
T_st[1]=500 [C]
P_st[1]=12500 [kPa]
P_st[2]=10 [kPa]
x_st[2]=0.92 "quality"
"Compressor Analysis:"
"Conservation of mass for the compressor m_dot_air_in = m_dot_air_out =m_dot_air"
"Conservation of energy for the compressor is:"
E_dot_comp_in - E_dot_comp_out = DELTAE_dot_comp
DELTAE_dot_comp = 0                    "Steady flow requirement"
E_dot_comp_in=m_dot_air*(enthalpy(air,T=T_air[1])) + W_dot_comp_in
E_dot_comp_out=m_dot_air*(enthalpy(air,T=T_air[2]))
"Compressor adiabatic efficiency:"
Eta_comp=W_dot_comp_in_isen/W_dot_comp_in
W_dot_comp_in_isen=m_dot_air*(enthalpy(air,T=T_air_isen[2])-enthalpy(air,T=T_air[1]))
s_air[1]=entropy(air,T=T_air[1],P=P_air[1])
s_air[2]=entropy(air,T=T_air[2],P=P_air[2])
s_air_isen[2]=entropy(air, T=T_air_isen[2],P=P_air[2])
s_air_isen[2]=s_air[1]
"Turbine Analysis:"
"Conservation of mass for the turbine m_dot_st_in = m_dot_st_out =m_dot_st"
"Conservation of energy for the turbine is:"
E_dot_turb_in - E_dot_turb_out = DELTAE_dot_turb
DELTAE_dot_turb = 0                    "Steady flow requirement"
E_dot_turb_in=m_dot_st*h_st[1]
h_st[1]=enthalpy(steam,T=T_st[1], P=P_st[1])
E_dot_turb_out=m_dot_st*h_st[2]+W_dot_turb_out
h_st[2]=enthalpy(steam,P=P_st[2], x=x_st[2])
"Turbine adiabatic efficiency:"
Eta_turb=W_dot_turb_out/W_dot_turb_out_isen
W_dot_turb_out_isen=m_dot_st*(h_st[1]-h_st_isen[2])
s_st[1]=entropy(steam,T=T_st[1],P=P_st[1])
h_st_isen[2]=enthalpy(steam, P=P_st[2],s=s_st[1])
"Note: When Eta_turb is specified as an independent variable in
the Parametric Table, the iteration process may put the steam state 2 in the
superheat region, where the quality is undefined. Thus, s_st[2], T_st[2] are
calculated at P_st[2], h_st[2] and not P_st[2] and x_st[2]"
s_st[2]=entropy(steam,P=P_st[2],h=h_st[2])
T_st[2]=temperature(steam,P=P_st[2], h=h_st[2])
s_st_isen[2]=s_st[1]
"Net work done by the process:"
W_dot_net=W_dot_turb_out-W_dot_comp_in
"Entropy generation:"
"Since both the compressor and turbine are adiabatic, and thus there is no heat transfer
to the surroundings, the entropy generation for the two steady flow devices becomes:"
S_dot_gen_comp=m_dot_air*( s_air[2]-s_air[1])
```

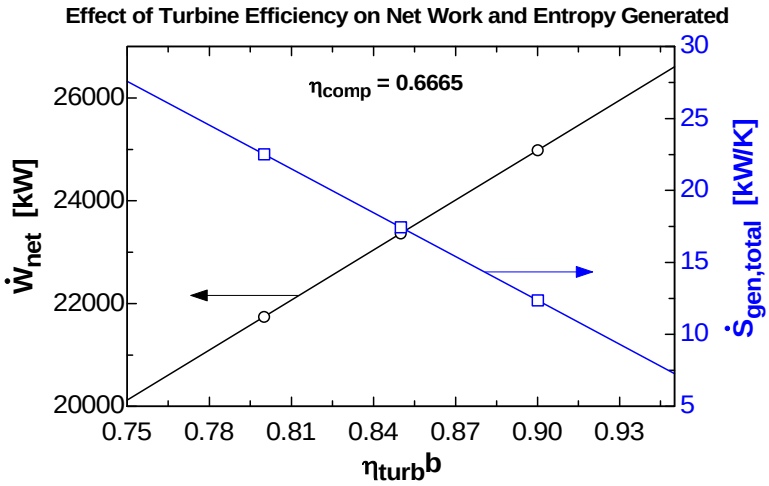
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```
S_dot_gen_turb=m_dot_st*(s_st[2]-s_st[1])
S_dot_gen_total=S_dot_gen_comp+S_dot_gen_turb
"To generate the data for Plot Window 1, Comment out the line ' T_air[2]=(700-273) C'
and select values for Eta_comp in the Parmetric Table, then press F3 to solve the table.
EES then solves for the unknown value of T_air[2] for each Eta_comp."
"To generate the data for Plot Window 2, Comment out the two lines ' x_st[2]=0.92 quality '
and ' h_st[2]=enthalpy(steam,P=P_st[2], x=x_st[2]) ' and select values for Eta_turb in the
Parmetric Table, then press F3 to solve the table.  EES then solves for the h_st[2] for each
Eta_turb."
```

W_{net} [kW]	$S_{gentotal}$ [kW/K]	η_{turb}	η_{comp}
20124	27.59	0.75	0.6665
21745	22.51	0.8	0.6665
23365	17.44	0.85	0.6665
24985	12.36	0.9	0.6665
26606	7.281	0.95	0.6665

W_{net} [kW]	$S_{gentotal}$ [kW/K]	η_{turb}	η_{comp}
19105	30	0.7327	0.6
19462	29.51	0.7327	0.65
19768	29.07	0.7327	0.7
20033	28.67	0.7327	0.75
20265	28.32	0.7327	0.8





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7-206 The pressure in a hot water tank rises to 2 MPa, and the tank explodes. The explosion energy of the water is to be determined, and expressed in terms of its TNT equivalence.

Assumptions **1** The expansion process during explosion is isentropic. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer with the surroundings during explosion is negligible.

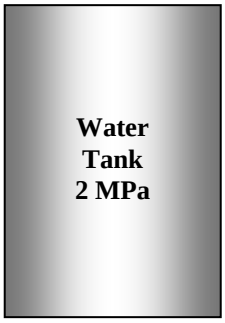
Properties The explosion energy of TNT is 3250 kJ/kg. From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 2 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@2\text{ MPa}} = 0.001177 \text{ m}^3/\text{kg} \\ u_1 = u_{f@2\text{ MPa}} = 906.12 \text{ kJ/kg} \\ s_1 = s_{f@2\text{ MPa}} = 2.4467 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} u_f = 417.40, \quad u_{fg} = 2088.2 \text{ kJ/kg} \\ s_f = 1.3028, \quad s_{fg} = 6.0562 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{2.4467 - 1.3028}{6.0562} = 0.1889$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + (0.1889)(2088.2) = 811.83 \text{ kJ/kg}$$



Analysis We idealize the water tank as a closed system that undergoes a reversible adiabatic process with negligible changes in kinetic and potential energies. The work done during this idealized process represents the explosive energy of the tank, and is determined from the closed system energy balance to be

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$E_{\text{exp}} = W_{\text{b,out}} = m(u_1 - u_2)$$

where

$$m = \frac{V}{v_1} = \frac{0.080 \text{ m}^3}{0.001177 \text{ m}^3/\text{kg}} = 67.99 \text{ kg}$$

Substituting,

$$E_{\text{exp}} = (67.99 \text{ kg})(906.12 - 811.83)\text{kJ/kg} = 6410 \text{ kJ}$$

which is equivalent to

$$m_{\text{TNT}} = \frac{6410 \text{ kJ}}{3250 \text{ kJ/kg}} = \mathbf{1.972 \text{ kg TNT}}$$

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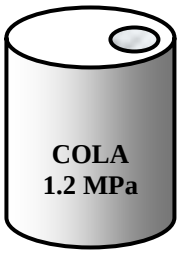
7-207 A 0.35-L canned drink explodes at a pressure of 1.2 MPa. The explosive energy of the drink is to be determined, and expressed in terms of its TNT equivalence.

Assumptions **1** The expansion process during explosion is isentropic. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer with the surroundings during explosion is negligible. **4** The drink can be treated as pure water.

Properties The explosion energy of TNT is 3250 kJ/kg. From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 1.2 \text{ MPa} \\ \text{Comp. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@1.2\text{ MPa}} = 0.001138 \text{ m}^3/\text{kg} \\ u_1 = u_{f@1.2\text{ MPa}} = 796.96 \text{ kJ/kg} \\ s_1 = s_{f@1.2\text{ MPa}} = 2.2159 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} u_f = 417.40, \quad u_{fg} = 2088.2 \text{ kJ/kg} \\ s_f = 1.3028, \quad s_{fg} = 6.0562 \text{ kJ/kg}\cdot\text{K} \end{array}$$



$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{2.2159 - 1.3028}{6.0562} = 0.1508$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + (0.1508)(2088.2) = 732.26 \text{ kJ/kg}$$

Analysis We idealize the canned drink as a closed system that undergoes a reversible adiabatic process with negligible changes in kinetic and potential energies. The work done during this idealized process represents the explosive energy of the can, and is determined from the closed system energy balance to be

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$E_{\text{exp}} = W_{\text{b,out}} = m(u_1 - u_2)$$

where

$$m = \frac{V}{v_1} = \frac{0.00035 \text{ m}^3}{0.001138 \text{ m}^3/\text{kg}} = 0.3074 \text{ kg}$$

Substituting,

$$E_{\text{exp}} = (0.3074 \text{ kg})(796.96 - 732.26)\text{kJ/kg} = \mathbf{19.9\ kJ}$$

which is equivalent to

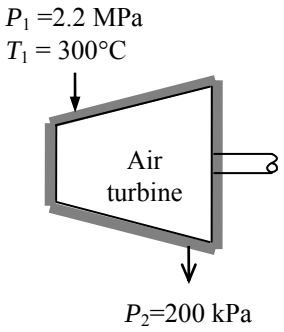
$$m_{\text{TNT}} = \frac{19.9 \text{ kJ}}{3250 \text{ kJ/kg}} = \mathbf{0.00612\ \ kg\ TNT}$$

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7-208 Air is expanded by an adiabatic turbine with an isentropic efficiency of 85%. The outlet temperature and the work produced are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at the anticipated average temperature of 400 K are c_p = 1.013 kJ/kg·°C and k = 1.395 (Table A-2a).



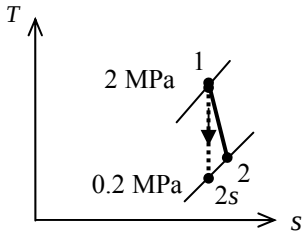
Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the turbine, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$



The isentropic exit temperature is

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300 + 273\text{ K})\left(\frac{200\text{ kPa}}{2200\text{ kPa}}\right)^{0.395/1.395} = 290.6\text{ K}$$

From the definition of the isentropic efficiency,

$$w_{a,\text{out}} = \eta_T w_{s,\text{out}} = \eta_T c_p(T_1 - T_{2s}) = (0.85)(1.013\text{ kJ/kg}\cdot\text{K})(573 - 290.6)\text{K} = \mathbf{243.2\ kJ/kg}$$

The actual exit temperature is then

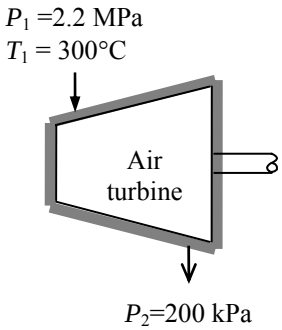
$$w_{a,\text{out}} = c_p(T_1 - T_{2a}) \longrightarrow T_{2a} = T_1 - \frac{w_{a,\text{out}}}{c_p} = T_1 - \frac{w_{a,\text{out}}}{c_p} = 573\text{ K} - \frac{243.2\text{ kJ/kg}}{1.013\text{ kJ/kg}\cdot\text{K}} = \mathbf{333\ K}$$

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7-209 Air is expanded by an adiabatic turbine with an isentropic efficiency of 85%. The outlet temperature, the work produced, and the entropy generation are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at the anticipated average temperature of 400 K are c_p = 1.013 kJ/kg·°C and k = 1.395 (Table A-2b). Also, R = 0.287 kJ/kg·K (Table A-2a).



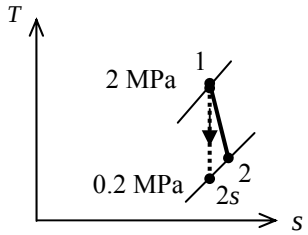
Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the turbine, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$



The isentropic exit temperature is

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (300 + 273\text{ K})\left(\frac{200\text{ kPa}}{2200\text{ kPa}}\right)^{0.395/1.395} = 290.6\text{ K}$$

From the definition of the isentropic efficiency,

$$w_{a,\text{out}} = \eta_T w_{s,\text{out}} = \eta_T c_p(T_1 - T_{2s}) = (0.90)(1.013\text{ kJ/kg}\cdot\text{K})(573 - 290.6)\text{K} = \mathbf{257.5\ kJ/kg}$$

The actual exit temperature is then

$$w_{a,\text{out}} = c_p(T_1 - T_{2a}) \longrightarrow T_{2a} = T_1 - \frac{w_{a,\text{out}}}{c_p} = T_1 - \frac{w_{a,\text{out}}}{c_p} = 573\text{ K} - \frac{257.5\text{ kJ/kg}}{1.013\text{ kJ/kg}\cdot\text{K}} = \mathbf{318.8\ K}$$

The rate of entropy generation in the turbine is determined by applying the rate form of the entropy balance on the turbine:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } \dot{Q} = 0)$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

$$s_{\text{gen}} = s_2 - s_1$$

Then, from the entropy change relation of an ideal gas,

$$s_{\text{gen}} = s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (1.013\text{ kJ/kg}\cdot\text{K})\ln\frac{318.8\text{ K}}{573\text{ K}} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{200\text{ kPa}}{2200\text{ kPa}}$$

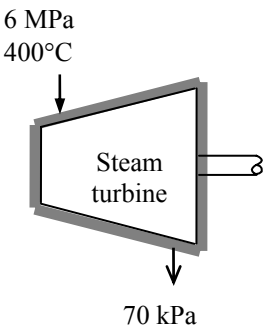
$$= \mathbf{0.0944\ kJ/kg}\cdot\mathbf{K}$$

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7-210 A throotle valve is placed in the steam line of an adiabatic turbine. The work output is to be determined with and without throttle valve cases.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as



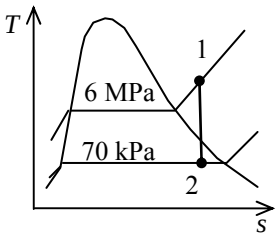
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$

When the valve is fully open, from the steam tables (Tables A-4 through A-6),



$$\left.\begin{array}{l} P_1 = 6 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3178.3 \text{ kJ/kg} \\ s_1 = 6.5432 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 70 \text{ kPa} \\ s_3 = s_1 = 6.5432 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} \begin{array}{l} x_3 = 0.8511 \\ h_3 = 2319.6 \text{ kJ/kg} \end{array}$$

Then,

$$w_{\text{out}} = h_1 - h_3 = 3178.3 - 2319.6 = \mathbf{858.6\ kJ/kg}$$

The flow through the throttle valve is isenthalpic (constant entahlpy). When the valve is partially closed, from the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_2 = 3 \text{ MPa} \\ h_2 = h_1 = 3178.3 \text{ kJ/kg} \end{array}\right\} s_2 = 6.8427 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} P_3 = 70 \text{ kPa} \\ s_3 = s_2 = 6.8427 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} \begin{array}{l} x_3 = 0.8988 \\ h_3 = 2428.4 \text{ kJ/kg} \end{array}$$

Then,

$$w_{\text{out}} = h_2 - h_3 = 3178.3 - 2428.4 = \mathbf{749.9\ kJ/kg}$$

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7-211 Oxygen tanks are filled by an isentropic compressor. The work required by the compressor and the heat transfer from the tanksa re to be determined.

Assumptions **1** Changes in the kinetic and potential energies are negligible. **4** Oxygen is an ideal gas with constant specific heats.

Properties The properties of oxygen at room temperature are R = 0.2598 kPa·m^3/kg·K, c_p = 0.918 kJ/kg·K, c_v = 0.658 kJ/kg·K, k = 1.395 (Table A-2a).

Analysis As the tank is being filled, the pressure in the tank increases as time passes, but the temperature does not. In the line between the compressor and tank, the pressure always matches that in the compressor, and as a result, the temperature changes in this line with time. Applying the isentropic process relations to the compressor yields the temperature in this line as

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

Reducing the mass balance to the conditions of the tank gives

$$\frac{dm_{cv}}{dt} = \dot{m}_{\text{in}}$$

but

$$m_{cv} = \frac{P\boldsymbol{V}}{RT}$$

Combining these two results produces

$$\dot{m}_{\text{in}} = \frac{d(P\boldsymbol{V}/RT)}{\boldsymbol{dt}} = \frac{\boldsymbol{V}}{RT}\frac{dP}{dt}$$

where the last step incorporates the fact that the tank volume and temperature do not change as time passes. The mass in the tank at the end of the compression is

$$m_{\text{final}} = \frac{P_{\text{final}}\boldsymbol{V}}{RT} = \frac{(13{,}000\,\text{kPa})(1\,\text{m}^3)}{(0.2598\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})} = 170.8\,\text{kg}$$

Adapting the first law to the tank produces

$$\dot{Q} = \frac{d(mu)}{dt} - \dot{m}_{\text{in}}h_{\text{in}} = c_v T\frac{dm}{dt} - c_p T_2\dot{m}_{\text{in}}$$
$$= c_v T\frac{dm}{dt} - c_p T_1\left(\frac{P}{P_1}\right)^{(k-1)/k}\frac{\boldsymbol{V}}{RT}\frac{dP}{dt}$$

Integrating this result from the beginning of the compression to the end of compression yields

$$Q_{\text{in}} = c_v T\int_i^f dm - c_p T_1\left(\frac{1}{P_1}\right)^{(k-1)/k}\frac{\boldsymbol{V}}{RT}\int_i^f P^{(k-1)/k}\,dP$$
$$= c_v T m_f - c_p T_1\left(\frac{1}{P_1}\right)^{(k-1)/k}\frac{\boldsymbol{V}}{RT}\left(\frac{k}{2k-1}\right)P_f{}^{(k-1)/k}P_f$$
$$= c_v T m_f - c_p T_1\left(\frac{P_f}{P_1}\right)^{(k-1)/k}\left(\frac{k}{2k-1}\right)m_f$$
$$= (0.658)(293)(170.8) - (0.918)(293)\left(\frac{13{,}000}{150}\right)^{0.395/1.395}\left(\frac{1.395}{2(1.395)-1}\right)(170.8)$$
$$= \mathbf{-93{,}720\,kJ}$$

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The negative sign indicates that heat is lost from the tank. Adopting the first law to the compressor and tank as the system gives

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} = \frac{d(mu)}{dt} - \dot{m}_{\text{in}} h_{\text{in}}$$

Recognizing that the enthalpy of the oxygen entering the compressor remains constant, this results integrates to

$$Q_{\text{in}} + W_{\text{in}} = m_f u_f - h_1 m_f$$

or

$$\begin{aligned} W_{\text{in}} &= -Q_{\text{in}} - m_f h_1 + m_f u_f \\ &= -Q_{\text{in}} - m_f (c_p T_1 - c_v T_f) \\ &= -Q_{\text{in}} - m_f (c_p - c_v) T \\ &= 93{,}720 - (170.8)(0.918 - 0.658) \times 293 \\ &= \mathbf{80{,}710\ kJ} \end{aligned}$$

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7-212 Two rigid tanks that contain water at different states are connected by a valve. The valve is opened and steam flows from tank A to tank B until the pressure in tank A drops to a specified value. Tank B loses heat to the surroundings. The final temperature in each tank and the entropy generated during this process are to be determined.

Assumptions **1** Tank A is insulated, and thus heat transfer is negligible. **2** The water that remains in tank A undergoes a reversible adiabatic process. **3** The thermal energy stored in the tanks themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible. **5** There are no work interactions.

Analysis (*a*) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

Tank A:

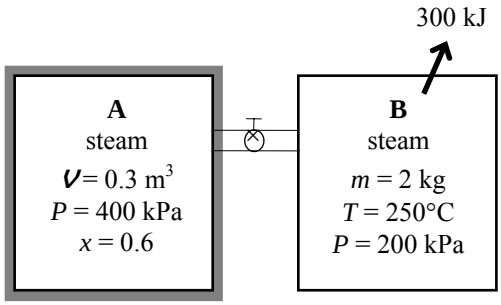
$$\left.\begin{array}{l} P_1 = 400 \text{ kPa} \\ x_1 = 0.6 \end{array}\right\} \begin{array}{l} v_{1,A} = v_f + x_1 v_{fg} = 0.001084 + (0.6)(0.46242 - 0.001084) = 0.27788 \text{ m}^3/\text{kg} \\ u_{1,A} = u_f + x_1 u_{fg} = 604.22 + (0.6)(1948.9) = 1773.6 \text{ kJ/kg} \\ s_{1,A} = s_f + x_1 s_{fg} = 1.7765 + (0.6)(5.1191) = 4.8479 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$T_{2,A} = T_{\text{sat@200 kPa}} = \mathbf{120.2°C}$$

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ s_2 = s_1 \\ (\text{sat. mixture}) \end{array}\right\} \begin{array}{l} x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{4.8479 - 1.5302}{5.59680} = 0.5928 \\ v_{2,A} = v_f + x_{2,A} v_{fg} = 0.001061 + (0.5928)(0.8858 - 0.001061) = 0.52552 \text{ m}^3/\text{kg} \\ u_{2,A} = u_f + x_{2,A} u_{fg} = 504.50 + (0.5928)(2024.6 \text{ kJ/kg}) = 1704.7 \text{ kJ/kg} \end{array}$$

Tank B:

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 250°\text{C} \end{array}\right\} \begin{array}{l} v_{1,B} = 1.1989 \text{ m}^3/\text{kg} \\ u_{1,B} = 2731.4 \text{ kJ/kg} \\ s_{1,B} = 7.7100 \text{ kJ/kg} \cdot \text{K} \end{array}$$



The initial and the final masses in tank A are

$$m_{1,A} = \frac{V_A}{v_{1,A}} = \frac{0.3 \text{ m}^3}{0.27788 \text{ m}^3/\text{kg}} = 1.080 \text{ kg}$$

and

$$m_{2,A} = \frac{V_A}{v_{2,A}} = \frac{0.3 \text{ m}^3}{0.52552 \text{ m}^3/\text{kg}} = 0.5709 \text{ kg}$$

Thus, 1.080 - 0.5709 = 0.5091 kg of mass flows into tank B. Then,

$$m_{2,B} = m_{1,B} + 0.5091 = 2 + 0.5091 = 2.509 \text{ kg}$$

The final specific volume of steam in tank B is determined from

$$v_{2,B} = \frac{V_B}{m_{2,B}} = \frac{(m_1 v_1)_B}{m_{2,B}} = \frac{(2 \text{ kg})(1.1989 \text{ m}^3/\text{kg})}{2.509 \text{ kg}} = 0.9558 \text{ m}^3/\text{kg}$$

We take the entire contents of both tanks as the system, which is a closed system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = (\Delta U)_A + (\Delta U)_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$-Q_{\text{out}} = (m_2 u_2 - m_1 u_1)_A + (m_2 u_2 - m_1 u_1)_B$$

Substituting,

$$-300 = \{(0.5709)(1704.7) - (1.080)(1773.6)\} + \{(2.509) u_{2,B} - (2)(2731.4)\}$$

$$u_{2,B} = 2433.3 \text{ kJ/kg}$$

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Thus,

$$\left.\begin{array}{l} v_{2,B} = 0.9558 \text{ m}^3/\text{kg} \\ u_{2,B} = 2433.3 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_{2,B} = \mathbf{116.1°C} \\ s_{2,B} = 6.9156 \text{ kJ/kg} \cdot \text{K} \end{array}$$

(*b*) The total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes both tanks and their immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,surr}}} + S_{\text{gen}} = \Delta S_{\text{A}} + \Delta S_{\text{B}}$$

Rearranging and substituting, the total entropy generated during this process is determined to be

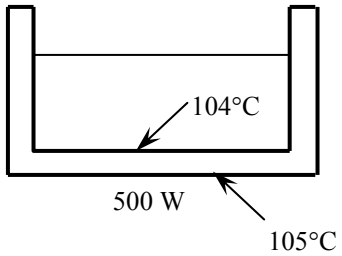
$$\begin{aligned} S_{\text{gen}} &= \Delta S_A + \Delta S_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}} = \left(m_2 s_2 - m_1 s_1\right)_A + \left(m_2 s_2 - m_1 s_1\right)_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}} \\ &= \{(0.5709)(4.8479) - (1.080)(4.8479)\} + \{(2.509)(6.9156) - (2)(7.7100)\} + \frac{300 \text{ kJ}}{290 \text{ K}} \\ &= \mathbf{0.498 \text{ kJ/K}} \end{aligned}$$

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7-213 Heat is transferred steadily to boiling water in a pan through its bottom. The rate of entropy generation within the bottom plate is to be determined.

Assumptions Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values.

Analysis We take the bottom of the pan to be the system,



which is a closed system. Under steady conditions, the rate form of the entropy balance for this system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,system}} = 0$$

$$\frac{500\ \text{W}}{378\ \text{K}} - \frac{500\ \text{W}}{377\ \text{K}} + \dot{S}_{\text{gen,system}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,system}} = \mathbf{0.00351\ W/K}$$

Discussion Note that there is a small temperature drop across the bottom of the pan, and thus a small amount of entropy generation.

7-214 An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature and the entropy generated during this process are to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible.

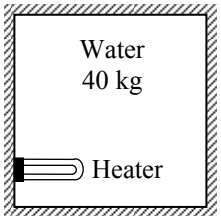
Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis Taking the water in the container as the system, which is a closed system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = (\Delta U)_{\text{water}}$$

$$\dot{W}_{\text{e,in}} \Delta t = mc(T_2 - T_1)_{\text{water}}$$



Substituting,

$$(1200\ \text{J/s})\Delta t = (40\ \text{kg})(4180\ \text{J/kg·°C})(50 - 20)\text{°C}$$

Solving for Δt gives

$$\Delta t = \mathbf{4180\ s = 69.7\ min = 1.16\ h}$$

Again we take the water in the tank to be the system. Noting that no heat or mass crosses the boundaries of this system and the energy and entropy contents of the heater are negligible, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{water}}$$

Therefore, the entropy generated during this process is

$$S_{\text{gen}} = \Delta S_{\text{water}} = mc \ln \frac{T_2}{T_1} = (40\ \text{kg})(4.18\ \text{kJ/kg} \cdot \text{K}) \ln \frac{323\ \text{K}}{293\ \text{K}} = \mathbf{16.3\ kJ/K}$$

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7-215 The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted steam to the feedwater and entropy generation per unit mass of feedwater are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the device to the surroundings is negligible.

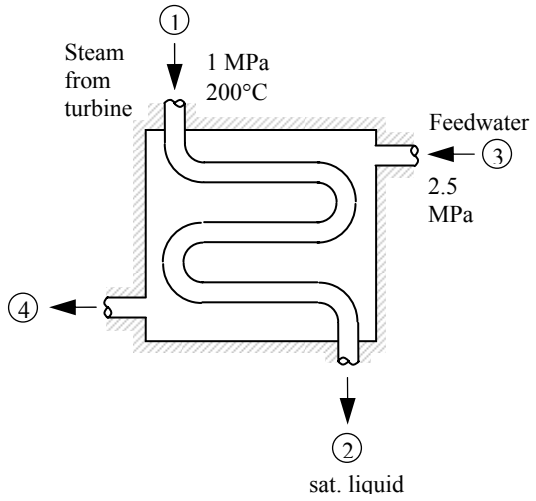
Properties The properties of steam and feedwater are (Tables A-4 through A-6)

$$\left.\begin{matrix} P_1 = 1\ \text{MPa} \\ T_1 = 200°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 2828.3\ \text{kJ/kg} \\ s_1 = 6.6956\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 1\ \text{MPa} \\ \text{sat. liquid} \end{matrix}\right\} \begin{matrix} h_2 = h_{f@1\ \text{MPa}} = 762.51\ \text{kJ/kg} \\ s_2 = s_{f@1\ \text{MPa}} = 2.1381\ \text{kJ/kg}\cdot\text{K} \\ T_2 = 179.88°\text{C} \end{matrix}$$

$$\left.\begin{matrix} P_3 = 2.5\ \text{MPa} \\ T_3 = 50°\text{C} \end{matrix}\right\} \begin{matrix} h_3 \cong h_{f@50°\text{C}} = 209.34\ \text{kJ/kg} \\ s_3 \cong s_{f@50°\text{C}} = 0.7038\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 2.5\ \text{MPa} \\ T_4 = T_2 - 10°\text{C} \cong 170°\text{C} \end{matrix}\right\} \begin{matrix} h_4 \cong h_{f@170°\text{C}} = 719.08\ \text{kJ/kg} \\ s_4 \cong s_{f@170°\text{C}} = 2.0417\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$



Analysis (*a*) We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_s \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_{fw}$$

Energy balance (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_s (h_2 - h_1) = \dot{m}_{fw} (h_3 - h_4)$

Dividing by $\dot{m}_{fw}$ and substituting,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_4 - h_3}{h_1 - h_2} = \frac{(719.08 - 209.34)\ \text{kJ/kg}}{(2828.3 - 762.51)\ \text{kJ/kg}} = \mathbf{0.247}$$

(*b*) The total entropy change (or entropy generation) during this process per unit mass of feedwater can be determined from an entropy balance expressed in the rate form as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\swarrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{m}_s (s_1 - s_2) + \dot{m}_{fw} (s_3 - s_4) + \dot{S}_{\text{gen}} = 0$$

$$\frac{\dot{S}_{\text{gen}}}{\dot{m}_{fw}} = \frac{\dot{m}_s}{\dot{m}_{fw}} (s_2 - s_1) + (s_4 - s_3) = (0.247)(2.1381 - 6.6956) + (2.0417 - 0.7038)$$

$$= \mathbf{0.213\ kJ/K}\ \text{per kg of feedwater}$$

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7-216 Problem 7-215 is reconsidered. The effect of the state of the steam at the inlet to the feedwater heater is to be investigated. The entropy of the extraction steam is assumed to be constant at the value for 1 MPa, 200°C, and the extraction steam pressure is to be varied from 1 MPa to 100 kPa. Both the ratio of the mass flow rates of the extracted steam and the feedwater heater and the total entropy change for this process per unit mass of the feedwater are to be plotted as functions of the extraction pressure.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
WorkFluid$ = 'Steam_iapws'
"P[3] = 1000 [kPa]"  "place {} around P[3] and T[3] eqations to solve the table"
T[3] = 200 [C]
P[4] = P[3]
x[4]=0
T[4]=temperature(WorkFluid$,P=P[4],x=x[4])
P[1] = 2500 [kPa]
T[1] = 50 [C]
P[2] = 2500  [kPa]
T[2] = T[4] - 10"[C]"

"Since we don't know the mass flow rates and we want to determine the ratio of mass flow rate of the extracted steam and the feedwater, we can assume the mass flow rate of the feedwater is 1 kg/s without loss of generality. We write the conservation of energy."
"Conservation of mass for the steam extracted from the turbine: "
m_dot_steam[3]= m_dot_steam[4]

"Conservation of mass for the condensate flowing through the feedwater heater:"
m_dot_fw[1] = 1
m_dot_fw[2]= m_dot_fw[1]

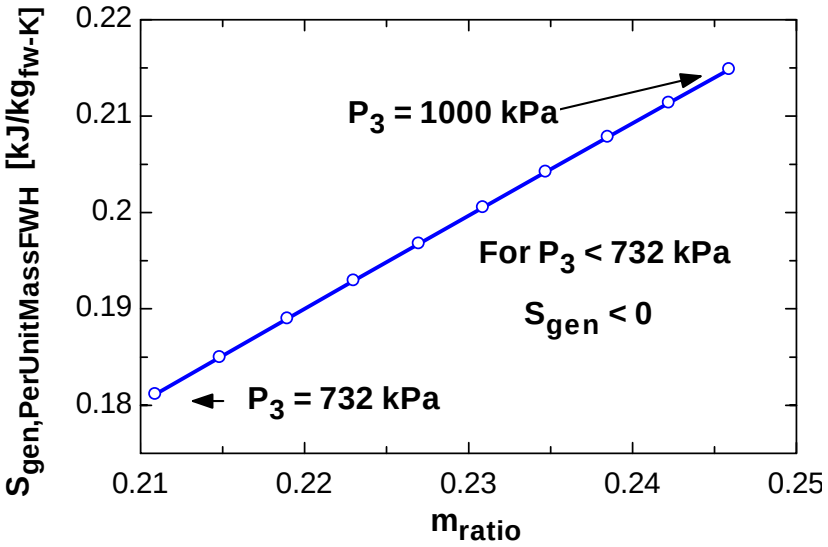
"Conservation of Energy - SSSF energy balance for the feedwater heater -- neglecting the change in potential energy, no  heat transfer, no work:"
h[3]=enthalpy(WorkFluid$,P=P[3],T=T[3])
"To solve the table, place {} around s[3] and remove them from the 2nd and 3rd equations"
s[3]=entropy(WorkFluid$,P=P[3],T=T[3])
{s[3] =6.693 [kJ/kg-K]  "This s[3] is  for the initial T[3], P[3]"
T[3]=temperature(WorkFluid$,P=P[3],s=s[3])  "Use this equation for T[3] only when s[3] is given."}
h[4]=enthalpy(WorkFluid$,P=P[4],x=x[4])
s[4]=entropy(WorkFluid$,P=P[4],x=x[4])
h[1]=enthalpy(WorkFluid$,P=P[1],T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
h[2]=enthalpy(WorkFluid$,P=P[2],T=T[2])
s[2]=entropy(WorkFluid$,P=P[2],T=T[2])

"For the feedwater heater:"
E_dot_in = E_dot_out
E_dot_in = m_dot_steam[3]*h[3] +m_dot_fw[1]*h[1]
E_dot_out= m_dot_steam[4]*h[4] + m_dot_fw[2]*h[2]
m_ratio = m_dot_steam[3]/ m_dot_fw[1]

"Second Law analysis:"
S_dot_in - S_dot_out + S_dot_gen = DELTAS_dot_sys
DELTAS_dot_sys = 0 "[KW/K]" "steady-flow result"
S_dot_in = m_dot_steam[3]*s[3] +m_dot_fw[1]*s[1]
S_dot_out= m_dot_steam[4]*s[4] + m_dot_fw[2]*s[2]
S_gen_PerUnitMassFWH = S_dot_gen/m_dot_fw[1]"[kJ/kg_fw-K]"
```

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m_{ratio}	$S_{gen,PerUnitMass}$ [kJ/kg-K]	P_3 [kPa]
0.2109	0.1811	732
0.2148	0.185	760
0.219	0.189	790
0.223	0.1929	820
0.227	0.1968	850
0.2309	0.2005	880
0.2347	0.2042	910
0.2385	0.2078	940
0.2422	0.2114	970
0.2459	0.2149	1000



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7-217E A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and is charged until the tank contains saturated liquid at a specified pressure. The mass of R-134a that entered the tank, the heat transfer with the surroundings at 100°F, and the entropy generated during this process are to be determined.

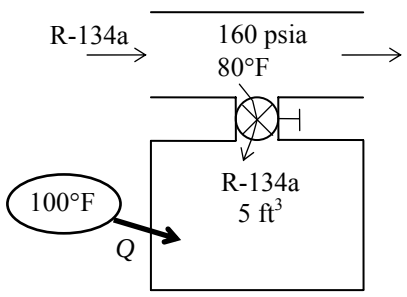
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 80 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@80\text{ psia}} = 0.59750 \text{ ft}^3/\text{lbm} \\ u_1 = u_{g@80\text{ psia}} = 103.35 \text{ Btu/lbm} \\ s_1 = s_{g@80\text{ psia}} = 0.22040 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_2 = v_{f@120\text{ psia}} = 0.01360 \text{ ft}^3/\text{lbm} \\ u_2 = u_{f@120\text{ psia}} = 41.49 \text{ Btu/lbm} \\ s_2 = s_{f@120\text{ psia}} = 0.08589 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_i = 160 \text{ psia} \\ T_i = 80°\text{F} \end{array}\right\} \begin{array}{l} h_i \cong h_{f@80°\text{F}} = 38.17 \text{ Btu/lbm} \\ s_i \cong s_{f@80°\text{F}} = 0.07934 \text{ Btu/lbm}\cdot\text{R} \end{array}$$



Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{5 \text{ ft}^3}{0.59750 \text{ ft}^3/\text{lbm}} = 8.368 \text{ lbm}$$

$$m_2 = \frac{V}{v_2} = \frac{5 \text{ ft}^3}{0.01360 \text{ ft}^3/\text{lbm}} = 367.58 \text{ lbm}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = 367.58 - 8.368 = \mathbf{359.2 \text{ lbm}}$$

(*b*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(359.2 \text{ lbm})(38.17 \text{ Btu/lbm}) + (367.58 \text{ lbm})(41.49 \text{ Btu/lbm}) - (8.368 \text{ lbm})(103.35 \text{ Btu/lbm}) \\ &= \mathbf{673 \text{ Btu}} \end{aligned}$$

(*c*) The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the tank and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}} \longrightarrow \frac{Q_{\text{in}}}{T_{\text{b,in}}} + m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1$$

Therefore, the total entropy generated during this process is

$$\begin{aligned} S_{\text{gen}} &= -m_i s_i + (m_2 s_2 - m_1 s_1) - \frac{Q_{\text{in}}}{T_{\text{b,in}}} \\ &= -(359.2)(0.07934) + (367.58)(0.08589) - (8.368)(0.22040) - \frac{673 \text{ Btu}}{560 \text{ R}} = \mathbf{0.0264 \text{ Btu/R}} \end{aligned}$$

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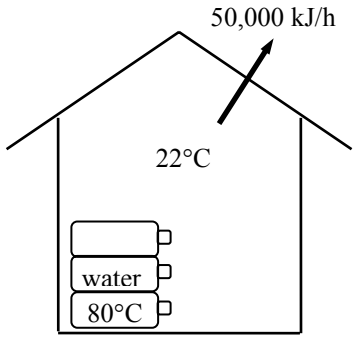
7-218 The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night and the amount of entropy generated that night are to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times.

Properties The density and specific heat of water at room temperature are $\rho = 997$ kg/m^3 and $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis The total mass of water is

$$m_w = \rho V = (0.997 \text{ kg/L})(50 \times 20 \text{ L}) = 997 \text{ kg}$$



Taking the contents of the house, including the water as our system, the energy balance relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$\begin{aligned} W_{\text{e,in}} - Q_{\text{out}} &= \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}} \\ &= (\Delta U)_{\text{water}} \\ &= mc(T_2 - T_1)_{\text{water}} \end{aligned}$$

or,

$$\dot{W}_{\text{e,in}} \Delta t - Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$(15 \text{ kJ/s})\Delta t - (50{,}000 \text{ kJ/h})(10 \text{ h}) = (997 \text{ kg})(4.18 \text{ kJ/kg·°C})(22 - 80)°\text{C}$$

It gives

$$\Delta t = 17{,}219 \text{ s} = \mathbf{4.78\ h}$$

We take the house as the system, which is a closed system. The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the house and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for the extended system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{air}} \overset{\nearrow 0}{} = \Delta S_{\text{water}}$$

since the state of air in the house remains unchanged. Then the entropy generated during the 10-h period that night is

$$\begin{aligned} S_{\text{gen}} &= \Delta S_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \left(mc \ln \frac{T_2}{T_1} \right)_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{surr}}} \\ &= (997 \text{ kg})(4.18 \text{ kJ/kg·K}) \ln \frac{295 \text{ K}}{353 \text{ K}} + \frac{500{,}000 \text{ kJ}}{276 \text{ K}} \\ &= -748 + 1811 = \mathbf{1063\ kJ/K} \end{aligned}$$

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7-219 A room is to be heated by hot water contained in a tank placed in the room. The minimum initial temperature of the water needed to meet the heating requirements of this room for a 24-h period and the entropy generated are to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 24-h period.

Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis Heat loss from the room during a 24-h period is

$$Q_{\text{loss}} = (10{,}000 \text{ kJ/h})(24 \text{ h}) = 240{,}000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \quad \rightarrow -Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + \cancelto{0}{(\Delta U)_{\text{air}}}$$

or

$$-Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

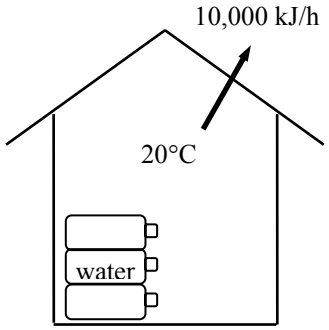
Substituting,

$$-240{,}000 \text{ kJ} = (1500 \text{ kg})(4.18 \text{ kJ/kg·°C})(20 - T_1)$$

It gives

$$T_1 = \mathbf{58.3°C}$$

where T_1 is the temperature of the water when it is first brought into the room.



(*b*) We take the house as the system, which is a closed system. The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the house and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for the extended system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{water}} + \cancelto{0}{\Delta S_{\text{air}}} = \Delta S_{\text{water}}$$

since the state of air in the house (and thus its entropy) remains unchanged. Then the entropy generated during the 24 h period becomes

$$\begin{aligned} S_{\text{gen}} &= \Delta S_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \left(mc \ln \frac{T_2}{T_1} \right)_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{surr}}} \\ &= (1500 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{293 \text{ K}}{331.3 \text{ K}} + \frac{240{,}000 \text{ kJ}}{278 \text{ K}} \\ &= -770.3 + 863.3 = \mathbf{93.0 \text{ kJ/K}} \end{aligned}$$

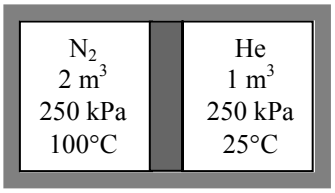
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7-220 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the entropy generated are to be determined for the cases of the piston being fixed and moving freely.

Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

Properties The gas constants and the constant volume specific heats are $R = 0.2968$ kPa.m^3/kg.K, $c_v = 0.743$ kJ/kg·°C and $c_p = 1.039$ kJ/kg·°C for N_2, and $R = 2.0769$ kPa.m^3/kg.K, $c_v = 3.1156$ kJ/kg·°C, and $c_p = 5.1926$ kJ/kg·°C for He (Tables A-1 and A-2)

Analysis The mass of each gas in the cylinder is



$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(250\ \text{kPa})(2\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(373\ \text{K})} = 4.516\ \text{kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(250\ \text{kPa})(1\ \text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\ \text{K})} = 0.4039\ \text{kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{\text{He}} \longrightarrow 0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{\text{He}}$$

Substituting, $$(4.516\ \text{kg})(0.743\ \text{kJ/kg}\cdot\text{°C})(T_f - 100)\text{°C} + (0.4039\ \text{kg})(3.1156\ \text{kJ/kg}\cdot\text{°C})(T_f - 25)\text{°C} = 0$$

It gives

T_f = **79.5°C**

where T_f is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

(*b*) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{N_2} + \Delta S_{\text{He}}$$

But first we determine the final pressure in the cylinder:

$$N_{\text{total}} = N_{N_2} + N_{\text{He}} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{\text{He}} = \frac{4.516\ \text{kg}}{28\ \text{kg/kmol}} + \frac{0.4039\ \text{kg}}{4\ \text{kg/kmol}} = 0.2623\ \text{kmol}$$

$$P_2 = \frac{N_{\text{total}} R_u T}{V_{\text{total}}} = \frac{(0.2623\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(352.5\ \text{K})}{3\ \text{m}^3} = 256.3\ \text{kPa}$$

Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$

$$= (4.516\ \text{kg})\left[(1.039\ \text{kJ/kg}\cdot\text{K})\ln\frac{352.5\ \text{K}}{373\ \text{K}} - (0.2968\ \text{kJ/kg}\cdot\text{K})\ln\frac{256.3\ \text{kPa}}{250\ \text{kPa}}\right]$$

$$= -0.2978\ \text{kJ/K}$$

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$$\Delta S_{\text{He}} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \right)_{\text{He}}$$

$$= (0.4039\ \text{kg})\left[(5.1926\ \text{kJ/kg}\cdot\text{K})\ln\frac{352.5\ \text{K}}{298\ \text{K}} - (2.0769\ \text{kJ/kg}\cdot\text{K})\ln\frac{256.3\ \text{kPa}}{250\ \text{kPa}} \right]$$

$$= 0.3318\ \text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{N}_2} + \Delta S_{\text{He}} = -0.2978 + 0.3318 = \mathbf{0.0340\ kJ/K}$$

If the piston were not free to move, we would still have T_2 = 352.5 K but the volume of each gas would remain constant in this case:

$$\Delta S_{\text{N}_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0} \right)_{\text{N}_2} = (4.516\ \text{kg})(0.743\ \text{kJ/kg}\cdot\text{K})\ln\frac{352.5\ \text{K}}{373\ \text{K}} = -0.1893\ \text{kJ/K}$$

$$\Delta S_{\text{He}} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0} \right)_{\text{He}} = (0.4039\ \text{kg})(3.1156\ \text{kJ/kg}\cdot\text{K})\ln\frac{352.5\ \text{K}}{298\ \text{K}} = 0.2115\ \text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{N}_2} + \Delta S_{\text{He}} = -0.1893 + 0.2115 = \mathbf{0.0223\ kJ/K}$$

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7-221 Problem 7-220 is reconsidered. The results for constant specific heats to those obtained using variable specific heats are to be compared using built-in EES or other functions.

Analysis The problem is solved using EES, and the results are given below.

```
"Knowns:"
R_u=8.314 [kJ/kmol-K]
V_N2[1]=2 [m^3]
Cv_N2=0.743 [kJ/kg-K]  "From Table A-2(a) at 27C"
R_N2=0.2968 [kJ/kg-K]  "From Table A-2(a)"
T_N2[1]=100 [C]
P_N2[1]=250 [kPa]
Cp_N2=R_N2+Cv_N2
V_He[1]=1 [m^3]
Cv_He=3.1156 [kJ/kg-K] "From Table A-2(a) at 27C"
T_He[1]=25 [C]
P_He[1]=250 [kPa]
R_He=2.0769 [kJ/kg-K]  "From Table A-2(a)"
Cp_He=R_He+Cv_He

"Solution:"

"mass calculations:"
P_N2[1]*V_N2[1]=m_N2*R_N2*(T_N2[1]+273)
P_He[1]*V_He[1]=m_He*R_He*(T_He[1]+273)

"The entire cylinder is considered to be a closed system, allowing the piston to move."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE
E_in =0 [kJ]
E_out = 0 [kJ]

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE= m_N2*Cv_N2*(T_2-T_N2[1])+m_He*Cv_He*(T_2-T_He[1])

"Total volume of gases:"
V_total=V_N2[1]+V_He[1]
MM_He = 4 [kg/kmol]
MM_N2 = 28 [kg/kmol]

N_total = m_He/MM_He+m_N2/MM_N2
"Final pressure at equilibrium:"
"Allowing the piston to move, the pressure on both sides is the same, P_2 is:"
P_2*V_total=N_total*R_u*(T_2+273)

S_gen_PistonMoving = DELTAS_He_PM+DELTAS_N2_PM
DELTAS_He_PM=m_He*(Cp_He*ln((T_2+273)/(T_He[1]+273))-R_He*ln(P_2/P_He[1]))
DELTAS_N2_PM=m_N2*(Cp_N2*ln((T_2+273)/(T_N2[1]+273))-R_N2*ln(P_2/P_N2[1]))

"The final temperature of the system when the piston does not move will be the same as when it does move.
The volume of the gases remain constant and the entropy changes are given by:"

S_gen_PistNotMoving = DELTAS_He_PNM+DELTAS_N2_PNM
DELTAS_He_PNM=m_He*(Cv_He*ln((T_2+273)/(T_He[1]+273)))
DELTAS_N2_PNM=m_N2*(Cv_N2*ln((T_2+273)/(T_N2[1]+273)))

"The following uses the EES functions for the nitrogen.  Since helium is monatomic, we use the constant specific
heat approach to find its property changes."
```

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E_in - E_out = DELTAE_VP
DELTAE_VP= m_N2*(INTENERGY(N2,T=T_2_VP)-INTENERGY(N2,T=T_N2[1]))+m_He*Cv_He*(T_2_VP-T_He[1])

"Final Pressure for moving piston:"
P_2_VP*V_total=N_total*R_u*(T_2_VP+273)
S_gen_PistMoving_VP = DELTAS_He_PM_VP+DELTAS_N2_PM_VP
DELTAS_N2_PM_VP=m_N2*(ENTROPY(N2,T=T_2_VP,P=P_2_VP)-ENTROPY(N2,T=T_N2[1],P=P_N2[1]))
DELTAS_He_PM_VP=m_He*(Cp_He*ln((T_2+273)/(T_He[1]+273))-R_He*ln(P_2/P_He[1]))

"Fianl N2 Pressure for piston not moving."
P_2_N2_VP*V_N2[1]=m_N2*R_N2*(T_2_VP+273)

S_gen_PistNotMoving_VP = DELTAS_He_PNM_VP+DELTAS_N2_PNM_VP
DELTAS_N2_PNM_VP = m_N2*(ENTROPY(N2,T=T_2_VP,P=P_2_N2_VP)-ENTROPY(N2,T=T_N2[1],P=P_N2[1]))
DELTAS_He_PNM_VP=m_He*(Cv_He*ln((T_2_VP+273)/(T_He[1]+273)))

SOLUTION

Cp_He=5.193 [kJ/kg-K]
Cp_N2=1.04 [kJ/kg-K]
Cv_He=3.116 [kJ/kg-K]
Cv_N2=0.743 [kJ/kg-K]
DELTAE=0 [kJ]
DELTAE_VP=0 [kJ]
DELTAS_He_PM=0.3318 [kJ/K]
DELTAS_He_PM_VP=0.3318 [kJ/K]
DELTAS_He_PNM=0.2115 [kJ/K]
DELTAS_He_PNM_VP=0.2116 [kJ/K]
DELTAS_N2_PM=-0.298 [kJ/K]
DELTAS_N2_PM_VP=-0.298 [kJ/K]
DELTAS_N2_PNM=-0.1893 [kJ/K]
DELTAS_N2_PNM_VP=-0.1893 [kJ/K]
E_in=0 [kJ]
E_out=0 [kJ]
MM_He=4 [kg/kmol]
MM_N2=28 [kg/kmol]
m_He=0.4039 [kg]
m_N2=4.516 [kg]
N_total=0.2623 [kmol]
P_2=256.3 [kPa]
P_2_N2_VP=236.3 [kPa]
P_2_VP=256.3 [kPa]
R_He=2.077 [kJ/kg-K]
R_N2=0.2968 [kJ/kg-K]
R_u=8.314 [kJ/kmol-K]
S_gen_PistMoving_VP=0.0338 [kJ/K]
S_gen_PistNotMoving=0.02226 [kJ/K]
S_gen_PistNotMoving_VP=0.02238 [kJ/K]
S_gen_PistonMoving=0.03379 [kJ/K]
T_2=79.54 [C]
T_2_VP=79.58 [C]
V_total=3 [m^3]

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7-222 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the entropy generated are to be determined for the cases of the piston being fixed and moving freely.

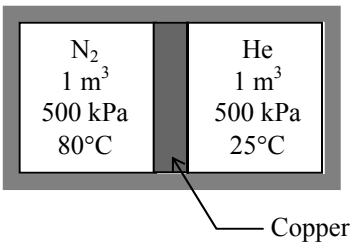
Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

Properties The gas constants and the constant volume specific heats are $R = 0.2968$ kPa.m^3/kg.K, $c_v = 0.743$ kJ/kg·°C and $c_p = 1.039$ kJ/kg·°C for N_2, and $R = 2.0769$ kPa.m^3/kg.K, $c_v = 3.1156$ kJ/kg·°C, and $c_p = 5.1926$ kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of the copper at room temperature is $c = 0.386$ kJ/kg·°C (Table A-3).

Analysis The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\ \text{K})} = 4.77\ \text{kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\ \text{kPa})(1\text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\ \text{K})} = 0.808\ \text{kg}$$



Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He} + [mc(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\,Cu} = (80 + 25)/2 = 52.5°\text{C}$$

Substituting,

$$(4.77\ \text{kg})(0.743\ \text{kJ/kg}\cdot°\text{C})(T_f - 80)°\text{C} + (0.808\ \text{kg})(3.1156\ \text{kJ/kg}\cdot°\text{C})(T_f - 25)°\text{C} + (5.0\ \text{kg})(0.386\ \text{kJ/kg}\cdot°\text{C})(T_f - 52.5)°\text{C} = 0$$

It gives

$$T_f = \mathbf{56.0°C}$$

where T_f is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

(*b*) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston}$$

But first we determine the final pressure in the cylinder:

$$N_{total} = N_{N_2} + N_{He} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{He} = \frac{4.77\ \text{kg}}{28\ \text{kg/kmol}} + \frac{0.808\ \text{kg}}{4\ \text{kg/kmol}} = 0.3724\ \text{kmol}$$

$$P_2 = \frac{N_{total} R_u T}{V_{total}} = \frac{(0.3724\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(329\ \text{K})}{2\ \text{m}^3} = 509.4\ \text{kPa}$$

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Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$
$$= (4.77\ \text{kg})\left[(1.039\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} - (0.2968\ \text{kJ/kg}\cdot\text{K})\ln\frac{509.4\ \text{kPa}}{500\ \text{kPa}}\right]$$
$$= -0.3749\ \text{kJ/K}$$
$$\Delta S_{He} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{He}$$
$$= (0.808\ \text{kg})\left[(5.1926\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{298\ \text{K}} - (2.0769\ \text{kJ/kg}\cdot\text{K})\ln\frac{509.4\ \text{kPa}}{500\ \text{kPa}}\right]$$
$$= 0.3845\ \text{kJ/K}$$
$$\Delta S_{piston} = \left(mc\ln\frac{T_2}{T_1}\right)_{piston} = (5\ \text{kg})(0.386\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{325.5\ \text{K}} = 0.021\ \text{kJ/K}$$
$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston} = -0.3749 + 0.3845 + 0.021 = \mathbf{0.0305\ kJ/K}$$

If the piston were not free to move, we would still have T_2 = 329 K but the volume of each gas would remain constant in this case:

$$\Delta S_{N_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{\cancel{V_2}^{\,\not 0}}{V_1}\right)_{N_2} = (4.77\ \text{kg})(0.743\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} = -0.249\ \text{kJ/K}$$
$$\Delta S_{He} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{\cancel{V_2}^{\,\not 0}}{V_1}\right)_{He} = (0.808\ \text{kg})(3.1156\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{298\ \text{K}} = 0.249\ \text{kJ/K}$$

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston} = -0.249 + 0.249 + 0.021 = \mathbf{0.021\ kJ/K}$$

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7-223 An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to a specified value. The amount of electrical work done during this process and the total entropy change are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

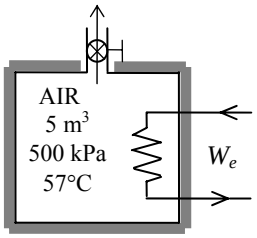
Properties The gas constant is $R = 0.287$ kPa.m^3/kg.K (Table A-1). The properties of air are (Table A-17)

$$T_e = 330 \text{ K} \longrightarrow h_e = 330.34 \text{ kJ/kg}$$
$$T_1 = 330 \text{ K} \longrightarrow u_1 = 235.61 \text{ kJ/kg}$$
$$T_2 = 330 \text{ K} \longrightarrow u_2 = 235.61 \text{ kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \text{ (since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(500 \text{ kPa})(5 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(330 \text{ K})} = 26.40 \text{ kg}$$
$$m_2 = \frac{P_2 V}{RT_2} = \frac{(200 \text{ kPa})(5 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(330 \text{ K})} = 10.56 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 26.40 - 10.56 = 15.84 \text{ kg}$$

$$W_{\text{e,in}} = m_e h_e + m_2 u_2 - m_1 u_1$$
$$= (15.84 \text{ kg})(330.34 \text{ kJ/kg}) + (10.56 \text{ kg})(235.61 \text{ kJ/kg}) - (26.40 \text{ kg})(235.61 \text{ kJ/kg}) = \mathbf{1501 \text{ kJ}}$$

(*b*) The total entropy change, or the total entropy generation within the tank boundaries is determined from an entropy balance on the tank expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}}$$

or,

$$S_{\text{gen}} = m_e s_e + \Delta S_{\text{tank}} = m_e s_e + (m_2 s_2 - m_1 s_1)$$
$$= (m_1 - m_2) s_e + (m_2 s_2 - m_1 s_1) = m_2 (s_2 - s_e) - m_1 (s_1 - s_e)$$

Assuming a constant average pressure of (500 + 200)/2 = 350 kPa for the exit stream, the entropy changes are determined to be

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$$s_2 - s_e = c_p \ln \frac{T_2}{T_e}^{\nearrow 0} - R \ln \frac{P_2}{P_e} = -R \ln \frac{P_2}{P_e} = -(0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{200 \text{ kPa}}{350 \text{ kPa}} = 0.1606 \text{ kJ/kg} \cdot \text{K}$$

$$s_1 - s_e = c_p \ln \frac{T_1}{T_e}^{\nearrow 0} - R \ln \frac{P_2}{P_e} = -R \ln \frac{P_1}{P_e} = -(0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{500 \text{ kPa}}{350 \text{ kPa}} = -0.1024 \text{ kJ/kg} \cdot \text{K}$$

Therefore, the total entropy generated within the tank during this process is

$$S_{\text{gen}} = (10.56 \text{ kg})(0.1606 \text{ kJ/kg} \cdot \text{K}) - (26.40 \text{ kg})(-0.1024 \text{ kJ/kg} \cdot \text{K}) = \mathbf{4.40 \ kJ/K}$$

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7-224 An insulated cylinder initially contains a saturated liquid-vapor mixture of water at a specified temperature. The entire vapor in the cylinder is to be condensed isothermally by adding ice inside the cylinder. The amount of ice added and the entropy generation are to be determined.

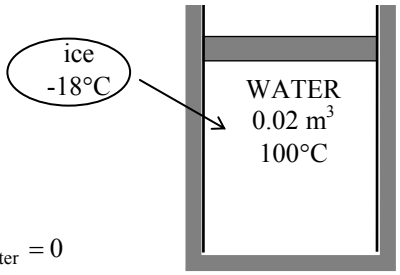
Assumptions **1** Thermal properties of the ice are constant. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

Properties The specific heat of ice at about 0°C is c = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg.

Analysis (*a*) We take the contents of the cylinder (ice and saturated water) as our system, which is a closed system. Noting that the temperature and thus the pressure remains constant during this phase change process and thus $W_b + \Delta U = \Delta H$, the energy balance for this system can be written as

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{b,in} = \Delta U$$
$$\Delta H = 0$$
$$\Delta H_{\text{ice}} + \Delta H_{\text{water}} = 0$$



or

$$[mc(0°\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0°\text{C})_{\text{liquid}}]_{\text{ice}} + [m(h_2 - h_1)]_{\text{water}} = 0$$

The properties of water at 100°C are (Table A-4)

$$v_f = 0.001043, \quad v_g = 1.6720 \text{ m}^3/\text{kg}$$
$$h_f = 419.17, \quad h_{fg} = 2256.4 \text{ kJ.kg}$$
$$s_f = 1.3072 \qquad s_{fg} = 6.0490 \text{ kJ/kg.K}$$

$$v_1 = v_f + x_1 v_{fg} = 0.001043 + (0.1)(1.6720 - 0.001043) = 0.16814 \text{ m}^3/\text{kg}$$
$$h_1 = h_f + x_1 h_{fg} = 419.17 + (0.1)(2256.4) = 644.81 \text{ kJ/kg}$$
$$s_1 = s_f + x_1 s_{fg} = 1.3072 + (0.1)(6.0470) = 1.9119 \text{ kJ/kg} \cdot \text{K}$$
$$h_2 = h_{f@100°\text{C}} = 419.17 \text{ kJ/kg}$$
$$s_2 = s_{f@100°\text{C}} = 1.3072 \text{ kJ/kg} \cdot \text{K}$$
$$m_{\text{steam}} = \frac{V_1}{v_1} = \frac{0.02 \text{ m}^3}{0.16814 \text{ m}^3/\text{kg}} = 0.119 \text{ kg}$$

Noting that $T_{1,\text{ ice}}$ = -18°C and T_2 = 100°C and substituting gives

$$m\{(2.11 \text{ kJ/kg.K})[0-(-18)] + 333.7 \text{ kJ/kg} + (4.18 \text{ kJ/kg·°C})(100-0)°\text{C}\}$$
$$+(0.119 \text{ kg})(419.17 - 644.81) \text{ kJ/kg} = 0$$

m = 0.034 kg = **34.0 g ice**

(*b*) We take the ice and the steam as our system, which is a closed system. Considering that the tank is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\text{Net entropy transfer by heat and mass}} + \underbrace{S_{\text{gen}}}_{\text{Entropy generation}} = \underbrace{\Delta S_{\text{system}}}_{\text{Change in entropy}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{ice}} + \Delta S_{\text{steam}}$$

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$$\Delta S_{\text{steam}} = m(s_2 - s_1) = (0.119\ \text{kg})(1.3072 - 1.9119)\text{kJ/kg}\cdot\text{K} = -0.0719\ \text{kJ/K}$$

$$\Delta S_{\text{ice}} = (\Delta S_{\text{solid}} + \Delta S_{\text{melting}} + \Delta S_{\text{liquid}})_{\text{ice}} = \left(\left(mc \ln \frac{T_{\text{melting}}}{T_1} \right)_{\text{solid}} + \frac{mh_{if}}{T_{\text{melting}}} + \left(mc \ln \frac{T_2}{T_1} \right)_{\text{liquid}} \right)_{\text{ice}}$$

$$= (0.034\ \text{kg}) \left((2.11\ \text{kJ/kg}\cdot\text{K}) \ln \frac{273.15\ \text{K}}{255.15\ \text{K}} + \frac{333.7\ \text{kJ/kg}}{273.15\ \text{K}} + (4.18\ \text{kJ/kg}\cdot\text{K}) \ln \frac{373.15\ \text{K}}{273.15\ \text{K}} \right) = 0.0907\ \text{kJ/K}$$

Then,

$$S_{\text{gen}} = \Delta S_{\text{steam}} + \Delta S_{\text{ice}} = -0.0719 + 0.0907 = \mathbf{0.0188\ kJ/K}$$

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7-225 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established and the amount of entropy generated are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1).

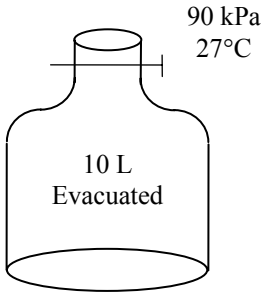
Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2$ (since $m_{\text{out}} = m_{\text{initial}} = 0$)

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \text{ (since } W \cong E_{\text{out}} = E_{\text{initial}} = \text{ke} \cong \text{pe} \cong 0)$$



Combining the two balances:

$$Q_{\text{in}} = m_2(u_2 - h_i)$$

where

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(100 \text{ kPa})(0.010 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})} = 0.01045 \text{ kg}$$

$$T_i = T_2 = 300 \text{ K} \xrightarrow{\text{Table A-17}} \begin{matrix} h_i = 300.19 \text{ kJ/kg} \\ u_2 = 214.07 \text{ kJ/kg} \end{matrix}$$

Substituting,

$$Q_{\text{in}} = (0.01045 \text{ kg})(214.07 - 300.19) \text{ kJ/kg} = -0.90 \text{ kJ} \rightarrow Q_{\text{out}} = \mathbf{0.9 \text{ kJ}}$$

Note that the negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the bottle and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$m_i s_i - \frac{Q_{\text{out}}}{T_{\text{surr}}} + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1 \cancelto{0}{} = m_2 s_2$$

Therefore, the entropy generated during this process is

$$S_{\text{gen}} = -m_i s_i + m_2 s_2 + \frac{Q_{\text{out}}}{T_{\text{surr}}} = m_2 (s_2 - s_i) \cancelto{0}{} + \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{0.9 \text{ kJ}}{300 \text{ K}} = \mathbf{0.0030 \text{ kJ/K}}$$

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7-226 Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. The electric power input to the heater and the rate of entropy generation are to be determined. The reduction in power input and entropy generation as a result of installing a 50% efficient regenerator are also to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** Heat losses from the pipe are negligible.

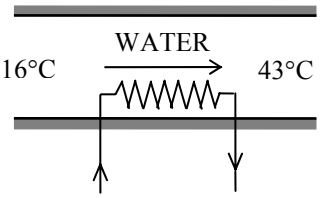
Properties The density of water is given to be ρ = 1 kg/L. The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-3).

Analysis (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \;\rightarrow\; \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}c(T_2 - T_1)$$



where

$$\dot{m} = \rho \dot{V} = (1 \text{ kg/L})(10 \text{ L/min}) = 10 \text{ kg/min}$$

Substituting,

$$\dot{W}_{e,in} = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot °\text{C})(43 - 16)°\text{C} = \mathbf{18.8\ kW}$$

The rate of entropy generation in the heating section during this process is determined by applying the entropy balance on the heating section. Noting that this is a steady-flow process and heat transfer from the heating section is negligible,

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\not\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{gen} = 0 \longrightarrow \dot{S}_{gen} = \dot{m}(s_2 - s_1)$$

Noting that water is an incompressible substance and substituting,

$$\dot{S}_{gen} = \dot{m}c \ln \frac{T_2}{T_1} = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{316 \text{ K}}{289 \text{ K}} = \mathbf{0.0622\ kJ/K}$$

(*b*) The energy recovered by the heat exchanger is

$$\dot{Q}_{saved} = \varepsilon \dot{Q}_{max} = \varepsilon \dot{m} C (T_{max} - T_{min}) = 0.5(10/60 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot °\text{C})(39 - 16)°\text{C} = 8.0 \text{ kJ/s} = 8.0 \text{ kW}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot{W}_{in,new} = \dot{W}_{in,old} - \dot{Q}_{saved} = 18.8 - 8.0 = \mathbf{10.8\ kW}$$

Taking the cold water stream in the heat exchanger as our control volume (a steady-flow system), the temperature at which the cold water leaves the heat exchanger and enters the electric resistance heating section is determined from

$$\dot{Q} = \dot{m}c(T_{c,out} - T_{c,in})$$

Substituting,

$$8 \text{ kJ/s} = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg·}^\circ\text{C})(T_{c,out} - 16^\circ\text{C})$$

It yields

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$$T_{c,out} = 27.5^{\circ}\text{C} = 300.5\text{K}$$

The rate of entropy generation in the heating section in this case is determined similarly to be

$$\dot{S}_{gen} = \dot{m}c\ln\frac{T_2}{T_1} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot\text{K})\ln\frac{316\ \text{K}}{300.5\ \text{K}} = 0.0350\ \text{kJ/K}$$

Thus the reduction in the rate of entropy generation within the heating section is

$$\dot{S}_{reduction} = 0.0622 - 0.0350 = \mathbf{0.0272\ kW/K}$$

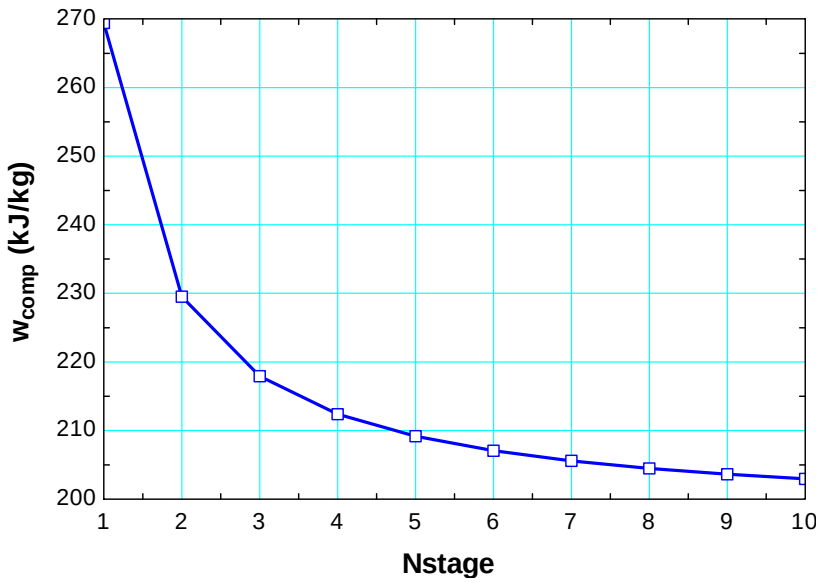
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7-227 Using EES (or other) software, the work input to a multistage compressor is to be determined for a given set of inlet and exit pressures for any number of stages. The pressure ratio across each stage is assumed to be identical and the compression process to be polytropic. The compressor work is to be tabulated and plotted against the number of stages for P_1 = 100 kPa, T_1 = 25°C, P_2 = 1000 kPa, and n = 1.35 for air.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
GAS$ = 'Air'
Nstage = 2  "number of stages of compression with intercooling, each having same pressure ratio."
n=1.35
MM=MOLARMASS(GAS$)
R_u = 8.314 [kJ/kmol-K]
R=R_u/MM
k=1.4
P1=100 [kPa]
T1=25 [C]
P2=1000 [kPa]
R_p = (P2/P1)^(1/Nstage)
W_dot_comp= Nstage*n*R*(T1+273)/(n-1)*((R_p)^((n-1)/n) - 1)
```

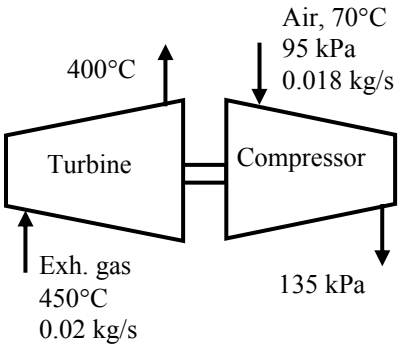
Nstage	w_{comp} [kJ/kg]
1	269.4
2	229.5
3	217.9
4	212.4
5	209.2
6	207.1
7	205.6
8	204.5
9	203.6
10	202.9



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7-228 The turbocharger of an internal combustion engine consisting of a turbine driven by hot exhaust gases and a compressor driven by the turbine is considered. The air temperature at the compressor exit and the isentropic efficiency of the compressor are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Exhaust gases have air properties and air is an ideal gas with constant specific heats.



Properties The specific heat of exhaust gases at the average temperature of 425ºC is c_p = 1.075 kJ/kg.K and properties of air at an anticipated average temperature of 100ºC are c_p = 1.011 kJ/kg.K and k =1.397 (Table A-2).

Analysis (*a*) The turbine power output is determined from

$$\dot{W}_{\rm T} = \dot{m}_{\rm exh} c_p (T_1 - T_2)$$
$$= (0.02\ {\rm kg/s})(1.075\ {\rm kJ/kg.°C})(450 - 400)°{\rm C} = 1.075\ {\rm kW}$$

For a mechanical efficiency of 95% between the turbine and the compressor,

$$\dot{W}_{\rm C} = \eta_m \dot{W}_{\rm T} = (0.95)(1.075\ {\rm kW}) = 1.021\ {\rm kW}$$

Then, the air temperature at the compressor exit becomes

$$\dot{W}_{\rm C} = \dot{m}_{\rm air} c_p (T_2 - T_1)$$
$$1.021\ {\rm kW} = (0.018\ {\rm kg/s})(1.011\ {\rm kJ/kg.°C})(T_2 - 70)°{\rm C}$$
$$T_2 = \mathbf{126.1°C}$$

(*b*) The air temperature at the compressor exit for the case of isentropic process is

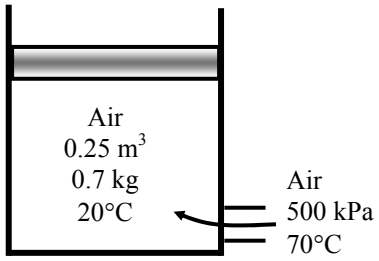
$$T_{2s} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (70 + 273\ {\rm K}) \left(\frac{135\ {\rm kPa}}{95\ {\rm kPa}} \right)^{(1.397-1)/1.397} = 379\ {\rm K} = 106°{\rm C}$$

The isentropic efficiency of the compressor is determined to be

$$\eta_{\rm C} = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{106 - 70}{126.1 - 70} = \mathbf{0.642}$$

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7-229 Air is allowed to enter an insulated piston-cylinder device until the volume of the air increases by 50%. The final temperature in the cylinder, the amount of mass that has entered, the work done, and the entropy generation are to be determined.



Assumptions **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K and the specific heats of air at room temperature are $c_p = 1.005$ kJ/kg.K, $c_v = 0.718$ kJ/kg.K (Table A-2).

Analysis The initial pressure in the cylinder is

$$P_1 = \frac{m_1 R T_1}{V_1} = \frac{(0.7\,\text{kg})(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\,\text{K})}{0.25\,\text{m}^3} = 235.5\,\text{kPa}$$

$$m_2 = \frac{P_2 V_2}{R T_2} = \frac{(235.5\,\text{kPa})(1.5\times 0.25\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{307.71}{T_2}$$

A mass balance on the system gives the expression for the mass entering the cylinder

$$m_i = m_2 - m_1 = \frac{307.71}{T_2} - 0.7$$

(*c*) Noting that the pressure remains constant, the boundary work is determined to be

$$W_{\text{b,out}} = P_1(V_2 - V_1) = (235.5\,\text{kPa})(1.5\times 0.25 - 0.5)\text{m}^3 = \mathbf{29.43\ kJ}$$

(*a*) An energy balance on the system may be used to determine the final temperature

$$m_i h_i - W_{\text{b,out}} = m_2 u_2 - m_1 u_1$$

$$m_i c_p T_i - W_{\text{b,out}} = m_2 c_v T_2 - m_1 c_v T_1$$

$$\left(\frac{307.71}{T_2} - 0.7\right)(1.005)(70+273) - 29.43 = \left(\frac{307.71}{T_2}\right)(0.718)T_2 - (0.7)(0.718)(20+273)$$

There is only one unknown, which is the final temperature. By a trial-error approach or using EES, we find

$$T_2 = \mathbf{308.0\ K}$$

(*b*) The final mass and the amount of mass that has entered are

$$m_2 = \frac{307.71}{308.0} = 0.999\,\text{kg}$$

$$m_i = m_2 - m_1 = 0.999 - 0.7 = \mathbf{0.299\ kg}$$

(*d*) The rate of entropy generation is determined from

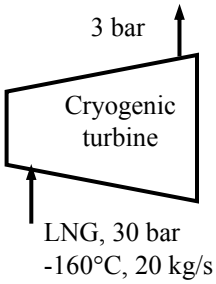
$$\begin{aligned} S_{\text{gen}} &= m_2 s_2 - m_1 s_1 - m_i s_i = m_2 s_2 - m_1 s_1 - (m_2 - m_1)s_i = m_2(s_2 - s_i) - m_1(s_1 - s_i) \\ &= m_2\left(c_p \ln\frac{T_2}{T_i} - R\ln\frac{P_2}{P_i}\right) - m_1\left(c_p \ln\frac{T_1}{T_i} - R\ln\frac{P_1}{P_i}\right) \\ &= (0.999\,\text{kg})\left[(1.005\,\text{kJ/kg.K})\ln\left(\frac{308\,\text{K}}{343\,\text{K}}\right) - (0.287\,\text{kJ/kg.K})\ln\left(\frac{235.5\,\text{kPa}}{500\,\text{kPa}}\right)\right] \\ &\quad - (0.7\,\text{kg})\left[(1.005\,\text{kJ/kg.K})\ln\left(\frac{293\,\text{K}}{343\,\text{K}}\right) - (0.287\,\text{kJ/kg.K})\ln\left(\frac{235.5\,\text{kPa}}{500\,\text{kPa}}\right)\right] \\ &= \mathbf{0.0673\ kJ/K} \end{aligned}$$

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7-230 A cryogenic turbine in a natural gas liquefaction plant produces 115 kW of power. The efficiency of the turbine is to be determined.

Assumptions **1** The turbine operates steadily.

Properties The density of natural gas is given to be 423.8 kg/m^3.



Analysis The maximum possible power that can be obtained from this turbine for the given inlet and exit pressures can be determined from

$$\dot{W}_{\max} = \frac{\dot{m}}{\rho}(P_{\text{in}} - P_{\text{out}}) = \frac{20\ \text{kg/s}}{423.8\ \text{kg/m}^3}(3000 - 300)\text{kPa} = 127.4\ \text{kW}$$

Given the actual power, the efficiency of this cryogenic turbine becomes

$$\eta = \frac{\dot{W}}{\dot{W}_{\max}} = \frac{115\ \text{kW}}{127.4\ \text{kW}} = 0.903 = \mathbf{90.3\%}$$

This efficiency is also known as hydraulic efficiency since the cryogenic turbine handles natural gas in liquid state as the hydraulic turbine handles liquid water.

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7-231 Heat is transferred from a tank to a heat reservoir until the temperature of the tank is reduced to the temperature of reservoir. The expressions for entropy changes are to be obtained and plotted against initial temperature in the tank.

Properties The constant-volume specific heat of air at 300 K is $c_v = 0.718$ kJ/kg·K (Table A-2a)

Analysis The entropy change of air in the tank is

$$\Delta S_{\text{air}} = m\left[c_v \ln\frac{T_2}{T_1} + R\ln\frac{V_2}{V_1}\right] = mc_v \ln\frac{T_2}{T_1}$$

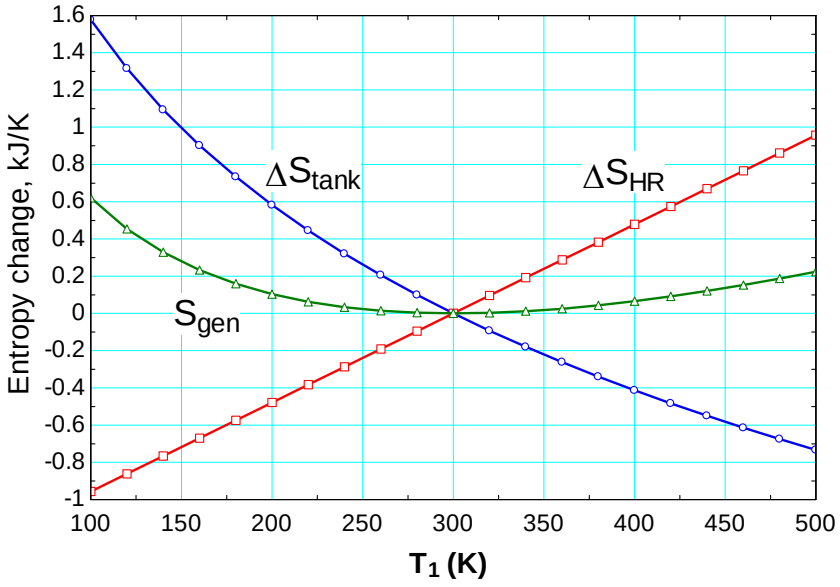
The entropy change of heat reservoir is

$$\Delta S_{\text{HR}} = \frac{Q}{T_L} = \frac{mc_v(T_1 - T_2)}{T_L}$$

The total entropy change (i.e., entropy generation) is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{air}} + \Delta S_{\text{HR}} = mc_v \ln\frac{T_2}{T_1} + \frac{mc_v(T_1 - T_2)}{T_L}$$

The heat transfer will continue until $T_2 = T_L$. Now using $m = 2$ kg, $c_v = 0.718$ kJ/kg·K and $T_L = 300$ K, we plot entropy change terms against initial temperature as shown in the figure.



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7-232 Heat is transferred from a tank to a heat engine which produces work. This work is stored in a work reservoir. The initial temperature of the air for maximum work and thermal efficiency, the total entropy change, work produced and thermal efficiency are to be determined for three initial temperatures.

Properties The constant-volume specific heat of air at 300 K is $c_v = 0.718$ kJ/kg·K (Table A-2a)

Analysis (a) The entropy change of air in the tank is

$$\Delta S_{\text{tank}} = m\left[c_v \ln\frac{T_2}{T_1} + R\ln\frac{V_2}{V_1}\right] = mc_v \ln\frac{T_2}{T_1}$$

The heat transfer will continue until $T_2 = T_L$. Thus,

$$\Delta S_{\text{tank}} = mc_v \ln\frac{T_L}{T_1}$$

The entropy change of heat reservoir is

$$\Delta S_{\text{HR}} = \frac{Q_L}{T_L}$$

The entropy change of heat engine is zero since the engine is reversible and produces maximum work. The work reservoir involves no entropy change. Then, the total entropy change (i.e., entropy generation) is

$$\begin{aligned} S_{\text{gen}} &= \Delta S_{\text{total}} = \Delta S_{\text{tank}} + \Delta S_{\text{HR}} + \Delta S_{\text{HE}} + \Delta S_{\text{WR}} \\ &= \Delta S_{\text{tank}} + \Delta S_{\text{HR}} + 0 + 0 \end{aligned}$$

which becomes

$$S_{\text{gen}} = mc_v \ln\frac{T_L}{T_1} + \frac{Q_L}{T_L} \quad (1)$$

The expression for the thermal efficiency is

$$\eta_{\text{th}} = \frac{W}{Q_H} \quad (2)$$

or $$\eta_{\text{th}} = 1 - \frac{Q_L}{Q_H} \quad (3)$$

Heat transfer from the tank is expressed as

$$Q_H = mc_v(T_1 - T_L) \quad (4)$$

In ideal operations, the entropy generation will be zero. Then using $m = 2$ kg, $c_v = 0.718$ kJ/kg·K, $T_L = 300$ K, $Q_L = 400$ kJ and solving equations (1), (2), (3) and (4) simultaneously using an equation solver such as EES we obtain

$$T_1 = \mathbf{759.2\ K}$$
$$\eta_{\text{th}} = \mathbf{0.3934}$$
$$W = 259.4\ \text{kJ}$$
$$Q_H = 659.4\ \text{kJ}$$

(b) At the initial air temperature of 759.2 K, the entropy generation is zero and

$$S_{\text{gen}} = \mathbf{0}$$
$$\eta_{\text{th}} = \mathbf{0.3934}$$
$$W = \mathbf{259.4\ kJ}$$

At the initial air temperature of 759.2-100 = 659.2 K, we obtain

$$S_{\text{gen}} = \mathbf{0.2029\ kJ/K}$$
$$\eta_{\text{th}} = \mathbf{0.2245}$$
$$W = \mathbf{115.8\ kJ}$$

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At the initial air temperature of 759.2 + 100 = 859.2 K, we obtain

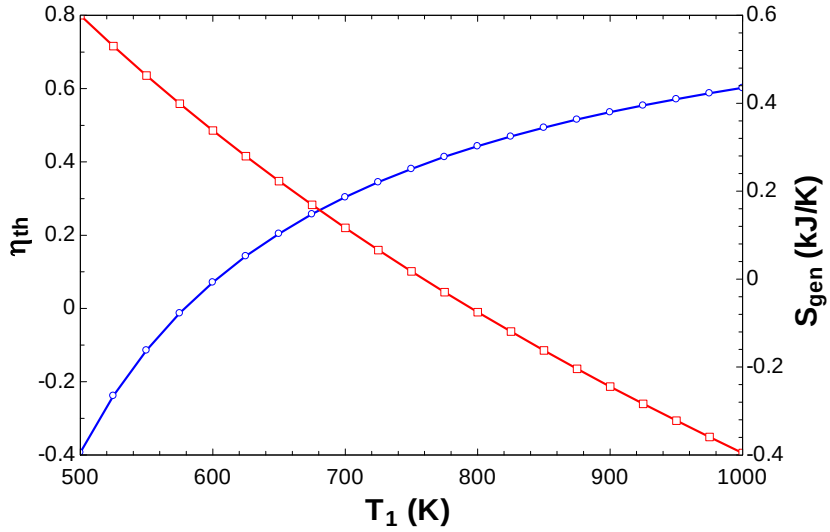
$S_{\text{gen}} = \mathbf{-0.1777\ kJ/K}$

$\eta_{\text{th}} = \mathbf{0.5019}$

$W = \mathbf{403.0\ kJ}$

A negative value for entropy generation indicates that this process is not possible with the given values.

(c) The thermal efficiency and the entropy generation as functions of the initial temperature of the air are plotted below:



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7-233 It is to be shown that for an ideal gas with constant specific heats the compressor and turbine isentropic efficiencies may be written as $\eta_C = \dfrac{(P_2 / P_1)^{(k-1)/k}}{(T_2 / T_1) - 1}$ and $\eta_T = \dfrac{(T_4 / T_3) - 1}{(P_4 / P_3)^{(k-1)/k} - 1}$.

Analysis The isentropic efficiency of a compressor for an ideal gas with constant specific heats is given by

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{c_p (T_{2s} - T_1)}{c_p (T_2 - T_1)} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

The temperature at the compressor exit fort he isentropic case is expressed as

$$T_{2s} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k}$$

Substituting,

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{T_1 \left(\dfrac{P_2}{P_1} \right)^{(k-1)/k} - T_1}{T_2 - T_1} = \frac{T_1 \left[\left(\dfrac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right]}{T_1 \left[\left(\dfrac{T_2}{T_1} \right) - 1 \right]} = \frac{\left(\dfrac{P_2}{P_1} \right)^{(k-1)/k} - 1}{\left(\dfrac{T_2}{T_1} \right) - 1}$$

The isentropic efficiency of a turbine for an ideal gas with constant specific heats is given by

$$\eta_T = \frac{h_4 - h_3}{h_{4s} - h_3} = \frac{c_p (T_4 - T_3)}{c_p (T_{4s} - T_3)} = \frac{T_4 - T_3}{T_{4s} - T_3}$$

The temperature at the turbine exit fort he isentropic case is expressed as

$$T_{4s} = T_3 \left(\frac{P_4}{P_3} \right)^{(k-1)/k}$$

Substituting,

$$\eta_C = \frac{T_4 - T_3}{T_{4s} - T_3} = \frac{T_4 - T_3}{T_3 \left(\dfrac{P_4}{P_3} \right)^{(k-1)/k} - T_3} = \frac{T_3 \left[\left(\dfrac{T_4}{T_3} \right) - 1 \right]}{T_3 \left[\left(\dfrac{P_4}{P_3} \right)^{(k-1)/k} - 1 \right]} = \frac{\left(\dfrac{T_4}{T_3} \right) - 1}{\left(\dfrac{P_4}{P_3} \right)^{(k-1)/k} - 1}$$

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7-234 An expression for the internal energy change of an ideal gas during the isentropic process is to be obtained.

Analysis The expression is obtained as follows:

$$du = Td\$^0 - Pdv = -Pdv$$

$$\Delta u = -\int Pdv, \ \ where \ \ P = const\, v^{-k}$$

$$\Delta u = -\int_1^2 const\, v^{-k} dv = -const \left. \frac{v^{1-k}}{1-k} \right|_1^2 = -\left. \frac{Pv^k v^{1-k}}{1-k} \right|_1^2$$

$$\Delta u = \left. \frac{Pv}{k-1} \right|_1^2 = \left. \frac{RT}{k-1} \right|_1^2 = \frac{R}{k-1}\left(T_2 - T_1\right)$$

$$\Delta u = C_V \left(T_2 - T_1\right)$$

7-235 The temperature of an ideal gas is given as functions of entropy and specific volume. The T-ν relation for this ideal gas undergoing an isentropic process is to be obtained.

Analysis The expressions for temperatures T_1 and T_2 are

$$T_1 = T(s_1, \nu_1) \text{ and } T_2 = T(s_2, \nu_2)$$

$$T_1 = A\nu_1^{1-k} \exp(s_1 / c_v)$$

$$T_2 = A\nu_2^{1-k} \exp(s_2 / c_v)$$

Now divide T_2 by T_1,

$$\frac{T_2}{T_1} = \frac{A\nu_2^{1-k} \exp(s_2 / c_v)}{A\nu_1^{1-k} \exp(s_1 / c_v)}$$

Since A and c_v are constants and the process is isentropic $s_2 = s_1$, the temperature ratio reduces to

$$\frac{T_2}{T_1} = \left(\frac{\nu_1}{\nu_2}\right)^{k-1}$$

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7-236 An ideal gas undergoes a reversible, steady-flow process. An expression for heat transfer is to be obtained.

Analysis (a) The conservation of mass and energy for the steady-flow process are

$$\sum_{\text{inlets}} \dot{m}_i = \sum_{\text{exits}} \dot{m}_e$$

$$\dot{Q}_{\text{net}} + \sum_{\text{inlets}} \dot{m}_i \left(h + \frac{V^2}{2} + gz \right)_i = \dot{W}_{\text{net}} + \sum_{\text{exits}} \dot{m}_e \left(h + \frac{V^2}{2} + gz \right)_e$$

where the sign of heat transfer is to the system and work is from the system. For one entrance, one exit, neglecting kinetic and potential energies, and unit mass flow, the conservation of mass and energy reduces to

$$q_{\text{net}} = w_{\text{net}} + (h_e - h_i)$$

The steady-flow, reversible work for the isothermal process is found from

$$w_{\text{net,sf,rev}} = -\int_i^e v dP = -\int_i^e \frac{RT}{P} dP = -RT \ln\left(\frac{P_e}{P_i} \right)$$

Then

$$\begin{aligned} q_{\text{net}} &= w_{\text{net}} + (h_e - h_i) \\ &= -RT - RT \ln\left(\frac{P_e}{P_i} \right) + c_p (T_e - T_i) \end{aligned}$$

However,

$$T_e = T_i$$

Thus,

$$q_{\text{net}} = -RT \ln\left(\frac{P_e}{P_i} \right)$$

(b) For the isothermal process, the heat transfer may be determined from

$$q_{\text{net}} = \int_i^e T ds = T \int_i^e ds = T(s_e - s_i)$$

The entropy change of an ideal gas is found from

$$s_e - s_i = \int_{T_i}^{T_e} \frac{c_p(T) dT}{T} - R \ln\left(\frac{P_e}{P_i} \right)$$

For the isothermal process the entropy change is

$$s_e - s_i = -R \ln\left(\frac{P_e}{P_i} \right)$$

The expression for heat transfer is

$$q_{\text{net}} = T(s_e - s_i) = -RT \ln\left(\frac{P_e}{P_i} \right)$$

which is the same as part (a).

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7-237 The temperature of an ideal gas is given as functions of entropy and specific volume. The expression for heat transfer is to be determined and compared to the result by a first law analysis.

Analysis The heat transfer may be expressed as

$$T(s,P) = AP^{(k-1)/k}\exp(s/c_p)$$

$$T(s,\nu) = AP^{(k-1)/k}\exp(s/c_p)$$

$$q = \frac{Q}{m} = \int_{s_1}^{s_2} T ds = \int_{s_1}^{s_2} AP^{(k-1)/k}\exp(s/c_p)ds$$

For P = constant, the integral becomes

$$q = AP^{(k-1)/k}\int_{s_1}^{s_2}\exp(s/c_p)ds = AP^{(k-1)/k}c_p\exp(s/c_p)\Big|_{s_1}^{s_2}$$
$$= AP^{(k-1)/k}c_p\left[\exp(s_2/c_p) - \exp(s_1/c_p)\right]$$

Noting that

$$T_1 = AP^{(k-1)/k}\exp(s_1/c_p)$$
$$T_2 = AP^{(k-1)/k}\exp(s_2/c_p)$$

We obtain

$$q = c_p(T_2 - T_1)$$

This result is the same as that given by applying the first law to a closed system undergoing a constant pressure process.

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7-238 A two-stage compression with intercooling process is considered. The isentropic efficiencies for the two compression processes are given. The optimum intercooling pressure is to be obtained.

Analysis The work of compression is

$$\begin{aligned} w_{\text{comp}} &= c_p(T_2 - T_1) + c_p(T_4 - T_3) \\ &= c_p \frac{1}{\eta_{C,L}}(T_{2s} - T_1) + c_p \frac{1}{\eta_{C,H}}(T_{4s} - T_3) \\ &= c_p \left[T_1 \frac{1}{\eta_{C,L}}(T_{2s}/T_1 - 1) + T_3 \frac{1}{\eta_{C,H}}(T_{4s}/T_3 - 1) \right] \\ &= c_p \left[T_1 \frac{1}{\eta_{C,L}} \left(\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right) + T_3 \frac{1}{\eta_{C,H}} \left(\left(\frac{P_4}{P_3} \right)^{(k-1)/k} - 1 \right) \right] \end{aligned}$$

We are to assume that intercooling takes place at constant pressure and the gases can be cooled to the inlet temperature for the compressor, such that $P_3 = P_2$ and $T_3 = T_1$. Then the total work supplied to the compressor becomes

$$\begin{aligned} w_{\text{comp}} &= c_p T_1 \left[\frac{1}{\eta_{C,L}} \left(\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right) + \frac{1}{\eta_{C,H}} \left(\left(\frac{P_4}{P_2} \right)^{(k-1)/k} - 1 \right) \right] \\ &= c_p T_1 \left[\frac{1}{\eta_{C,L}} \left(\frac{P_2}{P_1} \right)^{(k-1)/k} + \frac{1}{\eta_{C,H}} \left(\frac{P_4}{P_2} \right)^{(k-1)/k} - \frac{1}{\eta_{C,L}} - \frac{1}{\eta_{C,H}} \right] \end{aligned}$$

To find the unknown pressure P_2 that gives the minimum work input for fixed compressor inlet conditions T_1, P_1, exit pressure P_4, and isentropic efficiencies, we set

$$\frac{dw_{\text{comp}}(P_2)}{dP_2} = 0$$

This yields

$$P_2 = \sqrt{P_1 P_4} \left(\frac{\eta_{C,L}}{\eta_{C,H}} \right)^{\frac{k}{2(k-1)}}$$

$$\frac{P_2}{P_1} = (\eta_{C,L})^{\frac{-k}{k-1}} = \frac{P_4}{P_2} (\eta_{C,H})^{\frac{-k}{k-1}}$$

Since $P_2 = P_3$, the pressure ratios across the two compressors are related by

$$\frac{P_2}{P_1} = (\eta_{C,L})^{\frac{-k}{k-1}} = \frac{P_4}{P_3} (\eta_{C,H})^{\frac{-k}{k-1}}$$

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Fundamentals of Engineering (FE) Exam Problems

7-239 Steam is condensed at a constant temperature of 30°C as it flows through the condenser of a power plant by rejecting heat at a rate of 55 MW. The rate of entropy change of steam as it flows through the condenser is

(a) –1.83 MW/K (b) –0.18 MW/K (c) 0 MW/K (d) 0.56 MW/K (e) 1.22 MW/K

Answer (b) –0.18 MW/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=30 "C"
Q_out=55 "MW"
S_change=-Q_out/(T1+273)  "MW/K"

"Some Wrong Solutions with Common Mistakes:"
W1_S_change=0 "Assuming no change"
W2_S_change=Q_out/T1 "Using temperature in C"
W3_S_change=Q_out/(T1+273) "Wrong sign"
W4_S_change=-s_fg "Taking entropy of vaporization"
s_fg=(ENTROPY(Steam_IAPWS,T=T1,x=1)-ENTROPY(Steam_IAPWS,T=T1,x=0))
```

7-240 Steam is compressed from 6 MPa and 300°C to 10 MPa isentropically. The final temperature of the steam is

(a) 290°C (b) 300°C (c) 311°C (d) 371°C (e) 422°C

Answer (d) 371°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=6000 "kPa"
T1=300 "C"
P2=10000 "kPa"
s2=s1
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
T2=TEMPERATURE(Steam_IAPWS,s=s2,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
W2_T2=TEMPERATURE(Steam_IAPWS,x=0,P=P2) "Saturation temperature at P2"
W3_T2=TEMPERATURE(Steam_IAPWS,x=0,P=P2) "Saturation temperature at P1"
```

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7-241 An apple with an average mass of 0.12 kg and average specific heat of 3.65 kJ/kg.°C is cooled from 25°C to 5°C. The entropy change of the apple is

(a) –0.705 kJ/K (b) –0.254 kJ/K (c) -0.0304 kJ/K (d) 0 kJ/K (e) 0.348 kJ/K

Answer (c) –0.0304 kJ/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.65 "kJ/kg.K"
m=0.12 "kg"
T1=25 "C"
T2=5 "C"
S_change=m*C*ln((T2+273)/(T1+273))

"Some Wrong Solutions with Common Mistakes:"
W1_S_change=C*ln((T2+273)/(T1+273)) "Not using mass"
W2_S_change=m*C*ln(T2/T1) "Using C"
W3_S_change=m*C*(T2-T1) "Using Wrong relation"
```

7-242 A piston-cylinder device contains 5 kg of saturated water vapor at 3 MPa. Now heat is rejected from the cylinder at constant pressure until the water vapor completely condenses so that the cylinder contains saturated liquid at 3 MPa at the end of the process. The entropy change of the system during this process is

(a) 0 kJ/K (b) -3.5 kJ/K (c) -12.5 kJ/K (d) -17.7 kJ/K (e) -19.5 kJ/K

Answer (d) -17.7 kJ/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=3000 "kPa"
m=5 "kg"
s_fg=(ENTROPY(Steam_IAPWS,P=P1,x=1)-ENTROPY(Steam_IAPWS,P=P1,x=0))
S_change=-m*s_fg "kJ/K"
```

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7-243 Helium gas is compressed from 1 atm and 25°C to a pressure of 10 atm adiabatically. The lowest temperature of helium after compression is

(a) 25°C (b) 63°C (c) 250°C (d) 384°C (e) 476°C

Answer (e) 476°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
P1=101.325 "kPa"
T1=25 "C"
P2=10*101.325 "kPa"
"s2=s1"
"The exit temperature will be lowest for isentropic compression,"
T2=(T1+273)*(P2/P1)^((k-1)/k) "K"
T2_C= T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
W2_T2=T1*(P2/P1)^((k-1)/k) "Using C instead of K"
W3_T2=(T1+273)*(P2/P1)-273 "Assuming T is proportional to P"
W4_T2=T1*(P2/P1) "Assuming T is proportional to P, using C"
```

7-244 Steam expands in an adiabatic turbine from 4 MPa and 500°C to 0.1 MPa at a rate of 2 kg/s. If steam leaves the turbine as saturated vapor, the power output of the turbine is

(a) 2058 kW (b) 1910 kW (c) 1780 kW (d) 1674 kW (e) 1542 kW

Answer (e) 1542 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=4000 "kPa"
T1=500 "C"
P2=100 "kPa"
x2=1
m=2 "kg/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,x=x2,P=P2)
W_out=m*(h1-h2)

"Some Wrong Solutions with Common Mistakes:"
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
h2s=ENTHALPY(Steam_IAPWS, s=s1,P=P2)
W1_Wout=m*(h1-h2s)  "Assuming isentropic expansion"
```

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7-245 Argon gas expands in an adiabatic turbine from 3 MPa and 750°C to 0.2 MPa at a rate of 5 kg/s. The maximum power output of the turbine is

(a) 1.06 MW (b) 1.29 MW (c) 1.43 MW (d) 1.76 MW (e) 2.08 MW

Answer (d) 1.76 MW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=0.5203
k=1.667
P1=3000 "kPa"
T1=750+273 "K"
m=5 "kg/s"
P2=200 "kPa"
"s2=s1"
T2=T1*(P2/P1)^((k-1)/k)
W_max=m*Cp*(T1-T2)

"Some Wrong Solutions with Common Mistakes:"
Cv=0.2081"kJ/kg.K"
W1_Wmax=m*Cv*(T1-T2) "Using Cv"
T22=T1*(P2/P1)^((k-1)/k) "Using C instead of K"
W2_Wmax=m*Cp*(T1-T22)
W3_Wmax=Cp*(T1-T2) "Not using mass flow rate"
T24=T1*(P2/P1) "Assuming T is proportional to P, using C"
W4_Wmax=m*Cp*(T1-T24)
```

7-246 A unit mass of a substance undergoes an irreversible process from state 1 to state 2 while gaining heat from the surroundings at temperature T in the amount of q. If the entropy of the substance is s_1 at state 1, and s_2 at state 2, the entropy change of the substance Δs during this process is

(a) $\Delta s < s_2 - s_1$ (b) $\Delta s > s_2 - s_1$ (c) $\Delta s = s_2 - s_1$ (d) $\Delta s = s_2 - s_1 + q/T$ (e) $\Delta s > s_2 - s_1 + q/T$

Answer (c) $\Delta s = s_2 - s_1$

7-247 A unit mass of an ideal gas at temperature T undergoes a reversible isothermal process from pressure P_1 to pressure P_2 while loosing heat to the surroundings at temperature T in the amount of q. If the gas constant of the gas is R, the entropy change of the gas Δs during this process is

(a) $\Delta s = R \ln(P_2/P_1)$ (b) $\Delta s = R \ln(P_2/P_1) - q/T$ (c) $\Delta s = R \ln(P_1/P_2)$ (d) $\Delta s = R \ln(P_1/P_2) - q/T$ (e) $\Delta s = 0$

Answer (c) $\Delta s = R \ln(P_1/P_2)$

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7-248 Air is compressed from room conditions to a specified pressure in a reversible manner by two compressors: one isothermal and the other adiabatic. If the entropy change of air is Δs_{isot} during the reversible isothermal compression, and Δs_{adia} during the reversible adiabatic compression, the correct statement regarding entropy change of air per unit mass is

(a) $\Delta s_{isot} = \Delta s_{adia} = 0$ (b) $\Delta s_{isot} = \Delta s_{adia} > 0$ (c) $\Delta s_{adia} > 0$ (d) $\Delta s_{isot} < 0$ (e) $\Delta s_{isot} = 0$

Answer (d) $\Delta s_{isot} < 0$

7-249 Helium gas is compressed from 27°C and 3.5 m^3/kg to 0.775 m^3/kg in a reversible adiabatic manner. The temperature of helium after compression is

(a) 74°C (b) 122°C (c) 547°C (d) 709°C (e) 1082°C

Answer (c) 547°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
v1=3.5 "m^3/kg"
T1=27 "C"
v2=0.775 "m^3/kg"
"s2=s1"
"The exit temperature is determined from isentropic compression relation,"
T2=(T1+273)*(v1/v2)^(k-1) "K"
T2_C= T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
W2_T2=T1*(v1/v2)^(k-1) "Using C instead of K"
W3_T2=(T1+273)*(v1/v2)-273 "Assuming T is proportional to v"
W4_T2=T1*(v1/v2) "Assuming T is proportional to v, using C"
```

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7-250 Heat is lost through a plane wall steadily at a rate of 600 W. If the inner and outer surface temperatures of the wall are 20°C and 5°C, respectively, the rate of entropy generation within the wall is

(a) 0.11 W/K (b) 4.21 W/K (c) 2.10 W/K (d) 42.1 W/K (e) 90.0 W/K

Answer (a) 0.11 W/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q=600 "W"
T1=20+273 "K"
T2=5+273 "K"
"Entropy balance S_in - S_out + S_gen= DS_system for the wall for steady operation gives"
Q/T1-Q/T2+S_gen=0  "W/K"

"Some Wrong Solutions with Common Mistakes:"
Q/(T1+273)-Q/(T2+273)+W1_Sgen=0  "Using C instead of K"
W2_Sgen=Q/((T1+T2)/2)  "Using avegage temperature in K"
W3_Sgen=Q/((T1+T2)/2-273)  "Using avegage temperature in C"
W4_Sgen=Q/(T1-T2+273)  "Using temperature difference in K"
```

7-251 Air is compressed steadily and adiabatically from 17°C and 90 kPa to 200°C and 400 kPa. Assuming constant specific heats for air at room temperature, the isentropic efficiency of the compressor is

(a) 0.76 (b) 0.94 (c) 0.86 (d) 0.84 (e) 1.00

Answer (d) 0.84

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=1.005 "kJ/kg.K"
k=1.4
P1=90 "kPa"
T1=17 "C"
P2=400 "kPa"
T2=200 "C"
T2s=(T1+273)*(P2/P1)^((k-1)/k)-273
Eta_comp=(Cp*(T2s-T1))/(Cp*(T2-T1))

"Some Wrong Solutions with Common Mistakes:"
T2sW1=T1*(P2/P1)^((k-1)/k) "Using C instead of K in finding T2s"
W1_Eta_comp=(Cp*(T2sW1-T1))/(Cp*(T2-T1))
W2_Eta_comp=T2s/T2 "Using wrong definition for isentropic efficiency, and using C"
W3_Eta_comp=(T2s+273)/(T2+273) "Using wrong definition for isentropic efficiency, with K"
```

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7-252 Argon gas expands in an adiabatic turbine steadily from 600°C and 800 kPa to 80 kPa at a rate of 2.5 kg/s. For an isentropic efficiency of 88%, the power produced by the turbine is

(a) 240 kW (b) 361 kW (c) 414 kW (d) 602 kW (e) 777 kW

Answer (d) 602 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=0.5203 "kJ/kg-K"
k=1.667
m=2.5 "kg/s"
T1=600 "C"
P1=800 "kPa"
P2=80 "kPa"
T2s=(T1+273)*(P2/P1)^((k-1)/k)-273
Eta_turb=0.88
Eta_turb=(Cp*(T2-T1))/(Cp*(T2s-T1))
W_out=m*Cp*(T1-T2)

"Some Wrong Solutions with Common Mistakes:"
T2sW1=T1*(P2/P1)^((k-1)/k) "Using C instead of K to find T2s"
Eta_turb=(Cp*(T2W1-T1))/(Cp*(T2sW1-T1))
W1_Wout=m*Cp*(T1-T2W1)
Eta_turb=(Cp*(T2s-T1))/(Cp*(T2W2-T1))  "Using wrong definition for isentropic efficiency, and using C"
W2_Wout=m*Cp*(T1-T2W2)
W3_Wout=Cp*(T1-T2) "Not using mass flow rate"
Cv=0.3122 "kJ/kg.K"
W4_Wout=m*Cv*(T1-T2) "Using Cv instead of Cp"
```

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7-253 Water enters a pump steadily at 100 kPa at a rate of 35 L/s and leaves at 800 kPa. The flow velocities at the inlet and the exit are the same, but the pump exit where the discharge pressure is measured is 6.1 m above the inlet section. The minimum power input to the pump is

(a) 34 kW (b) 22 kW (c) 27 kW (d) 52 kW (e) 44 kW

Answer (c) 27 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=0.035 "m^3/s"
g=9.81 "m/s^2"
h=6.1 "m"
P1=100 "kPa"
T1=20 "C"
P2=800 "kPa"
"Pump power input is minimum when compression is reversible and thus w=v(P2-P1)+Dpe"
v1=VOLUME(Steam_IAPWS,T=T1,P=P1)
m=V/v1
W_min=m*v1*(P2-P1)+m*g*h/1000 "kPa.m^3/s=kW"
"(The effect of 6.1 m elevation difference turns out to be small)"

"Some Wrong Solutions with Common Mistakes:"
W1_Win=m*v1*(P2-P1) "Disregarding potential energy"
W2_Win=m*v1*(P2-P1)-m*g*h/1000  "Subtracting potential energy instead of adding"
W3_Win=m*v1*(P2-P1)+m*g*h "Not using the conversion factor 1000 in PE term"
W4_Win=m*v1*(P2+P1)+m*g*h/1000 "Adding pressures instead of subtracting"
```

7-254 Air at 15°C is compressed steadily and isothermally from 100 kPa to 700 kPa at a rate of 0.12 kg/s. The minimum power input to the compressor is

(a) 1.0 kW (b) 11.2 kW (c) 25.8 kW (d) 19.3 kW (e) 161 kW

Answer (d) 19.3 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
k=1.4
P1=100 "kPa"
T=15 "C"
m=0.12 "kg/s"
P2=700 "kPa"
Win=m*R*(T+273)*ln(P2/P1)

"Some Wrong Solutions with Common Mistakes:"
W1_Win=m*R*T*ln(P2/P1)  "Using C instead of K"
W2_Win=m*T*(P2-P1) "Using wrong relation"
W3_Win=R*(T+273)*ln(P2/P1)  "Not using mass flow rate"
```

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7-255 Air is to be compressed steadily and isentropically from 1 atm to 16 atm by a two-stage compressor. To minimize the total compression work, the intermediate pressure between the two stages must be

(a) 3 atm (b) 4 atm (c) 8.5 atm (d) 9 atm (e) 12 atm

Answer (b) 4 atm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1 "atm"
P2=16 "atm"
P_mid=SQRT(P1*P2)

"Some Wrong Solutions with Common Mistakes:"
W1_P=(P1+P2)/2  "Using average pressure"
W2_P=P1*P2/2  "Half of product"
```

7-256 Helium gas enters an adiabatic nozzle steadily at 500°C and 600 kPa with a low velocity, and exits at a pressure of 90 kPa. The highest possible velocity of helium gas at the nozzle exit is

(a) 1475 m/s (b) 1662 m/s (c) 1839 m/s (d) 2066 m/s (e) 3040 m/s

Answer (d) 2066 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=5.1926 "kJ/kg.K"
Cv=3.1156 "kJ/kg.K"
T1=500 "C"
P1=600 "kPa"
Vel1=0
P2=90 "kPa"
"s2=s1 for maximum exit velocity"
"The exit velocity will be highest for isentropic expansion,"
T2=(T1+273)*(P2/P1)^((k-1)/k)-273 "C"
"Energy balance for this case is h+ke=constant for the fluid stream (Q=W=pe=0)"
(0.5*Vel1^2)/1000+Cp*T1=(0.5*Vel2^2)/1000+Cp*T2

"Some Wrong Solutions with Common Mistakes:"
T2a=T1*(P2/P1)^((k-1)/k) "Using C for temperature"
(0.5*Vel1^2)/1000+Cp*T1=(0.5*W1_Vel2^2)/1000+Cp*T2a
T2b=T1*(P2/P1)^((k-1)/k) "Using Cv"
(0.5*Vel1^2)/1000+Cv*T1=(0.5*W2_Vel2^2)/1000+Cv*T2b
T2c=T1*(P2/P1)^k "Using wrong relation"
(0.5*Vel1^2)/1000+Cp*T1=(0.5*W3_Vel2^2)/1000+Cp*T2c
```

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7-257 Combustion gases with a specific heat ratio of 1.3 enter an adiabatic nozzle steadily at 800°C and 800 kPa with a low velocity, and exit at a pressure of 85 kPa. The lowest possible temperature of combustion gases at the nozzle exit is

(a) 43°C (b) 237°C (c) 367°C (d) 477°C (e) 640°C

Answer (c) 367°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.3
T1=800 "C"
P1=800 "kPa"
P2=85 "kPa"
"Nozzle exit temperature will be lowest for isentropic operation"
T2=(T1+273)*(P2/P1)^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1*(P2/P1)^((k-1)/k) "Using C for temperature"
W2_T2=(T1+273)*(P2/P1)^((k-1)/k) "Not converting the answer to C"
W3_T2=T1*(P2/P1)^k "Using wrong relation"
```

7-258 Steam enters an adiabatic turbine steadily at 400°C and 5 MPa, and leaves at 20 kPa. The highest possible percentage of mass of steam that condenses at the turbine exit and leaves the turbine as a liquid is

(a) 4% (b) 8% (c) 12% (d) 18% (e) 0%

Answer (d) 18%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=5000 "kPa"
T1=400 "C"
P2=20 "kPa"
s2=s1
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
x2=QUALITY(Steam_IAPWS,s=s2,P=P2)
moisture=1-x2
```

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7-259 Liquid water enters an adiabatic piping system at 15°C at a rate of 8 kg/s. If the water temperature rises by 0.2°C during flow due to friction, the rate of entropy generation in the pipe is

(a) 23 W/K (b) 55 W/K (c) 68 W/K (d) 220 W/K (e) 443 W/K

Answer (a) 23 W/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=4180 "J/kg.K"
m=8 "kg/s"
T1=15 "C"
T2=15.2 "C"
S_gen=m*Cp*ln((T2+273)/(T1+273)) "W/K"

"Some Wrong Solutions with Common Mistakes:"
W1_Sgen=m*Cp*ln(T2/T1) "Using deg. C"
W2_Sgen=Cp*ln(T2/T1) "Not using mass flow rate with deg. C"
W3_Sgen=Cp*ln((T2+273)/(T1+273)) "Not using mass flow rate with deg. C"
```

7-260 Liquid water is to be compressed by a pump whose isentropic efficiency is 75 percent from 0.2 MPa to 5 MPa at a rate of 0.15 m^3/min. The required power input to this pump is

(*a*) 4.8 kW (*b*) 6.4 kW (*c*) 9.0 kW (*d*) 16.0 kW (*e*) 12.0 kW

Answer (*d*) 16.0 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=0.15/60 "m^3/s"
rho=1000 "kg/m^3"
v1=1/rho
m=rho*V "kg/s"
P1=200 "kPa"
Eta_pump=0.75
P2=5000 "kPa"
"Reversible pump power input is w =mv(P2-P1) = V(P2-P1)"
W_rev=m*v1*(P2-P1) "kPa.m^3/s=kW"
W_pump=W_rev/Eta_pump

"Some Wrong Solutions with Common Mistakes:"
W1_Wpump=W_rev*Eta_pump "Multiplying by efficiency"
W2_Wpump=W_rev "Disregarding efficiency"
W3_Wpump=m*v1*(P2+P1)/Eta_pump "Adding pressures instead of subtracting"
```

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7-261 Steam enters an adiabatic turbine at 8 MPa and 500°C at a rate of 18 kg/s, and exits at 0.2 MPa and 300°C. The rate of entropy generation in the turbine is

(*a*) 0 kW/K (*b*) 7.2 kW/K (*c*) 21 kW/K (*d*) 15 kW/K (*e*) 17 kW/K

Answer (*c*) 21 kW/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=8000 "kPa"
T1=500 "C"
m=18 "kg/s"
P2=200 "kPa"
T2=300 "C"
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
s2=ENTROPY(Steam_IAPWS,T=T2,P=P2)
S_gen=m*(s2-s1) "kW/K"

"Some Wrong Solutions with Common Mistakes:"
W1_Sgen=0 "Assuming isentropic expansion"
```

7-262 Helium gas is compressed steadily from 90 kPa and 25°C to 800 kPa at a rate of 2 kg/min by an adiabatic compressor. If the compressor consumes 80 kW of power while operating, the isentropic efficiency of this compressor is

(*a*) 54.0% (*b*) 80.5% (*c*) 75.8% (*d*) 90.1% (*e*) 100%

Answer (*d*) 90.1%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=5.1926 "kJ/kg-K"
Cv=3.1156 "kJ/kg.K"
k=1.667
m=2/60 "kg/s"
T1=25 "C"
P1=90 "kPa"
P2=800 "kPa"
W_comp=80 "kW"
T2s=(T1+273)*(P2/P1)^((k-1)/k)-273
W_s=m*Cp*(T2s-T1)
Eta_comp=W_s/W_comp

"Some Wrong Solutions with Common Mistakes:"
T2sA=T1*(P2/P1)^((k-1)/k) "Using C instead of K"
W1_Eta_comp=m*Cp*(T2sA-T1)/W_comp
W2_Eta_comp=m*Cv*(T2s-T1)/W_comp "Using Cv instead of Cp"
```

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7-263 … 7-268 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 8
EXERGY – A MEASURE OF WORK POTENTIAL

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Exergy, Irreversibility, Reversible Work, and Second-Law Efficiency

8-1C Reversible work and irreversibility are identical for processes that involve no actual useful work.

8-2C The dead state.

8-3C Yes; exergy is a function of the state of the surroundings as well as the state of the system.

8-4C Useful work differs from the actual work by the surroundings work. They are identical for systems that involve no surroundings work such as steady-flow systems.

8-5C Yes.

8-6C No, not necessarily. The well with the higher temperature will have a higher exergy.

8-7C The system that is at the temperature of the surroundings has zero exergy. But the system that is at a lower temperature than the surroundings has some exergy since we can run a heat engine between these two temperature levels.

8-8C They would be identical.

8-9C The second-law efficiency is a measure of the performance of a device relative to its performance under reversible conditions. It differs from the first law efficiency in that it is not a conversion efficiency.

8-10C No. The power plant that has a lower thermal efficiency may have a higher second-law efficiency.

8-11C No. The refrigerator that has a lower COP may have a higher second-law efficiency.

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8-12C A processes with $W_{rev} = 0$ is reversible if it involves no actual useful work. Otherwise it is irreversible.

8-13C Yes.

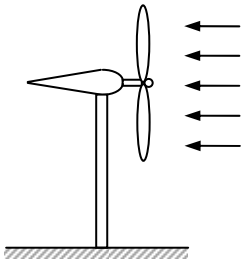
8-14 Windmills are to be installed at a location with steady winds to generate power. The minimum number of windmills that need to be installed is to be determined.

Assumptions Air is at standard conditions of 1 atm and 25°C

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1).

Analysis The exergy or work potential of the blowing air is the kinetic energy it possesses,

$$\text{Exergy} = \text{ke} = \frac{V^2}{2} = \frac{(6\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.0180\text{ kJ/kg}$$



At standard atmospheric conditions (25°C, 101 kPa), the density and the mass flow rate of air are

$$\rho = \frac{P}{RT} = \frac{101\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 1.18\text{ m}^3/\text{kg}$$

and

$$\dot{m} = \rho A V_1 = \rho\frac{\pi D^2}{4}V_1 = (1.18\text{ kg/m}^3)(\pi/4)(20\text{ m})^2(6\text{ m/s}) = 2225\text{ kg/s}$$

Thus,

$$\text{Available Power} = \dot{m}\text{ke} = (2225\text{ kg/s})(0.0180\text{ kJ/kg}) = 40.05\text{ kW}$$

The minimum number of windmills that needs to be installed is

$$N = \frac{\dot{W}_{\text{total}}}{\dot{W}} = \frac{900\text{ kW}}{40.05\text{ kW}} = 22.5 \cong \mathbf{23\ windmills}$$

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8-15E Saturated steam is generated in a boiler by transferring heat from the combustion gases. The wasted work potential associated with this heat transfer process is to be determined. Also, the effect of increasing the temperature of combustion gases on the irreversibility is to be discussed.

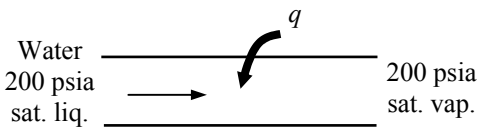
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The properties of water at the inlet and outlet of the boiler and at the dead state are (Tables A-4E through A-6E)

$$\left.\begin{array}{l} P_1 = 200 \text{ psia} \\ x_1 = 0 \ \text{(sat. liq.)} \end{array}\right\} \begin{array}{l} h_1 = h_f = 355.46 \text{ Btu/lbm} \\ s_1 = s_f = 0.54379 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ psia} \\ x_2 = 1 \ \text{(sat. vap.)} \end{array}\right\} \begin{array}{l} h_2 = h_g = 1198.8 \text{ Btu/lbm} \\ s_2 = s_g = 1.5460 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} T_0 = 80°\text{F} \\ P_0 = 14.7 \text{ psia} \end{array}\right\} \begin{array}{l} h_0 \cong h_{f@80°\text{F}} = 48.07 \text{ Btu/lbm} \\ s_0 \cong s_{f@80°\text{F}} = 0.09328 \text{ Btu/lbm} \cdot \text{R} \end{array}$$



The heat transfer during the process is

$$q_{\text{in}} = h_2 - h_1 = 1198.8 - 355.46 = 843.3 \text{ Btu/lbm}$$

The entropy generation associated with this process is

$$\begin{aligned} s_{\text{gen}} &= \Delta s_w + \Delta s_R = (s_2 - s_1) - \frac{q_{\text{in}}}{T_R} \\ &= (1.5460 - 0.54379)\text{Btu/lbm} \cdot \text{R} - \frac{843.3 \text{ Btu/lbm}}{(500 + 460)\text{R}} \\ &= 0.12377 \text{ Btu/lbm} \cdot \text{R} \end{aligned}$$

The wasted work potential (exergy destruction is)

$$x_{\text{dest}} = T_0 s_{\text{gen}} = (80 + 460 \text{ R})(0.12377 \text{ Btu/lbm} \cdot \text{R}) = \mathbf{66.8 \text{ Btu/lbm}}$$

The work potential (exergy) of the steam stream is

$$\begin{aligned} \Delta\psi_w &= h_2 - h_1 - T_0(s_2 - s_1) \\ &= (1198.8 - 355.46)\text{Btu/lbm} - (540 \text{ R})(1.5460 - 0.54379)\text{Btu/lbm} \cdot \text{R} \\ &= 302.1 \text{ Btu/lbm} \end{aligned}$$

Increasing the temperature of combustion gases does not effect the work potential of steam stream since it is determined by the states at which water enters and leaves the boiler.

Discussion This problem may also be solved as follows:

Exergy transfer by heat transfer:

$$x_{\text{heat}} = q\left(1 - \frac{T_0}{T_R}\right) = (843.3)\left(1 - \frac{540}{960}\right) = 368.9 \text{ Btu/lbm}$$

Exergy increase of steam:

$$\Delta\psi_w = 302.1 \text{ Btu/lbm}$$

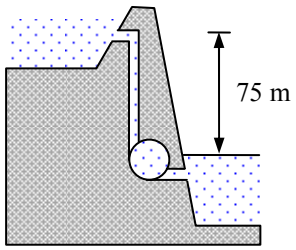
The net exergy destruction:

$$x_{\text{dest}} = x_{\text{heat}} - \Delta\psi_w = 368.9 - 302.1 = \mathbf{66.8 \text{ Btu/lbm}}$$

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8-16 Water is to be pumped to a high elevation lake at times of low electric demand for use in a hydroelectric turbine at times of high demand. For a specified energy storage capacity, the minimum amount of water that needs to be stored in the lake is to be determined.

Assumptions The evaporation of water from the lake is negligible.



Analysis The exergy or work potential of the water is the potential energy it possesses,

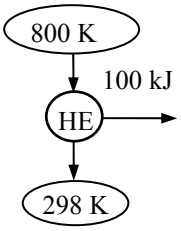
$$\text{Exergy} = \text{PE} = mgh$$

Thus,

$$m = \frac{PE}{gh} = \frac{5\times10^6\ \text{kWh}}{(9.8\ \text{m/s}^2)(75\ \text{m})}\left(\frac{3600\ \text{s}}{1\ \text{h}}\right)\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kW}\cdot\text{s/kg}}\right) = \mathbf{2.45\times10^{10}\ kg}$$

8-17 A body contains a specified amount of thermal energy at a specified temperature. The amount that can be converted to work is to be determined.

Analysis The amount of heat that can be converted to work is simply the amount that a reversible heat engine can convert to work,

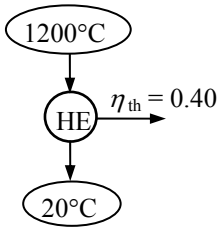


$$\eta_{\text{th,rev}} = 1 - \frac{T_0}{T_H} = 1 - \frac{298\ \text{K}}{800\ \text{K}} = 0.6275$$

$$W_{\text{max,out}} = W_{\text{rev,out}} = \eta_{\text{th,rev}} Q_{\text{in}}$$
$$= (0.6275)(100\ \text{kJ})$$
$$= \mathbf{62.75\ kJ}$$

8-18 The thermal efficiency of a heat engine operating between specified temperature limits is given. The second-law efficiency of a engine is to be determined.

Analysis The thermal efficiency of a reversible heat engine operating between the same temperature reservoirs is



$$\eta_{\text{th,rev}} = 1 - \frac{T_0}{T_H} = 1 - \frac{293\ \text{K}}{1200 + 273\ \text{K}} = 0.801$$

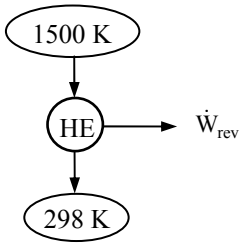
Thus,

$$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\text{th,rev}}} = \frac{0.40}{0.801} = \mathbf{49.9\%}$$

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8-19 A heat reservoir at a specified temperature can supply heat at a specified rate. The exergy of this heat supplied is to be determined.

Analysis The exergy of the supplied heat, in the rate form, is the amount of power that would be produced by a reversible heat engine,

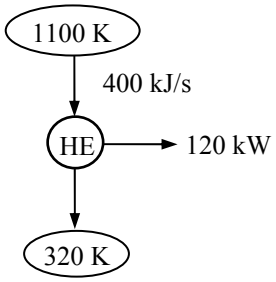


$$\eta_{th,max} = \eta_{th,rev} = 1 - \frac{T_0}{T_H} = 1 - \frac{298\ \text{K}}{1500\ \text{K}} = 0.8013$$

$$\text{Exergy} = \dot{W}_{max,out} = \dot{W}_{rev,out} = \eta_{th,rev}\dot{Q}_{in}$$
$$= (0.8013)(150{,}000 / 3600\ \text{kJ/s})$$
$$= \mathbf{33.4\ kW}$$

8-20 A heat engine receives heat from a source at a specified temperature at a specified rate, and rejects the waste heat to a sink. For a given power output, the reversible power, the rate of irreversibility, and the 2nd law efficiency are to be determined.

Analysis (a) The reversible power is the power produced by a reversible heat engine operating between the specified temperature limits,



$$\eta_{th,max} = \eta_{th,rev} = 1 - \frac{T_L}{T_H} = 1 - \frac{320\ \text{K}}{1100\ \text{K}} = 0.7091$$

$$\dot{W}_{rev,out} = \eta_{th,rev}\dot{Q}_{in} = (0.7091)(400\ \text{kJ/s}) = \mathbf{283.6\ kW}$$

(b) The irreversibility rate is the difference between the reversible power and the actual power output:

$$\dot{I} = \dot{W}_{rev,out} - \dot{W}_{u,out} = 283.6 - 120 = \mathbf{163.6\ kW}$$

(c) The second law efficiency is determined from its definition,

$$\eta_{II} = \frac{W_{u,out}}{W_{rev,out}} = \frac{120\ \text{kW}}{283.6\ \text{kW}} = 0.423 = \mathbf{42.3\%}$$

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8-21 Problem 8-20 is reconsidered. The effect of reducing the temperature at which the waste heat is rejected on the reversible power, the rate of irreversibility, and the second law efficiency is to be studied and the results are to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
T_H= 1100 [K]
Q_dot_H= 400 [kJ/s]
{T_L=320 [K]}
W_dot_out = 120 [kW]
T_Lsurr =25 [C]

"The reversible work is the maximum work done by the Carnot Engine between T_H and T_L:"
Eta_Carnot=1 - T_L/T_H
W_dot_rev=Q_dot_H*Eta_Carnot

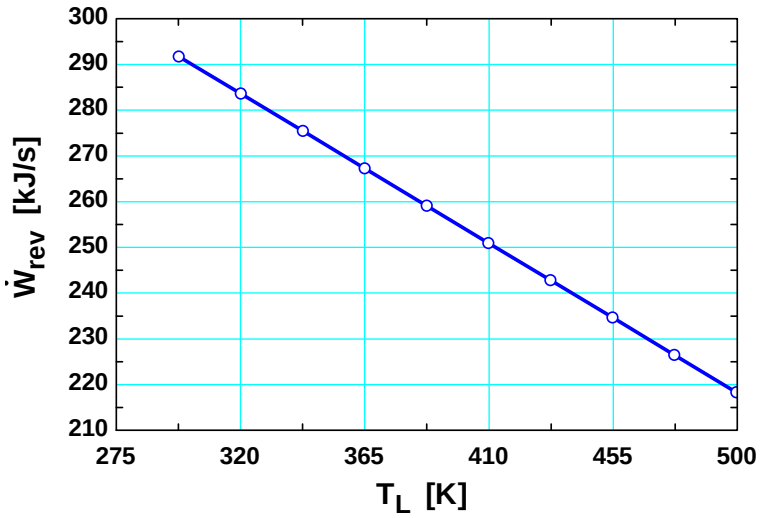
"The irreversibility is given as:"
I_dot = W_dot_rev-W_dot_out

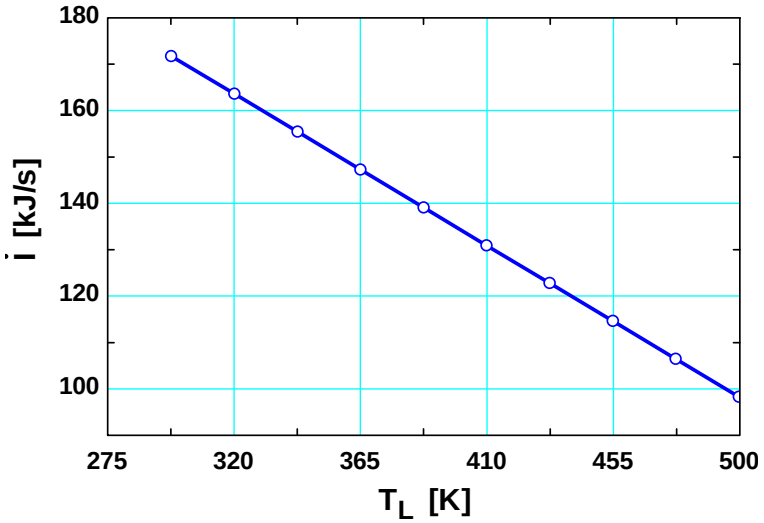
"The thermal efficiency is, in percent:"
Eta_th = Eta_Carnot*Convert(, %)

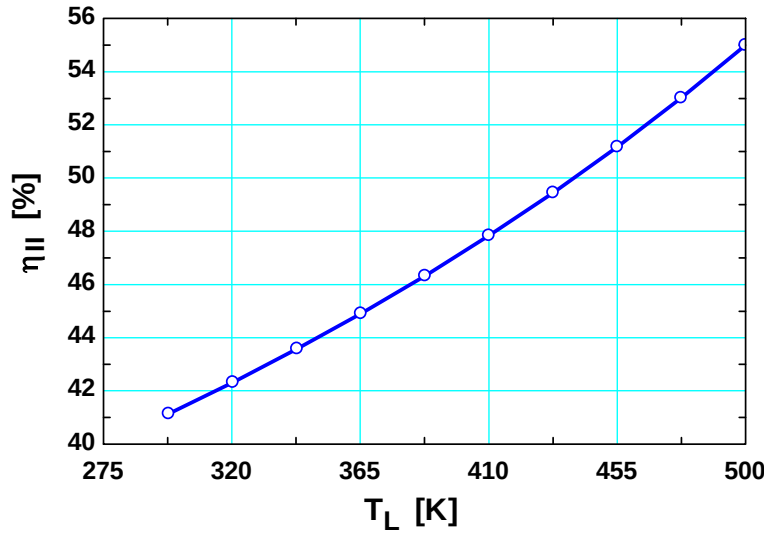
"The second law efficiency is, in percent:"
Eta_II = W_dot_out/W_dot_rev*Convert(, %)
```

T_L [K]	W_{rev} [kJ/s]	I [kJ/s]	η_{II} [%]
500	218.2	98.18	55
477.6	226.3	106.3	53.02
455.1	234.5	114.5	51.17
432.7	242.7	122.7	49.45
410.2	250.8	130.8	47.84
387.8	259	139	46.33
365.3	267.2	147.2	44.92
342.9	275.3	155.3	43.59
320.4	283.5	163.5	42.33
298	291.6	171.6	41.15

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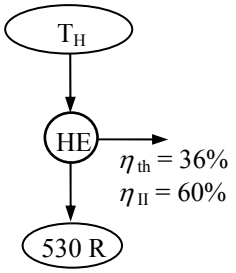






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8-22E The thermal efficiency and the second-law efficiency of a heat engine are given. The source temperature is to be determined.



Analysis From the definition of the second law efficiency,

$$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\text{th,rev}}} \longrightarrow \eta_{\text{th,rev}} = \frac{\eta_{\text{th}}}{\eta_{\text{II}}} = \frac{0.36}{0.60} = 0.60$$

Thus,

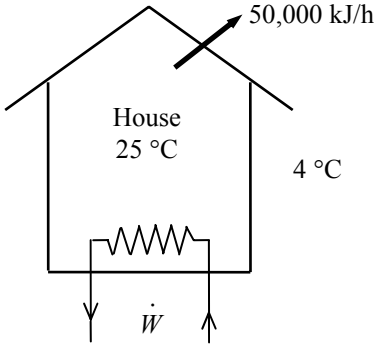
$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H} \longrightarrow T_H = T_L/(1-\eta_{\text{th,rev}}) = (530\text{ R})/0.40 = \mathbf{1325\ R}$$

8-23 A house is maintained at a specified temperature by electric resistance heaters. The reversible work for this heating process and irreversibility are to be determined.

Analysis The reversible work is the minimum work required to accomplish this process, and the irreversibility is the difference between the reversible work and the actual electrical work consumed. The actual power input is

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}} = \dot{Q}_H = 50{,}000\text{ kJ/h} = 13.89\text{ kW}$$

The COP of a reversible heat pump operating between the specified temperature limits is



$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L/T_H} = \frac{1}{1 - 277.15/298.15} = 14.20$$

Thus,

$$\dot{W}_{\text{rev,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,rev}}} = \frac{13.89\text{ kW}}{14.20} = \mathbf{0.978\ kW}$$

and

$$\dot{I} = \dot{W}_{\text{u,in}} - \dot{W}_{\text{rev,in}} = 13.89 - 0.978 = \mathbf{12.91\ kW}$$

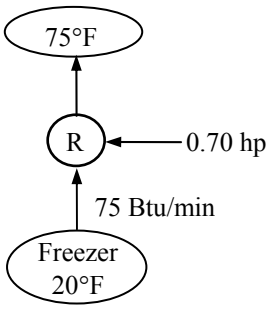
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8-24E A freezer is maintained at a specified temperature by removing heat from it at a specified rate. The power consumption of the freezer is given. The reversible power, irreversibility, and the second-law efficiency are to be determined.

Analysis (a) The reversible work is the minimum work required to accomplish this task, which is the work that a reversible refrigerator operating between the specified temperature limits would consume,

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H / T_L - 1} = \frac{1}{535/480 - 1} = 8.73$$

$$\dot{W}_{\text{rev,in}} = \frac{\dot{Q}_L}{COP_{\text{R,rev}}} = \frac{75\ \text{Btu/min}}{8.73}\left(\frac{1\ \text{hp}}{42.41\ \text{Btu/min}}\right) = \mathbf{0.20\ hp}$$



(b) The irreversibility is the difference between the reversible work and the actual electrical work consumed,

$$\dot{I} = \dot{W}_{\text{u,in}} - \dot{W}_{\text{rev,in}} = 0.70 - 0.20 = \mathbf{0.50\ hp}$$

(c) The second law efficiency is determined from its definition,

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_u} = \frac{0.20\ \text{hp}}{0.7\ \text{hp}} = \mathbf{28.9\%}$$

8-25 A geothermal power produces 5.1 MW power while the exergy destruction in the plant is 7.5 MW. The exergy of the geothermal water entering to the plant, the second-law efficiency of the plant, and the exergy of the heat rejected from the plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Water properties are used for geothermal water.

Analysis (*a*) The properties of geothermal water at the inlet of the plant and at the dead state are (Tables A-4 through A-6)

$$\left.\begin{matrix} T_1 = 150°\text{C} \\ x_1 = 0 \end{matrix}\right\} \begin{matrix} h_1 = 632.18\ \text{kJ/kg} \\ s_1 = 1.8418\ \text{kJ/kg.K} \end{matrix}$$

$$\left.\begin{matrix} T_0 = 25°\text{C} \\ x_0 = 0 \end{matrix}\right\} \begin{matrix} h_0 = 104.83\ \text{kJ/kg} \\ s_0 = 0.36723\ \text{kJ/kg.K} \end{matrix}$$

The exergy of geothermal water entering the plant is

$$\begin{aligned} \dot{X}_{\text{in}} &= \dot{m}\left[h_1 - h_0 - T_0(s_1 - s_0)\right] \\ &= (210\ \text{kg/s})\left[(632.18 - 104.83)\ \text{kJ/kg} - (25 + 273\ \text{K})(1.8418 - 0.36723)\text{kJ/kg.K}\right] \\ &= 18{,}460\ \text{kW} = \mathbf{18.46\ MW} \end{aligned}$$

(*b*) The second-law efficiency of the plant is the ratio of power produced to the exergy input to the plant

$$\eta_{II} = \frac{\dot{W}_{\text{out}}}{\dot{X}_{\text{in}}} = \frac{5100\ \text{kW}}{18{,}460\ \text{kW}} = 0.276 = \mathbf{27.6\%}$$

(*c*) The exergy of the heat rejected from the plant may be determined from an exergy balance on the plant

$$\dot{X}_{\text{heat,out}} = \dot{X}_{\text{in}} - \dot{W}_{\text{out}} - \dot{X}_{\text{dest}} = 18{,}460 - 5100 - 7500 = 5864\ \text{kW} = \mathbf{5.86\ MW}$$

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8-26 It is to be shown that the power produced by a wind turbine is proportional to the cube of the wind velocity and the square of the blade span diameter.

Analysis The power produced by a wind turbine is proportional to the kinetic energy of the wind, which is equal to the product of the kinetic energy of air per unit mass and the mass flow rate of air through the blade span area. Therefore,

$$\text{Wind power} = (\text{Efficiency})(\text{Kinetic energy})(\text{Mass flow rate of air})$$

$$= \eta_{\text{wind}} \frac{V^2}{2}(\rho A V) = \eta_{\text{wind}} \frac{V^2}{2} \rho \frac{\pi D^2}{4} V$$

$$= \eta_{\text{wind}} \rho \frac{\pi V^3 D^2}{8} = (\text{Constant}) V^3 D^2$$

which completes the proof that wind power is proportional to the cube of the wind velocity and to the square of the blade span diameter.

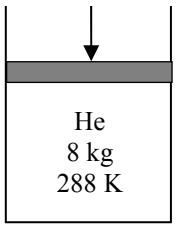
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Exergy Analysis of Closed Systems

8-27C Yes, it can. For example, the 1st law efficiency of a reversible heat engine operating between the temperature limits of 300 K and 1000 K is 70%. However, the second law efficiency of this engine, like all reversible devices, is 100%.

8-28 A fixed mass of helium undergoes a process from a specified state to another specified state. The increase in the useful energy potential of helium is to be determined.

Assumptions **1** At specified conditions, helium can be treated as an ideal gas.
2 Helium has constant specific heats at room temperature.



Properties The gas constant of helium is $R = 2.0769$ kJ/kg.K (Table A-1). The constant volume specific heat of helium is $c_v = 3.1156$ kJ/kg.K (Table A-2).

Analysis From the ideal-gas entropy change relation,

$$s_2 - s_1 = c_{v,\text{avg}} \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1}$$

$$= (3.1156\ \text{kJ/kg}\cdot\text{K}) \ln \frac{353\ \text{K}}{288\ \text{K}} + (2.0769\ \text{kJ/kg}\cdot\text{K}) \ln \frac{0.5\ \text{m}^3/\text{kg}}{3\ \text{m}^3/\text{kg}} = -3.087\ \text{kJ/kg}\cdot\text{K}$$

The increase in the useful potential of helium during this process is simply the increase in exergy,

$$\Phi_2 - \Phi_1 = -m\left[(u_1 - u_2) - T_0(s_1 - s_2) + P_0(v_1 - v_2)\right]$$
$$= -(8\ \text{kg})\{(3.1156\ \text{kJ/kg}\cdot\text{K})(288 - 353)\ \text{K} - (298\ \text{K})(3.087\ \text{kJ/kg}\cdot\text{K})$$
$$+ (100\ \text{kPa})(3 - 0.5)\text{m}^3/\text{kg}[\text{kJ/kPa}\cdot\text{m}^3]\}$$
$$= \mathbf{6980\ kJ}$$

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8-29E Air is expanded in an adiabatic closed system with an isentropic efficiency of 95%. The second law efficiency of the process is to be determined.

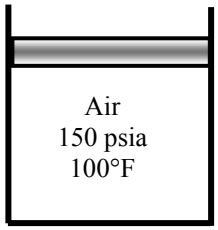
Assumptions **1** Kinetic and potential energy changes are negligible. **2** The process is adiabatic, and thus there is no heat transfer. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 0.240$ Btu/lbm·R, $c_v = 0.171$ Btu/lbm·R, $k = 1.4$, and $R = 0.06855$ Btu/lbm·R (Table A-2Ea).

Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

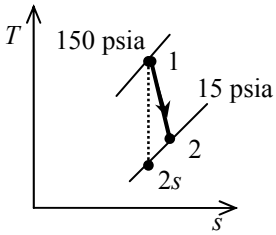
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-W_{b,\text{out}} = \Delta U = mc_v(T_2 - T_1)$$



The final temperature for the isentropic case is

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (560\,\text{R})\left(\frac{15\,\text{psia}}{150\,\text{psia}}\right)^{0.4/1.4} = 290.1\,\text{R}$$



The actual exit temperature from the isentropic relation is

$$\eta = \frac{T_1 - T_2}{T_1 - T_{2s}}$$

$$T_2 = T_1 - \eta(T_1 - T_{2s}) = 560 - (0.95)(560 - 290.1) = 303.6\,\text{R}$$

The boundary work output is

$$w_{b,\text{out}} = c_v(T_1 - T_2) = (0.171\,\text{Btu/lbm}\cdot\text{R})(560 - 303.6)\text{R} = 43.84\,\text{Btu/lbm}$$

The entropy change of air is

$$\Delta s_{\text{air}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (0.240\,\text{Btu/lbm}\cdot\text{R})\ln\frac{303.6\,\text{R}}{560\,\text{R}} - (0.06855\,\text{Btu/lbm}\cdot\text{R})\ln\frac{15\,\text{psia}}{150\,\text{psia}}$$

$$= 0.01091\,\text{Btu/lbm}\cdot\text{R}$$

The exergy difference between states 1 and 2 is

$$\phi_1 - \phi_2 = u_1 - u_2 + P_0(v_1 - v_2) - T_0(s_1 - s_2)$$

$$= c_v(T_1 - T_2) + P_0R\left(\frac{T_1}{P_1} - \frac{T_2}{P_2}\right) - T_0(s_1 - s_2)$$

$$= 43.84\,\text{Btu/lbm} + (14.7\,\text{psia})(0.06855\,\text{Btu/lbm}\cdot\text{R})\left(\frac{560\,\text{R}}{150\,\text{psia}} - \frac{303.6\,\text{R}}{15\,\text{psia}}\right) - (537\,\text{R})(-0.01091\,\text{Btu/lbm}\cdot\text{R})$$

$$= 33.07\,\text{Btu/lbm}$$

The useful work is determined from

$$w_u = w_{b,\text{out}} - w_{\text{surr}} = c_v(T_1 - T_2) - P_0(v_2 - v_1) = c_v(T_1 - T_2) - P_0R\left(\frac{T_2}{P_2} - \frac{T_1}{P_1}\right)$$

$$= 43.84\,\text{Btu/lbm} - (14.7\,\text{psia})(0.06855\,\text{Btu/lbm}\cdot\text{R})\left(\frac{303.6\,\text{R}}{15\,\text{psia}} - \frac{560\,\text{R}}{150\,\text{psia}}\right)$$

$$= 27.21\,\text{Btu/lbm}$$

The second law efficiency is then

$$\eta_{\text{II}} = \frac{w_u}{\Delta\phi} = \frac{27.21\,\text{Btu/lbm}}{33.07\,\text{Btu/lbm}} = \mathbf{0.823}$$

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8-30E Air and helium at specified states are considered. The gas with the higher exergy content is to be identified.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Air and helium are ideal gases with constant specific heats.

Properties The properties of air at room temperature are $c_p = 0.240$ Btu/lbm·R, $c_v = 0.171$ Btu/lbm·R, $k = 1.4$, and $R = 0.06855$ Btu/lbm·R = 0.3704 psia·ft^3/lbm·R. For helium, $c_p = 1.25$ Btu/lbm·R, $c_v = 0.753$ Btu/lbm·R, $k = 1.667$, and $R = 0.4961$ Btu/lbm·R= 2.6809 psia·ft^3/lbm·R. (Table A-2E).

Analysis The mass of air in the system is

$$m = \frac{P\mathcal{V}}{RT} = \frac{(100\ \text{psia})(15\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(710\ \text{R})} = 5.704\ \text{lbm}$$

Air
15 ft^3
100 psia
250°F

The entropy change of air between the given state and the dead state is

$$\begin{aligned} s - s_0 &= c_p \ln\frac{T}{T_0} - R\ln\frac{P}{P_0} \\ &= (0.240\ \text{Btu/lbm}\cdot\text{R})\ln\frac{710\ \text{R}}{537\ \text{R}} - (0.06855\ \text{Btu/lbm}\cdot\text{R})\ln\frac{100\ \text{psia}}{14.7\ \text{psia}} \\ &= -0.06441\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

The air's specific volumes at the given state and dead state are

$$\nu = \frac{RT}{P} = \frac{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(710\ \text{R})}{100\ \text{psia}} = 2.630\ \text{ft}^3/\text{lbm}$$

$$\nu_0 = \frac{RT_0}{P_0} = \frac{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(537\ \text{R})}{14.7\ \text{psia}} = 13.53\ \text{ft}^3/\text{lbm}$$

The specific closed system exergy of the air is then

$$\begin{aligned} \phi &= u - u_0 + P_0(\nu - \nu_0) - T_0(s - s_0) \\ &= c_\nu(T - T_0) + P_0(\nu - \nu_0) - T_0(s - s_0) \\ &= (0.171\ \text{Btu/lbm}\cdot\text{R})(300 - 77)\text{R} + (14.7\ \text{psia})(2.630 - 13.53)\text{ft}^3/\text{lbm}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &\quad - (537\ \text{R})(-0.06441)\ \text{Btu/lbm}\cdot\text{R} \\ &= 34.52\ \text{Btu/lbm} \end{aligned}$$

The total exergy available in the air for the production of work is then

$$\Phi = m\phi = (5.704\ \text{lbm})(34.52\ \text{Btu/lbm}) = \mathbf{197\ Btu}$$

We now repeat the calculations for helium:

$$m = \frac{P\mathcal{V}}{RT} = \frac{(60\ \text{psia})(20\ \text{ft}^3)}{(2.6809\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(660\ \text{R})} = 0.6782\ \text{lbm}$$

Helium
20 ft^3
60 psia
200°F

$$\begin{aligned} s - s_0 &= c_p \ln\frac{T}{T_0} - R\ln\frac{P}{P_0} \\ &= (1.25\ \text{Btu/lbm}\cdot\text{R})\ln\frac{660\ \text{R}}{537\ \text{R}} - (0.4961\ \text{Btu/lbm}\cdot\text{R})\ln\frac{60\ \text{psia}}{14.7\ \text{psia}} \\ &= -0.4400\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\nu = \frac{RT}{P} = \frac{(2.6809\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(660\ \text{R})}{60\ \text{psia}} = 29.49\ \text{ft}^3/\text{lbm}$$

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$$v_0 = \frac{RT_0}{P_0} = \frac{(2.6809\ \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(537\ \text{R})}{14.7\ \text{psia}} = 97.93\ \text{ft}^3/\text{lbm}$$

$$\begin{aligned}\phi &= u - u_0 + P_0(v - v_0) - T_0(s - s_0) \\ &= c_v(T - T_0) + P_0(v - v_0) - T_0(s - s_0) \\ &= (0.753\ \text{Btu/lbm} \cdot \text{R})(200 - 77)\text{R} + (14.7\ \text{psia})(29.49 - 97.93)\text{ft}^3/\text{lbm}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia} \cdot \text{ft}^3}\right) \\ &\quad - (537\ \text{R})(-0.4400)\ \text{Btu/lbm} \cdot \text{R} \\ &= 142.7\ \text{Btu/lbm}\end{aligned}$$

$$\Phi = m\phi = (0.6782\ \text{lbm})(142.7\ \text{Btu/lbm}) = \mathbf{96.8\ Btu}$$

Comparison of two results shows that the air system has a greater potential for the production of work.

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8-31 Steam and R-134a at the same states are considered. The fluid with the higher exergy content is to be identified.

Assumptions Kinetic and potential energy changes are negligible.

Analysis The properties of water at the given state and at the dead state are

$$\left.\begin{array}{l} P = 800\ \text{kPa} \\ T = 180°\text{C} \end{array}\right\} \begin{array}{l} u = 2594.7\ \text{kJ/kg} \\ v = 0.24720\ \text{m}^3/\text{kg} \quad \text{(Table A - 6)} \\ s = 6.7155\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ P_0 = 100\ \text{kPa} \end{array}\right\} \begin{array}{l} u_0 \cong u_{f@25°\text{C}} = 104.83\ \text{kJ/kg} \\ v_0 \cong v_{f@25°\text{C}} = 0.001003\ \text{m}^3/\text{kg} \quad \text{(Table A - 4)} \\ s_0 \cong s_{f@25°\text{C}} = 0.3672\ \text{kJ/kg}\cdot\text{K} \end{array}$$

Steam
1 kg
800 kPa
180°C

The exergy of steam is

$$\begin{aligned} \Phi &= m[u - u_0 + P_0(v - v_0) - T_0(s - s_0)] \\ &= (1\ \text{kg})\left[\begin{array}{l} (2594.7 - 104.83)\text{kJ/kg} + (100\ \text{kPa})(0.24720 - 0.001003)\text{m}^3/\text{kg}\left(\dfrac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ -(298\ \text{K})(6.7155 - 0.3672)\text{kJ/kg}\cdot\text{K} \end{array}\right] \\ &= \mathbf{622.7\ kJ} \end{aligned}$$

For R-134a;

$$\left.\begin{array}{l} P = 800\ \text{kPa} \\ T = 180°\text{C} \end{array}\right\} \begin{array}{l} u = 386.99\ \text{kJ/kg} \\ v = 0.044554\ \text{m}^3/\text{kg} \quad \text{(Table A - 13)} \\ s = 1.3327\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ P_0 = 100\ \text{kPa} \end{array}\right\} \begin{array}{l} u_0 \cong u_{f@25°\text{C}} = 85.85\ \text{kJ/kg} \\ v_0 \cong v_{f@25°\text{C}} = 0.0008286\ \text{m}^3/\text{kg} \quad \text{(Table A - 11)} \\ s_0 \cong s_{f@25°\text{C}} = 0.32432\ \text{kJ/kg}\cdot\text{K} \end{array}$$

R-134a
1 kg
800 kPa
180°C

$$\begin{aligned} \Phi &= m[u - u_0 + P_0(v - v_0) - T_0(s - s_0)] \\ &= (1\ \text{kg})\left[\begin{array}{l} (386.99 - 85.85)\text{kJ/kg} + (100\ \text{kPa})(0.044554 - 0.0008286)\text{m}^3/\text{kg}\left(\dfrac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ -(298\ \text{K})(1.3327 - 0.32432)\text{kJ/kg}\cdot\text{K} \end{array}\right] \\ &= \mathbf{5.02\ kJ} \end{aligned}$$

The steam can therefore has more work potential than the R-134a.

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8-32 A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled and condensed at constant pressure. The exergy of the refrigerant at the initial and final states, and the exergy destroyed during this process are to be determined.

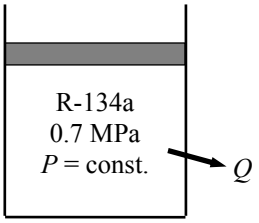
Assumptions The kinetic and potential energies are negligible.

Properties From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 0.7\ \text{MPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.034875\ \text{m}^3/\text{kg} \\ u_1 = 274.01\ \text{kJ/kg} \\ s_1 = 1.0256\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\ \text{MPa} \\ T_2 = 24°\text{C} \end{array}\right\} \begin{array}{l} v_2 \cong v_{f@24°\text{C}} = 0.0008261\ \text{m}^3/\text{kg} \\ u_2 \cong u_{f@24°\text{C}} = 84.44\ \text{kJ/kg} \\ s_2 \cong s_{f@24°\text{C}} = 0.31958\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_0 = 0.1\ \text{MPa} \\ T_0 = 24°\text{C} \end{array}\right\} \begin{array}{l} v_0 = 0.23718\ \text{m}^3/\text{kg} \\ u_0 = 251.84\ \text{kJ/kg} \\ s_0 = 1.1033\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis (*a*) From the closed system exergy relation,

$$\begin{aligned} X_1 = \Phi_1 &= m\{(u_1 - u_0) - T_0(s_1 - s_0) + P_0(v_1 - v_0)\} \\ &= (5\ \text{kg})\{(274.01 - 251.84)\ \text{kJ/kg} - (297\ \text{K})(1.0256 - 1.1033)\ \text{kJ/kg}\cdot\text{K} \\ &\quad + (100\ \text{kPa})(0.034875 - 0.23718)\text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)\} \\ &= \mathbf{125.1\ kJ} \end{aligned}$$

and

$$\begin{aligned} X_2 = \Phi_2 &= m\{(u_2 - u_0) - T_0(s_2 - s_0) + P_0(v_2 - v_0)\} \\ &= (5\ \text{kg})\{(84.44 - 251.84)\ \text{kJ/kg} - (297\ \text{K})(0.31958 - 1.1033)\ \text{kJ/kg}\cdot\text{K} \\ &\quad + (100\ \text{kPa})(0.0008261 - 0.23718)\text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)\} \\ &= \mathbf{208.6\ kJ} \end{aligned}$$

(*b*) The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}}_{\substack{\nearrow 0\ \text{(reversible)} \\ \text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$W_{\text{rev,in}} = X_2 - X_1 = 208.6 - 125.1 = 83.5\ \text{kJ}$$

Noting that the process involves only boundary work, the useful work input during this process is simply the boundary work in excess of the work done by the surrounding air,

$$\begin{aligned} W_{\text{u,in}} &= W_{\text{in}} - W_{\text{surr,in}} = W_{\text{in}} - P_0(V_1 - V_2) = P(V_1 - V_2) - P_0 m(v_1 - v_2) \\ &= m(P - P_0)(v_1 - v_2) \\ &= (5\ \text{kg})(700 - 100\ \text{kPa})(0.034875 - 0.0008261\ \text{m}^3/\text{kg})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 102.1\ \text{kJ} \end{aligned}$$

Knowing both the actual useful and reversible work inputs, the exergy destruction or irreversibility that is the difference between the two is determined from its definition to be

$$X_{\text{destroyed}} = I = W_{\text{u,in}} - W_{\text{rev,in}} = 102.1 - 83.5 = \mathbf{18.6\ kJ}$$

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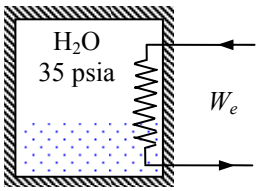
8-33E An insulated rigid tank contains saturated liquid-vapor mixture of water at a specified pressure. An electric heater inside is turned on and kept on until all the liquid is vaporized. The exergy destruction and the second-law efficiency are to be determined.

Assumptions Kinetic and potential energies are negligible.

Properties From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 35 \text{ psia} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.01708 + 0.25 \times (11.901 - 0.01708) = 2.9880 \text{ ft}^3/\text{lbm} \\ u_1 = u_f + x_1 u_{fg} = 227.92 + 0.25 \times 862.19 = 443.47 \text{ Btu/lbm} \\ s_1 = s_f + x_1 s_{fg} = 0.38093 + 0.25 \times 1.30632 = 0.70751 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} v_2 = v_1 \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} u_2 = u_{g@v_g = 2.9880 \text{ ft}^3/\text{lbm}} = 1110.9 \text{ Btu/lbm} \\ s_2 = s_{g@v_g = 2.9880 \text{ ft}^3/\text{lbm}} = 1.5692 \text{ Btu/lbm} \cdot \text{R} \end{array}$$



Analysis (*a*) The irreversibility can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the tank, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

Substituting,

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = m T_0 (s_2 - s_1) \\ &= (6 \text{ lbm})(535 \text{ R})(1.5692 - 0.70751)\text{Btu/lbm} \cdot \text{R} = \mathbf{2766\ Btu} \end{aligned}$$

(*b*) Noting that V = constant during this process, the W and W_u are identical and are determined from the energy balance on the closed system energy equation,

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U = m(u_2 - u_1)$$

or,

$$W_{\text{e,in}} = (6 \text{ lbm})(1110.9 - 443.47)\text{Btu/lbm} = 4005 \text{ Btu}$$

Then the reversible work during this process and the second-law efficiency become

$$W_{\text{rev,in}} = W_{\text{u,in}} - X_{\text{destroyed}} = 4005 - 2766 = 1239 \text{ Btu}$$

Thus,

$$\eta_{\text{II}} = \frac{W_{\text{rev}}}{W_u} = \frac{1239 \text{ Btu}}{4005 \text{ Btu}} = \mathbf{30.9\%}$$

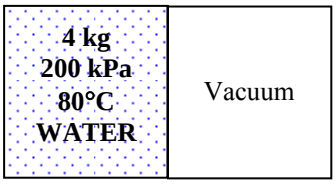
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8-34 A rigid tank is divided into two equal parts by a partition. One part is filled with compressed liquid while the other side is evacuated. The partition is removed and water expands into the entire tank. The exergy destroyed during this process is to be determined.

Assumptions Kinetic and potential energies are negligible.

Analysis The properties of the water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ T_1 = 80°\text{C} \end{array}\right\} \begin{array}{l} v_1 \cong v_{f@80°\text{C}} = 0.001029\ \text{m}^3/\text{kg} \\ u_1 \cong u_{f@80°\text{C}} = 334.97\ \text{kJ/kg} \\ s_1 \cong s_{f@80°\text{C}} = 1.0756\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Noting that $v_2 = 2v_1 = 2 \times 0.001029 = 0.002058\ \text{m}^3/\text{kg}$,

$$\left.\begin{array}{l} P_2 = 40\ \text{kPa} \\ v_2 = 0.002058\ \text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{v_2 - v_f}{v_{fg}} = \dfrac{0.002058 - 0.001026}{3.9933 - 0.001026} = 0.0002584 \\ u_2 = u_f + x_2 u_{fg} = 317.58 + 0.0002584 \times 2158.8 = 318.14\ \text{kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 1.0261 + 0.0002584 \times 6.6430 = 1.0278\ \text{kJ/kg}\cdot\text{K} \end{array}$$

Taking the direction of heat transfer to be *to* the tank, the energy balance on this closed system becomes

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1)$$

or

$$Q_{\text{in}} = (4\ \text{kg})(318.14 - 334.97)\text{kJ/kg} = -67.30\ \text{kJ} \ \rightarrow\ Q_{\text{out}} = 67.30\ \text{kJ}$$

The irreversibility can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on an *extended system* that includes the tank and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

$$S_{\text{gen}} = m(s_2 - s_1) + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0 \left(m(s_2 - s_1) + \frac{Q_{\text{out}}}{T_{\text{surr}}} \right)$$

$$= (298\ \text{K}) \left[(4\ \text{kg})(1.0278 - 1.0756)\text{kJ/kg}\cdot\text{K} + \frac{67.30\ \text{kJ}}{298\ \text{K}} \right]$$

$$\mathbf{= 10.3\ kJ}$$

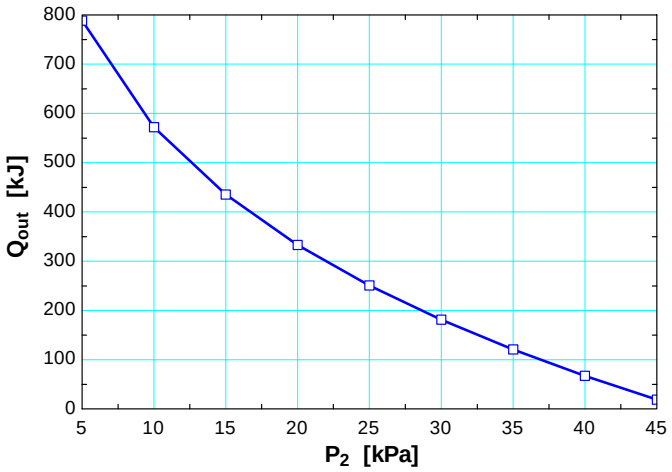
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8-35 Problem 8-34 is reconsidered. The effect of final pressure in the tank on the exergy destroyed during the process is to be investigated.

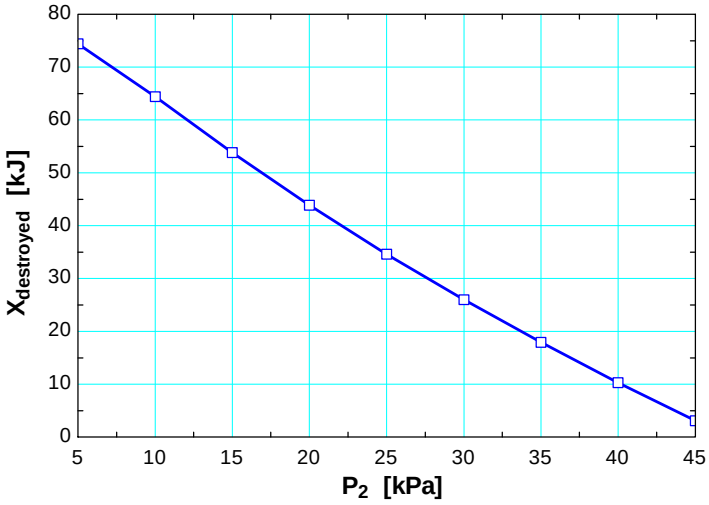
Analysis The problem is solved using EES, and the solution is given below.

```
T_1=80 [C]
P_1=200 [kPa]
m=4 [kg]
P_2=40 [kPa]
T_o=25 [C]
P_o=100 [kPa]
T_surr = T_o

"Conservation of energy for closed system is:"
E_in - E_out = DELTAE
DELTAE = m*(u_2 - u_1)
E_in=0
E_out= Q_out
u_1 =intenergy(steam_iapws,P=P_1,T=T_1)
v_1 =volume(steam_iapws,P=P_1,T=T_1)
s_1 =entropy(steam_iapws,P=P_1,T=T_1)
v_2 = 2*v_1
u_2 = intenergy(steam_iapws, v=v_2,P=P_2)
s_2 = entropy(steam_iapws, v=v_2,P=P_2)
S_in -S_out+S_gen=DELTAS_sys
S_in=0 [kJ/K]
S_out=Q_out/(T_surr+273)
DELTAS_sys=m*(s_2 - s_1)
X_destroyed = (T_o+273)*S_gen
```



P_2 [kPa]	$X_{destroyed}$ [kJ]	Q_{out} [kJ]
5	74.41	788.4
10	64.4	571.9
15	53.8	435.1
20	43.85	332.9
25	34.61	250.5
30	25.99	181
35	17.91	120.7
40	10.3	67.15
45	3.091	18.95



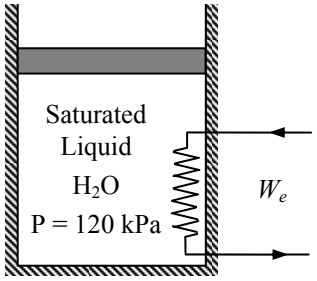
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8-36 An insulated cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically at constant pressure. The minimum work by which this process can be accomplished and the exergy destroyed are to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis (a) From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 120\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} u_1 = u_{f@120\,\text{kPa}} = 439.27\ \text{kJ/kg} \\ v_1 = v_{f@120\,\text{kPa}} = 0.001047\ \text{m}^3/\text{kg} \\ h_1 = h_{f@120\,\text{kPa}} = 439.36\ \text{kJ/kg} \\ s_1 = s_{f@120\,\text{kPa}} = 1.3609\ \text{kJ/kg}\cdot\text{K} \end{array}$$



The mass of the steam is

$$m = \frac{V}{v_1} = \frac{0.008\ \text{m}^3}{0.001047\ \text{m}^3/\text{kg}} = 7.639\ \text{kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U$$

$$W_{\text{e,in}} = m(h_2 - h_1)$$

since $\Delta U + W_{\text{b}} = \Delta H$ during a constant pressure quasi-equilibrium process. Solving for h_2,

$$h_2 = h_1 + \frac{W_{e,in}}{m} = 439.36 + \frac{1400\ \text{kJ}}{7.639\ \text{kg}} = 622.63\ \text{kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_2 = 120\ \text{kPa} \\ h_2 = 622.63\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{h_2 - h_f}{h_{fg}} = \dfrac{622.63 - 439.36}{2243.7} = 0.08168 \\ s_2 = s_f + x_2 s_{fg} = 1.3609 + 0.08168 \times 5.93687 = 1.8459\ \text{kJ/kg}\cdot\text{K} \\ u_2 = u_f + x_2 u_{fg} = 439.24 + 0.08168 \times 2072.4 = 608.52\ \text{kJ/kg} \\ v_2 = v_f + x_2 v_{fg} = 0.001047 + 0.08168 \times (1.4285 - 0.001047) = 0.1176\ \text{m}^3/\text{kg} \end{array}$$

The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}} \rightarrow W_{\text{rev,in}} = X_2 - X_1$$

Substituting the closed system exergy relation, the reversible work input during this process is determined to be

$$\begin{aligned} W_{\text{rev,in}} &= -m\left[(u_1 - u_2) - T_0(s_1 - s_2) + P_0(v_1 - v_2)\right] \\ &= -(7.639\ \text{kg})\{(439.27 - 608.52)\ \text{kJ/kg} - (298\ \text{K})(1.3609 - 1.8459)\ \text{kJ/kg}\cdot\text{K} \\ &\quad + (100\ \text{kPa})(0.001047 - 0.1176)\text{m}^3/\text{kg}[1\ \text{kJ}/1\ \text{kPa}\cdot\text{m}^3]\} \\ &= \mathbf{278\ kJ} \end{aligned}$$

(b) The exergy destruction (or irreversibility) associated with this process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the cylinder, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = mT_0(s_2 - s_1) = (298\ \text{K})(7.639\ \text{kg})(1.8459 - 1.3609)\text{kJ/kg}\cdot\text{K} = \mathbf{1104\ kJ}$$

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8-37 Problem 8-36 is reconsidered. The effect of the amount of electrical work on the minimum work and the exergy destroyed is to be investigated.

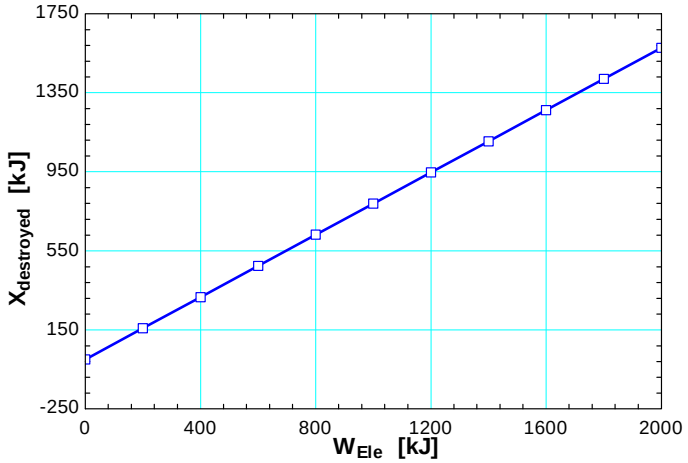
Analysis The problem is solved using EES, and the solution is given below.

```
x_1=0
P_1=120 [kPa]
V=8 [L]
P_2=P_1
{W_Ele = 1400 [kJ]}
T_o=25 [C]
P_o=100 [kPa]

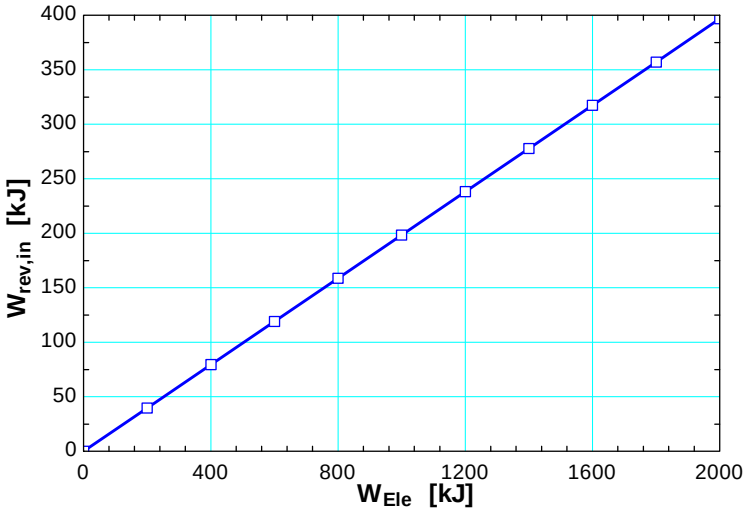
"Conservation of energy for closed system is:"
E_in - E_out = DELTAE
DELTAE = m*(u_2 - u_1)
E_in=W_Ele
E_out= W_b
W_b = m*P_1*(v_2-v_1)
u_1 =intenergy(steam_iapws,P=P_1,x=x_1)
v_1 =volume(steam_iapws,P=P_1,x=x_1)
s_1 =entropy(steam_iapws,P=P_1,x=x_1)
u_2 = intenergy(steam_iapws, v=v_2,P=P_2)
s_2 = entropy(steam_iapws, v=v_2,P=P_2)
m=V*convert(L,m^3)/v_1
W_rev_in=m*(u_2 - u_1 -(T_o+273.15)
            *(s_2-s_1)+P_o*(v_2-v_1))

"Entropy Balance:"
S_in - S_out+S_gen = DELTAS_sys
DELTAS_sys = m*(s_2 - s_1)
S_in=0 [kJ/K]
S_out= 0 [kJ/K]

"The exergy destruction or irreversibility is:"
X_destroyed = (T_o+273.15)*S_gen
```



W_{Ele} [kJ]	$W_{rev,in}$ [kJ]	$X_{destroyed}$ [kJ]
0	0	0
200	39.68	157.8
400	79.35	315.6
600	119	473.3
800	158.7	631.1
1000	198.4	788.9
1200	238.1	946.7
1400	277.7	1104
1600	317.4	1262
1800	357.1	1420
2000	396.8	1578



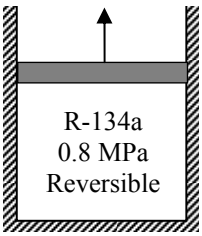
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8-38 An insulated cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant expands in a reversible manner until the pressure drops to a specified value. The change in the exergy of the refrigerant during this process and the reversible work are to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

Analysis This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 0.8\ \text{MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@0.8\ \text{MPa}} = 0.02562\ \text{m}^3/\text{kg} \\ u_1 = u_{g@0.8\ \text{MPa}} = 246.79\ \text{kJ/kg} \\ s_1 = s_{g@0.8\ \text{MPa}} = 0.9183\ \text{kJ/kg}\cdot\text{K} \end{array}$$



The mass of the refrigerant is

$$m = \frac{V}{v_1} = \frac{0.05\ \text{m}^3}{0.02562\ \text{m}^3/\text{kg}} = 1.952\ \text{kg}$$

$$\left.\begin{array}{l} P_2 = 0.2\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{0.9183 - 0.15457}{0.78316} = 0.9753 \\ v_2 = v_f + x_2 v_{fg} = 0.0007533 + 0.099867 \times (0.099867 - 0.0007533) = 0.09741\ \text{m}^3/\text{kg} \\ u_2 = u_f + x_2 u_{fg} = 38.28 + 0.9753 \times 186.21 = 219.88\ \text{kJ/kg} \end{array}$$

The reversible work output, which represents the maximum work output $W_{\text{rev,out}}$ can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$\begin{aligned} -W_{\text{rev,out}} &= X_2 - X_1 \\ W_{\text{rev,out}} &= X_1 - X_2 \\ &= \Phi_1 - \Phi_2 \end{aligned}$$

Therefore, the change in exergy and the reversible work are identical in this case. Using the definition of the closed system exergy and substituting, the reversible work is determined to be

$$\begin{aligned} W_{\text{rev,out}} &= \Phi_1 - \Phi_2 = m\left[(u_1 - u_2) - T_0 (s_1 - s_2)^{\nearrow 0} + P_0 (v_1 - v_2)\right] = m\left[(u_1 - u_2) + P_0 (v_1 - v_2)\right] \\ &= (1.952\ \text{kg})[(246.79 - 219.88)\ \text{kJ/kg} + (100\ \text{kPa})(0.02562 - 0.09741)\text{m}^3/\text{kg}[\text{kJ/kPa}\cdot\text{m}^3]] \\ &= \mathbf{38.5\ kJ} \end{aligned}$$

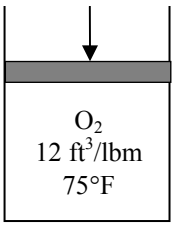
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8-39E Oxygen gas is compressed from a specified initial state to a final specified state. The reversible work and the increase in the exergy of the oxygen during this process are to be determined.

Assumptions At specified conditions, oxygen can be treated as an ideal gas with constant specific heats.

Properties The gas constant of oxygen is $R = 0.06206$ Btu/lbm.R (Table A-1E). The constant-volume specific heat of oxygen at the average temperature is

$$T_{\text{avg}} = (T_1 + T_2)/2 = (75 + 525)/2 = 300°\text{F} \longrightarrow c_{v,\text{avg}} = 0.164\ \text{Btu/lbm}\cdot\text{R}$$



Analysis The entropy change of oxygen is

$$\begin{aligned} s_2 - s_1 &= c_{v,\text{avg}} \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{v_2}{v_1}\right) \\ &= (0.164\ \text{Btu/lbm}\cdot\text{R})\ln\left(\frac{985\ \text{R}}{535\ \text{R}}\right) + (0.06206\ \text{Btu/lbm}\cdot\text{R})\ln\left(\frac{1.5\ \text{ft}^3/\text{lbm}}{12\ \text{ft}^3/\text{lbm}}\right) \\ &= -0.02894\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} \quad \underbrace{-X_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}} \ \rightarrow\ W_{\text{rev,in}} = X_2 - X_1$$

Therefore, the change in exergy and the reversible work are identical in this case. Substituting the closed system exergy relation, the reversible work input during this process is determined to be

$$\begin{aligned} w_{\text{rev,in}} &= \phi_2 - \phi_1 = -[(u_1 - u_2) - T_0(s_1 - s_2) + P_0(v_1 - v_2)] \\ &= -\{(0.164\ \text{Btu/lbm}\cdot\text{R})(535 - 985)\text{R} - (535\ \text{R})(0.02894\ \text{Btu/lbm}\cdot\text{R}) \\ &\quad + (14.7\ \text{psia})(12 - 1.5)\text{ft}^3/\text{lbm}[\text{Btu}/5.4039\ \text{psia}\cdot\text{ft}^3]\} \\ &= \mathbf{60.7\ Btu/lbm} \end{aligned}$$

Also, the increase in the exergy of oxygen is

$$\phi_2 - \phi_1 = w_{\text{rev,in}} = \mathbf{60.7\ \ Btu/lbm}$$

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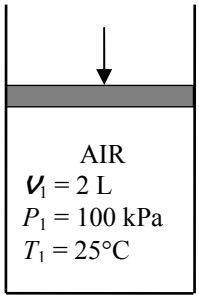
8-40 A cylinder initially contains air at atmospheric conditions. Air is compressed to a specified state and the useful work input is measured. The exergy of the air at the initial and final states, and the minimum work input to accomplish this compression process, and the second-law efficiency are to be determined

Assumptions **1** Air is an ideal gas with constant specific heats. **2** The kinetic and potential energies are negligible.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). The specific heats of air at the average temperature of (298+423)/2=360 K are $c_p = 1.009$ kJ/kg·K and $c_v = 0.722$ kJ/kg·K (Table A-2).

Analysis (a) We realize that $X_1 = \Phi_1 = \mathbf{0}$ since air initially is at the dead state. The mass of air is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(100\ \text{kPa})(0.002\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\ \text{K})} = 0.00234\ \text{kg}$$



Also,

$$\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1} \longrightarrow V_2 = \frac{P_1 T_2}{P_2 T_1} V_1 = \frac{(100\ \text{kPa})(423\ \text{K})}{(600\ \text{kPa})(298\ \text{K})}(2\ \text{L}) = 0.473\ \text{L}$$

and

$$\begin{aligned} s_2 - s_0 &= c_{p,\text{avg}} \ln\frac{T_2}{T_0} - R\ln\frac{P_2}{P_0} \\ &= (1.009\ \text{kJ/kg}\cdot\text{K})\ln\frac{423\ \text{K}}{298\ \text{K}} - (0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{600\ \text{kPa}}{100\ \text{kPa}} \\ &= -0.1608\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Thus, the exergy of air at the final state is

$$\begin{aligned} X_2 = \Phi_2 &= m\left[c_{v,\text{avg}}(T_2 - T_0) - T_0(s_2 - s_0)\right] + P_0(V_2 - V_0) \\ &= (0.00234\ \text{kg})\left[(0.722\ \text{kJ/kg}\cdot\text{K})(423 - 298)\text{K} - (298\ \text{K})(-0.1608\ \text{kJ/kg}\cdot\text{K})\right] \\ &\quad + (100\ \text{kPa})(0.000473 - 0.002)\text{m}^3[\text{kJ/m}^3\cdot\text{kPa}] \\ &= \mathbf{0.171\ kJ} \end{aligned}$$

(*b*) The minimum work input is the reversible work input, which can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}}$$

$$\begin{aligned} W_{\text{rev,in}} &= X_2 - X_1 \\ &= 0.171 - 0 = \mathbf{0.171 kJ} \end{aligned}$$

(c) The second-law efficiency of this process is

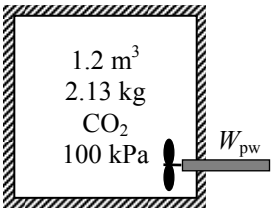
$$\eta_{\text{II}} = \frac{W_{\text{rev,in}}}{W_{\text{u,in}}} = \frac{0.171\ \text{kJ}}{1.2\ \text{kJ}} = \mathbf{14.3\%}$$

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8-41 An insulated tank contains CO_2 gas at a specified pressure and volume. A paddle-wheel in the tank stirs the gas, and the pressure and temperature of CO_2 rises. The actual paddle-wheel work and the minimum paddle-wheel work by which this process can be accomplished are to be determined.

Assumptions **1** At specified conditions, CO_2 can be treated as an ideal gas with constant specific heats at the average temperature. **2** The surroundings temperature is 298 K.

Properties The gas constant of CO_2 is 0.1889 kJ/kg·K (Table A-1)



Analysis (a) The initial and final temperature of CO_2 are

$$T_1 = \frac{P_1 V_1}{mR} = \frac{(100\,\text{kPa})(1.2\,\text{m}^3)}{(2.13\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 298.2\,\text{K}$$

$$T_2 = \frac{P_2 V_2}{mR} = \frac{(120\,\text{kPa})(1.2\,\text{m}^3)}{(2.13\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 357.9\,\text{K}$$

$$T_{\text{avg}} = (T_1 + T_2)/2 = (298.2 + 357.9)/2 = 328\,\text{K} \longrightarrow c_{v,\text{avg}} = 0.684\,\text{kJ/kg}\cdot\text{K} \quad \text{(Table A-2}b\text{)}$$

The actual paddle-wheel work done is determined from the energy balance on the CO gas in the tank,

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} = \Delta U = mc_v(T_2 - T_1)$$

or

$$W_{\text{pw,in}} = (2.13\,\text{kg})(0.684\,\text{kJ/kg}\cdot\text{K})(357.9 - 298.2)\text{K} = \mathbf{87.0\,kJ}$$

(b) The minimum paddle-wheel work with which this process can be accomplished is the reversible work,

which can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}} \rightarrow W_{\text{rev,in}} = X_2 - X_1$$

Substituting the closed system exergy relation, the reversible work input for this process is determined to be

$$\begin{aligned} W_{\text{rev,in}} &= m\left[(u_2 - u_1) - T_0(s_2 - s_1) + P_0(v_2^{\cancelto{0}{} } - v_1)\right] \\ &= m\left[c_{v,\text{avg}}(T_2 - T_1) - T_0(s_2 - s_1)\right] \\ &= (2.13\,\text{kg})\left[(0.684\,\text{kJ/kg}\cdot\text{K})(357.9 - 298.2)\text{K} - (298.2)(0.1253\,\text{kJ/kg}\cdot\text{K})\right] \\ &= \mathbf{7.74\ kJ} \end{aligned}$$

since

$$s_2 - s_1 = c_{v,\text{avg}} \ln\frac{T_2}{T_1} + R\ln\frac{v_2}{v_1}^{\cancelto{0}{}} = (0.684\,\text{kJ/kg}\cdot\text{K})\ln\left(\frac{357.9\,\text{K}}{298.2\,\text{K}}\right) = 0.1253\,\text{kJ/kg}\cdot\text{K}$$

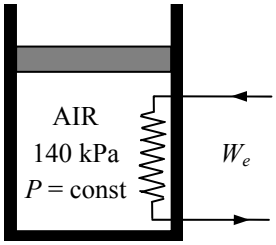
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8-42 An insulated cylinder initially contains air at a specified state. A resistance heater inside the cylinder is turned on, and air is heated for 10 min at constant pressure. The exergy destruction during this process is to be determined.

Assumptions Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

Analysis The mass of the air and the electrical work done during this process are



$$m = \frac{P_1 V_1}{RT_1} = \frac{(140\,\text{kPa})(0.020\,\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\,\text{K})} = 0.03250\,\text{kg}$$

$$W_e = \dot{W}_e \Delta t = (0.100\,\text{kJ/s})(10\times 60\,\text{s}) = 60\,\text{kJ}$$

Also,

$$T_1 = 300\,\text{K} \longrightarrow h_1 = 300.19\,\text{kJ/kg} \quad \text{and} \quad s_1^{\text{o}} = 1.70202\,\text{kJ/kg}\cdot\text{K}$$

The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U$$

$$W_{\text{e,in}} = m(h_2 - h_1)$$

since $\Delta U + W_{\text{b}} = \Delta H$ during a constant pressure quasi-equilibrium process. Thus,

$$h_2 = h_1 + \frac{W_{\text{e,in}}}{m} = 300.19 + \frac{60\,\text{kJ}}{0.03250\,\text{kg}} = 2146.3\,\text{kJ/kg} \xrightarrow{\text{Table A–17}} \begin{matrix} T_2 = 1915\,\text{K} \\ s_2^{\text{o}} = 3.7452\,\text{kJ/kg}\cdot\text{K} \end{matrix}$$

Also,

$$s_2 - s_1 = s_2^{\text{o}} - s_1^{\text{o}} - R\ln\left(\frac{P_2}{P_1}\right)^{\nearrow 0} = s_2^{\text{o}} - s_1^{\text{o}} = 3.7452 - 1.70202 = 2.0432\,\text{kJ/kg}\cdot\text{K}$$

The exergy destruction (or irreversibility) associated with this process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the cylinder, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = mT_0(s_2 - s_1) = (0.03250\,\text{kg})(300\,\text{K})(2.0432\,\text{kJ/kg}\cdot\text{K}) = \mathbf{19.9\ kJ}$$

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8-43 One side of a partitioned insulated rigid tank contains argon gas at a specified temperature and pressure while the other side is evacuated. The partition is removed, and the gas fills the entire tank. The exergy destroyed during this process is to be determined.

Assumptions Argon is an ideal gas with constant specific heats, and thus ideal gas relations apply.

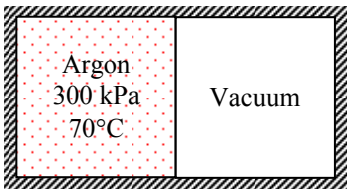
Properties The gas constant of argon is $R = 0.2081$ kJ/kg.K (Table A-1).

Analysis Taking the entire rigid tank as the system, the energy balance can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1)$$

$$u_2 = u_1 \quad \rightarrow \quad T_2 = T_1$$



since $u = u(T)$ for an ideal gas.

The exergy destruction (or irreversibility) associated with this process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the entire tank, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = m\left(c_{v,\text{avg}} \ln \frac{T_2}{T_1}^{\nearrow 0} + R \ln \frac{v_2}{v_1} \right) = mR \ln \frac{v_2}{v_1}$$

$$= (3\,\text{kg})(0.2081\,\text{kJ/kg} \cdot \text{K}) \ln(2) = 0.433\,\text{kJ/K}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = mT_0(s_2 - s_1) = (298\,\text{K})(0.433\,\text{kJ/K}) = \mathbf{129\ kJ}$$

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8-44E A hot copper block is dropped into water in an insulated tank. The final equilibrium temperature of the tank and the work potential wasted during this process are to be determined.

Assumptions **1** Both the water and the copper block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer.

Properties The density and specific heat of water at the anticipated average temperature of 90°F are $\rho = 62.1$ lbm/ft^3 and $c_p = 1.00$ Btu/lbm.°F. The specific heat of copper at the anticipated average temperature of 100°F is $c_p = 0.0925$ Btu/lbm.°F (Table A-3E).

Analysis (*a*) We take the entire contents of the tank, water + copper block, as the *system*, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$



or

$$\Delta U_{\text{Cu}} + \Delta U_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{Cu}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

where

$$m_w = \rho V = (62.1\ \text{lbm/ft}^3)(1.2\ \text{ft}^3) = 74.52\ \text{lbm}$$

Substituting,

$$0 = (55\ \text{lbm})(0.0925\ \text{Btu/lbm}\cdot°\text{F})(T_2 - 180°\text{F}) + (74.52\ \text{lbm})(1.0\ \text{Btu/lbm}\cdot°\text{F})(T_2 - 75°\text{F})$$

$$T_2 = \mathbf{81.7°F} = 541.7\ \text{R}$$

(*b*) The wasted work potential is equivalent to the exergy destruction (or irreversibility), and it can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the system, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = \Delta S_{\text{water}} + \Delta S_{\text{copper}}$$

where

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (55\ \text{lbm})(0.092\ \text{Btu/lbm}\cdot\text{R}) \ln\left(\frac{541.7\ \text{R}}{640\ \text{R}}\right) = -0.8483\ \text{Btu/R}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (74.52\ \text{lbm})(1.0\ \text{Btu/lbm}\cdot\text{R}) \ln\left(\frac{541.7\ \text{R}}{535\ \text{R}}\right) = 0.9250\ \text{Btu/R}$$

Substituting,

$$X_{\text{destroyed}} = (535\ \text{R})(-0.8483 + 0.9250)\text{Btu/R} = \mathbf{43.1\ Btu}$$

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8-45 A hot iron block is dropped into water in an insulated tank that is stirred by a paddle-wheel. The mass of the iron block and the exergy destroyed during this process are to be determined. √

Assumptions **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer.

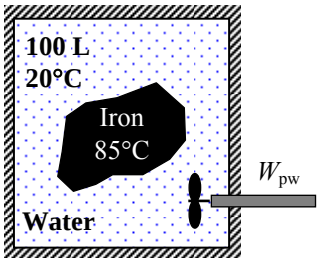
Properties The density and specific heat of water at 25°C are ρ = 997 kg/m^3 and c_p = 4.18 kJ/kg.°F. The specific heat of iron at room temperature (the only value available in the tables) is c_p = 0.45 kJ/kg.°C (Table A-3).

Analysis We take the entire contents of the tank, water + iron block, as the system, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} = \Delta U = \Delta U_{\text{iron}} + \Delta U_{\text{water}}$$

$$W_{\text{pw,in}} = [mc(T_2 - T_1)]_{\text{iron}} + [mc(T_2 - T_1)]_{\text{water}}$$



where

$$m_{\text{water}} = \rho V = (997\ \text{kg/m}^3)(0.1\ \text{m}^3) = 99.7\ \text{kg}$$

$$W_{\text{pw}} = \dot{W}_{\text{pw,in}}\,\Delta t = (0.2\ \text{kJ/s})(20 \times 60\ \text{s}) = 240\ \text{kJ}$$

Substituting,

$$240\ \text{kJ} = m_{\text{iron}}\,(0.45\ \text{kJ/kg}\cdot°\text{C})(24 - 85)°\text{C} + (99.7\ \text{kg})(4.18\ \text{kJ/kg}\cdot°\text{C})(24 - 20)°\text{C}$$

$$m_{\text{iron}} = \mathbf{52.0\ kg}$$

(*b*) The exergy destruction (or irreversibility) can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the system, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = \Delta S_{\text{iron}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (52.0\ \text{kg})(0.45\ \text{kJ/kg}\cdot\text{K}) \ln\left(\frac{297\ \text{K}}{358\ \text{K}}\right) = -4.371\ \text{kJ/K}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (99.7\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K}) \ln\left(\frac{297\ \text{K}}{293\ \text{K}}\right) = 5.651\ \text{kJ/K}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293\ \text{K})(-4.371 + 5.651)\ \text{kJ/K} = \mathbf{375.0\ kJ}$$

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8-46 An iron block and a copper block are dropped into a large lake where they cool to lake temperature. The amount of work that could have been produced is to be determined.

Assumptions **1** The iron and copper blocks and water are incompressible substances with constant specific heats at room temperature. **2** Kinetic and potential energies are negligible.

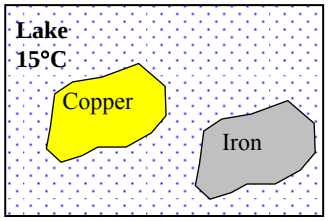
Properties The specific heats of iron and copper at room temperature are $c_{p,\text{ iron}} = 0.45$ kJ/kg.°C and $c_{p,\text{copper}} = 0.386$ kJ/kg.°C (Table A-3).

Analysis The thermal-energy capacity of the lake is very large, and thus the temperatures of both the iron and the copper blocks will drop to the lake temperature (15°C) when the thermal equilibrium is established.

We take both the iron and the copper blocks as the system, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = \Delta U_{\text{iron}} + \Delta U_{\text{copper}}$$



or,

$$Q_{out} = [mc(T_1 - T_2)]_{\text{iron}} + [mc(T_1 - T_2)]_{copper}$$

Substituting,

$$\begin{aligned} Q_{\text{out}} &= (50\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})(353-288)\text{K} + (20\text{ kg})(0.386\text{ kJ/kg}\cdot\text{K})(353-288)\text{K} \\ &= 1964\text{ kJ} \end{aligned}$$

The work that could have been produced is equal to the wasted work potential. It is equivalent to the exergy destruction (or irreversibility), and it can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation is determined from an entropy balance on an *extended system* that includes the blocks and the water in their immediate surroundings so that the boundary temperature of the extended system is the temperature of the lake water at all times,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{system}} = \Delta S_{\text{iron}} + \Delta S_{\text{copper}}$$

$$S_{\text{gen}} = \Delta S_{\text{iron}} + \Delta S_{\text{copper}} + \frac{Q_{\text{out}}}{T_{\text{lake}}}$$

where

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (50\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288\text{ K}}{353\text{ K}}\right) = -4.579\text{ kJ/K}$$

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (20\text{ kg})(0.386\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288\text{ K}}{353\text{ K}}\right) = -1.571\text{ kJ/K}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293\text{ K})\left(-4.579 - 1.571 + \frac{1964\text{ kJ}}{288\text{ K}}\right)\text{kJ/K} = \mathbf{196\ kJ}$$

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8-47E A rigid tank is initially filled with saturated mixture of R-134a. Heat is transferred to the tank from a source until the pressure inside rises to a specified value. The amount of heat transfer to the tank from the source and the exergy destroyed are to be determined.

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There is no heat transfer with the environment.

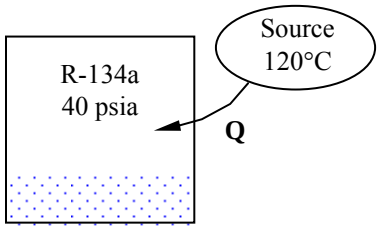
Properties From the refrigerant tables (Tables A-11E through A-13E),

$$\left.\begin{array}{l} P_1 = 40\ \text{psia} \\ x_1 = 0.55 \end{array}\right\} \begin{array}{l} u_1 = u_f + x_1 u_{fg} = 21.246 + 0.55 \times 77.307 = 63.76\ \text{Btu/lbm} \\ s_1 = s_f + x_1 s_{fg} = 0.04688 + 0.55 \times 0.17580 = 0.1436\ \text{Btu/lbm} \cdot \text{R} \\ \nu_1 = \nu_f + x_1 \nu_{fg} = 0.01232 + 0.55 \times 1.16368 = 0.65234\ \text{ft}^3/\text{lbm} \end{array}$$

$$\left.\begin{array}{l} P_2 = 60\ \text{psia} \\ (\nu_2 = \nu_1) \end{array}\right\} \begin{array}{l} x_2 = \dfrac{\nu_2 - \nu_f}{\nu_{fg}} = \dfrac{0.65234 - 0.01270}{0.79361 - 0.01270} = 0.8191 \\ s_2 = s_f + x_2 s_{fg} = 0.06029 + 0.8191 \times 0.16098 = 0.1922\ \text{Btu/lbm} \cdot \text{R} \\ u_2 = u_f + x_2 u_{fg} = 27.939 + 0.8191 \times 73.360 = 88.03\ \text{Btu/lbm} \end{array}$$

Analysis (*a*) The mass of the refrigerant is

$$m = \frac{V}{\nu_1} = \frac{12\ \text{ft}^3}{0.65234\ \text{ft}^3/\text{lbm}} = 18.40\ \text{lbm}$$



We take the tank as the system, which is a closed system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1)$$

Substituting,

$$Q_{\text{in}} = m(u_2 - u_1) = (18.40\ \text{lbm})(88.03 - 63.76)\ \text{Btu/lbm} = \mathbf{446.3\ Btu}$$

(*b*) The exergy destruction (or irreversibility) can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation is determined from an entropy balance on an *extended system* that includes the tank and the region in its immediate surroundings so that the boundary temperature of the extended system where heat transfer occurs is the source temperature,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} + S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1),$$

$$S_{\text{gen}} = m(s_2 - s_1) - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (535\ \text{R})\left[(18.40\ \text{lbm})(0.1922 - 0.1436)\text{Btu/lbm} \cdot \text{R} - \frac{446.3\ \text{Btu}}{580\ \text{R}}\right] = \mathbf{66.5\ Btu}$$

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8-48 Chickens are to be cooled by chilled water in an immersion chiller that is also gaining heat from the surroundings. The rate of heat removal from the chicken and the rate of exergy destruction during this process are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Thermal properties of chickens and water are constant. **3** The temperature of the surrounding medium is 25°C.

Properties The specific heat of chicken is given to be 3.54 kJ/kg.°C. The specific heat of water at room temperature is 4.18 kJ/kg.°C (Table A-3).

Analysis (a) Chickens are dropped into the chiller at a rate of 700 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (700\ \text{chicken/h})(1.6\ \text{kg/chicken}) = 1120\ \text{kg/h} = 0.3111\ \text{kg/s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m} c_p \Delta T)_{\text{chicken}} = (0.3111\ \text{kg/s})(3.54\ \text{kJ/kg.}^{\text{o}}\text{C})(15 - 3)^{\text{o}}\text{C} = \mathbf{13.22\ kW}$$

The chiller gains heat from the surroundings as a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.22\ \text{kW} + (400/3600)\ \text{kW} = 13.33\ \text{kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p \Delta T)_{\text{water}}} = \frac{13.33\ \text{kW}}{(4.18\ \text{kJ/kg.}^{\text{o}}\text{C})(2^{\text{o}}\text{C})} = 1.594\ \text{kg/s}$$

(*b*) The exergy destruction can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The rate of entropy generation during this chilling process is determined by applying the rate form of the entropy balance on an *extended system* that includes the chiller and the immediate surroundings so that the boundary temperature is the surroundings temperature:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \frac{Q_{\text{in}}}{T_{\text{surr}}} + \dot{S}_{\text{gen}} = 0$$

$$\dot{m}_{\text{chicken}} s_1 + \dot{m}_{\text{water}} s_3 - \dot{m}_{\text{chicken}} s_2 - \dot{m}_{\text{water}} s_4 + \frac{Q_{\text{in}}}{T_{\text{surr}}} + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{chicken}} (s_2 - s_1) + \dot{m}_{\text{water}} (s_4 - s_3) - \frac{Q_{\text{in}}}{T_{\text{surr}}}$$

Noting that both streams are incompressible substances, the rate of entropy generation is determined to be

$$\begin{aligned}\dot{S}_{\text{gen}} &= \dot{m}_{\text{chicken}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{water}} c_p \ln\frac{T_4}{T_3} - \frac{\dot{Q}_{in}}{T_{\text{surr}}} \\ &= (0.3111\ \text{kg/s})(3.54\ \text{kJ/kg.K}) \ln\frac{276}{288} + (1.594\ \text{kg/s})(4.18\ \text{kJ/kg.K}) \ln\frac{275.5}{273.5} - \frac{(400/3600)\ \text{kW}}{298\ \text{K}} \\ &= 0.001306\ \text{kW/K}\end{aligned}$$

Finally,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298\ \text{K})(0.001306\ \text{kW/K}) = \mathbf{0.389\ kW}$$

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8-49 Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air and the rate of exergy destruction due to this heat transfer are to be determined.

Assumptions 1 The thermal properties of the balls are constant. 2 There are no changes in kinetic and potential energies. 3 The balls are at a uniform temperature at the end of the process.

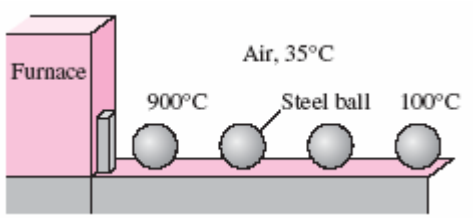
Properties The density and specific heat of the balls are given to be $\rho = 7833$ kg/m^3 and $c_p = 0.465$ kJ/kg.°C.

Analysis (*a*) We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc_p(T_1 - T_2)$$



The amount of heat transfer from a single ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (7833\ \text{kg/m}^3)\frac{\pi(0.008\ \text{m})^3}{6} = 0.00210\ \text{kg}$$

$$Q_{\text{out}} = mc_p(T_1 - T_2) = (0.0021\ \text{kg})(0.465\ \text{kJ/kg.°C})(900 - 100)\text{°C} = 781\ \text{J} = 0.781\ \text{kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

$$\dot{Q}_{\text{out}} = \dot{n}_{\text{ball}} Q_{\text{out}} = (1200\ \text{balls/h}) \times (0.781\ \text{kJ/ball}) = 936\ \text{kJ/h} = \mathbf{260\ W}$$

(*b*) The exergy destruction (or irreversibility) can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the ball and its immediate surroundings so that the boundary temperature of the extended system is at 35°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_p \ln\frac{T_2}{T_1} = (0.00210\ \text{kg})(0.465\ \text{kJ/kg.K})\ln\frac{100 + 273}{900 + 273} = -0.00112\ \text{kJ/K}$$

Substituting,

$$S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}} = \frac{0.781\ \text{kJ}}{308\ \text{K}} - 0.00112\ \text{kJ/K} = 0.00142\ \text{kJ/K} \quad \text{(per ball)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = S_{\text{gen}} \dot{n}_{\text{ball}} = (0.00142\ \text{kJ/K} \cdot \text{ball})(1200\ \text{balls/h}) = 1.704\ \text{kJ/h.K} = 0.000473\ \text{kW/K}$$

Finally,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (308\ \text{K})(0.000473\ \text{kW/K}) = 0.146\ \text{kW} = \mathbf{146\ W}$$

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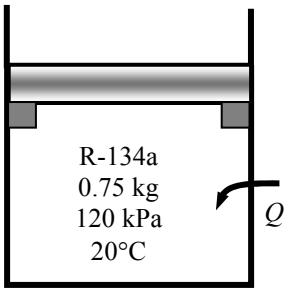
8-50 Heat is transferred to a piston-cylinder device with a set of stops. The work done, the heat transfer, the exergy destroyed, and the second-law efficiency are to be determined.

Assumptions **1** The device is stationary and kinetic and potential energy changes are zero. **2** There is no friction between the piston and the cylinder. 3 Heat is transferred to the refrigerant from a source at 150˚C.

Analysis (*a*) The properties of the refrigerant at the initial and final states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 120\ \text{kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.19390\ \text{m}^3/\text{kg} \\ u_1 = 248.51\ \text{kJ/kg} \\ s_1 = 1.0760\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 140\ \text{kPa} \\ T_2 = 90°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.20847\ \text{m}^3/\text{kg} \\ u_2 = 305.38\ \text{kJ/kg} \\ s_2 = 1.2553\ \text{kJ/kg.K} \end{array}$$



Noting that pressure remains constant at 140 kPa as the piston moves, the boundary work is determined to be

$$W_{\text{b,out}} = mP_2(v_2 - v_1) = (0.75\ \text{kg})(140\ \text{kPa})(0.20847 - 0.19390)\text{m}^3/\text{kg} = \mathbf{1.53\ kJ}$$

(*b*) The heat transfer can be determined from an energy balance on the system

$$Q_{\text{in}} = m(u_2 - u_1) + W_{\text{b,out}} = (0.75\ \text{kg})(305.38 - 248.51)\text{kJ/kg} + 1.53\ \text{kJ} = \mathbf{44.2\ kJ}$$

(*c*) The exergy destruction associated with this process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation is determined from an entropy balance on an *extended system* that includes the piston-cylinder device and the region in its immediate surroundings so that the boundary temperature of the extended system where heat transfer occurs is the source temperature,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} + S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1),$$

$$S_{\text{gen}} = m(s_2 - s_1) - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\ \text{K})\left[(0.75\ \text{kg})(1.2553 - 1.0760)\text{kJ/kg}\cdot\text{K} - \frac{44.2\ \text{kJ}}{150 + 273\ \text{K}}\right] = \mathbf{8.935\ kJ}$$

(*d*) Exergy expended is the work potential of the heat extracted from the source at 150˚C,

$$X_{\text{expended}} = X_Q = \eta_{\text{th,rev}} Q = \left(1 - \frac{T_L}{T_H}\right)Q = \left(1 - \frac{25 + 273\ \text{K}}{150 + 273\ \text{K}}\right)(44.2\ \text{kJ}) = 13.06\ \text{kJ}$$

Then the 2nd law efficiency becomes

$$\eta_{\text{II}} = \frac{X_{\text{recovered}}}{X_{\text{expended}}} = 1 - \frac{X_{\text{destroyed}}}{X_{\text{expended}}} = 1 - \frac{8.935\ \text{kJ}}{13.06\ \text{kJ}} = 0.316 \text{ or } \mathbf{31.6\%}$$

Discussion The second-law efficiency can also be determine as follows:

The exergy increase of the refrigerant is the exergy difference between the initial and final states,

$$\begin{aligned} \Delta X &= m[u_2 - u_1 - T_0(s_2 - s_1) + P_0(v_2 - v_1)] \\ &= (0.75\ \text{kg})[(305.38 - 248.51)\text{kJ/kg} - (298\ \text{K})(1.2553 - 1.0760)\text{kg.K} + (100\ \text{kPa})(0.20847 - 0.19390)\text{m}^3/\text{kg}] \\ &= 3.666\ \text{kJ} \end{aligned}$$

The useful work output for the process is

$$W_{\text{u,out}} = W_{\text{b,out}} - mP_0(v_2 - v_1) = 1.53\ \text{kJ} - (0.75\ \text{kg})(100\ \text{kPa})(0.20847 - 0.19390)\text{m}^3/\text{kg} = 0.437\ \text{kJ}$$

The exergy recovered is the sum of the exergy increase of the refrigerant and the useful work output,

$$X_{\text{recovered}} = \Delta X + W_{\text{u,out}} = 3.666 + 0.437 = 4.103\ \text{kJ}$$

Then the second-law efficiency becomes

$$\eta_{\text{II}} = \frac{X_{\text{recovered}}}{X_{\text{expended}}} = \frac{4.103\ \text{kJ}}{13.06\ \text{kJ}} = 0.314 \text{ or } \mathbf{31.4\%}$$

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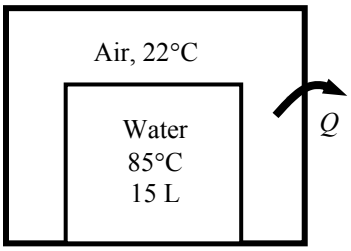
8-51 A tank containing hot water is placed in a larger tank. The amount of heat lost to the surroundings and the exergy destruction during the process are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats. **3** The larger tank is well-sealed.

Properties The properties of air at room temperature are $R = 0.287$ kPa.m^3/kg.K, $c_p = 1.005$ kJ/kg.K, $c_v = 0.718$ kJ/kg.K (Table A-2). The properties of water at room temperature are $\rho = 997$ kg/m^3, $c_w = 4.18$ kJ/kg.K (Table A-3).

Analysis (*a*) The final volume of the air in the tank is

$$V_{a2} = V_{a1} - V_w = 0.04 - 0.015 = 0.025\ \text{m}^3$$



The mass of the air in the large tank is

$$m_a = \frac{P_1 V_{a1}}{RT_{a1}} = \frac{(100\ \text{kPa})(0.04\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(22+273\ \text{K})} = 0.04724\ \text{kg}$$

The pressure of air at the final state is

$$P_{a2} = \frac{m_a RT_{a2}}{V_{a2}} = \frac{(0.04724\ \text{kg})(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(44+273\ \text{K})}{0.025\ \text{m}^3} = 171.9\ \text{kPa}$$

The mass of water is

$$m_w = \rho_w V_w = (997\ \text{kg/m}^3)(0.015\ \text{m}^3) = 14.96\ \text{kg}$$

An energy balance on the system consisting of water and air is used to determine heat lost to the surroundings

$$\begin{aligned} Q_{\text{out}} &= -\left[m_w c_w (T_2 - T_{w1}) + m_a c_v (T_2 - T_{a1})\right] \\ &= -(14.96\ \text{kg})(4.18\ \text{kJ/kg.K})(44-85) - (0.04724\ \text{kg})(0.718\ \text{kJ/kg.K})(44-22) \\ &= \mathbf{2563\ kJ} \end{aligned}$$

(*b*) An exergy balance written on the (system + immediate surroundings) can be used to determine exergy destruction. But we first determine entropy and internal energy changes

$$\Delta S_w = m_w c_w \ln\frac{T_{w1}}{T_2} = (14.96\ \text{kg})(4.18\ \text{kJ/kg.K})\ln\frac{(85+273)\ \text{K}}{(44+273)\ \text{K}} = 7.6059\ \text{kJ/K}$$

$$\begin{aligned} \Delta S_a &= m_a\left[c_p \ln\frac{T_{a1}}{T_2} - R\ln\frac{P_{a1}}{P_2}\right] \\ &= (0.04724\ \text{kg})\left[(1.005\ \text{kJ/kg.K})\ln\frac{(22+273)\ \text{K}}{(44+273)\ \text{K}} - (0.287\ \text{kJ/kg.K})\ln\frac{100\ \text{kPa}}{171.9\ \text{kPa}}\right] \\ &= 0.003931\ \text{kJ/K} \end{aligned}$$

$$\Delta U_w = m_w c_w (T_{1w} - T_2) = (14.96\ \text{kg})(4.18\ \text{kJ/kg.K})(85 - 44)\text{K} = 2564\ \text{kJ}$$

$$\Delta U_a = m_a c_v (T_{1a} - T_2) = (0.04724\ \text{kg})(0.718\ \text{kJ/kg.K})(22 - 44)\text{K} = -0.7462\ \text{kJ}$$

$$\begin{aligned} X_{\text{dest}} &= \Delta X_w + \Delta X_a \\ &= \Delta U_w - T_0 \Delta S_w + \Delta U_a - T_0 \Delta S_a \\ &= 2564\ \text{kJ} - (295\ \text{K})(7.6059\ \text{kJ/K}) + (-0.7462\ \text{kJ}) - (295\ \text{K})(0.003931\ \text{kJ/K}) \\ &= \mathbf{318.4\ kJ} \end{aligned}$$

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Exergy Analysis of Control Volumes

8-52 R-134a is is throttled from a specified state to a specified pressure. The temperature of R-134a at the outlet of the expansion valve, the entropy generation, and the exergy destruction are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer is negligible.

(a) The properties of refrigerant at the inlet and exit states of the throttling valve are (from R134a tables)

$$\left.\begin{array}{l} P_1 = 1200\ \text{kPa} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 117.77\ \text{kJ/kg} \\ s_1 = 0.4244\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200\ \text{kPa} \\ h_2 = h_1 = 117.77\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{-10.1^\circ C} \\ s_2 = 0.4562\ \text{kJ/kg}\cdot\text{K} \end{array}$$

(b) Noting that the throttling valve is adiabatic, the entropy generation is determined from

$$s_{\text{gen}} = s_2 - s_1 = (0.4562 - 0.4244)\text{kJ/kg}\cdot\text{K} = \mathbf{0.03176\ kJ/kg\cdot K}$$

Then the irreversibility (i.e., exergy destruction) of the process becomes

$$ex_{\text{dest}} = T_0 s_{\text{gen}} = (298\ \text{K})(0.03176\ \text{kJ/kg}\cdot\text{K}) = \mathbf{9.464\ kJ/kg}$$

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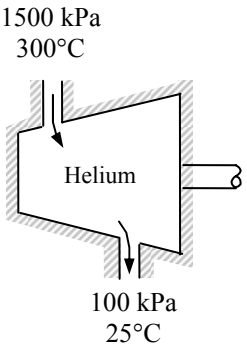
8-53 Heium expands in an adiabatic turbine from a specified inlet state to a specified exit state. The maximum work output is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The device is adiabatic and thus heat transfer is negligible. **3** Helium is an ideal gas. **4** Kinetic and potential energy changes are negligible.

Properties The properties of helium are c_p = 5.1926 kJ/kg.K and R = 2.0769 kJ/kg.K (Table A-1).

Analysis The entropy change of helium is

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (5.1926\,\text{kJ/kg}\cdot\text{K})\ln\frac{298\,\text{K}}{573\,\text{K}} - (2.0769\,\text{kJ/kg}\cdot\text{K})\ln\frac{100\,\text{kPa}}{1500\,\text{kPa}} \\ &= 2.2295\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$



The maximum (reversible) work is the exergy difference between the inlet and exit states

$$\begin{aligned} w_{\text{rev,out}} &= h_1 - h_2 - T_0(s_1 - s_2) \\ &= c_p(T_1 - T_2) - T_0(s_1 - s_2) \\ &= (5.1926\,\text{kJ/kg}\cdot\text{K})(300 - 25)\text{K} - (298\,\text{K})(-2.2295\,\text{kJ/kg}\cdot\text{K}) \\ &= \mathbf{2092\ kJ/kg} \end{aligned}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\begin{aligned} \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{m}h_1 &= \dot{W}_{\text{out}} + \dot{Q}_{\text{out}} + \dot{m}h_2 \\ \dot{W}_{\text{out}} &= \dot{m}(h_1 - h_2) - \dot{Q}_{\text{out}} \\ w_{\text{out}} &= (h_1 - h_2) - q_{\text{out}} \end{aligned}$$

Inspection of this result reveals that any rejection of heat will decrease the work that will be produced by the turbine since inlet and exit states (i.e., enthalpies) are fixed.

If there is heat loss from the turbine, the maximum work output is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{-\dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\begin{aligned} \dot{X}_{\text{in}} &= \dot{X}_{\text{out}} \\ \dot{m}\psi_1 &= \dot{W}_{\text{rev,out}} + \dot{Q}_{\text{out}}\left(1 - \frac{T_0}{T}\right) + \dot{m}\psi_2 \\ w_{\text{rev,out}} &= (\psi_1 - \psi_2) - q_{\text{out}}\left(1 - \frac{T_0}{T}\right) \\ &= (h_1 - h_2) - T_0(s_1 - s_2) - q_{\text{out}}\left(1 - \frac{T_0}{T}\right) \end{aligned}$$

Inspection of this result reveals that any rejection of heat will decrease the maximum work that could be produced by the turbine. Therefore, for the maximum work, the turbine must be adiabatic.

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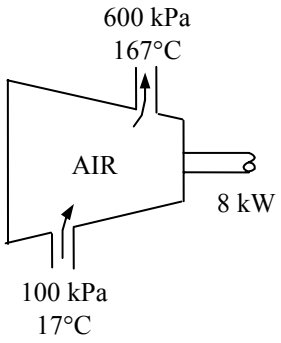
8-54 Air is compressed steadily by an 8-kW compressor from a specified state to another specified state. The increase in the exergy of air and the rate of exergy destruction are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1). From the air table (Table A-17)

$$T_1 = 290 \text{ K} \longrightarrow h_1 = 290.16 \text{ kJ/kg}, \quad s_1^{\circ} = 1.66802 \text{ kJ/kg} \cdot \text{K}$$

$$T_2 = 440 \text{ K} \longrightarrow h_2 = 441.61 \text{ kJ/kg}, \quad s_2^{\circ} = 2.0887 \text{ kJ/kg} \cdot \text{K}$$



Analysis The increase in exergy is the difference between the exit and inlet flow exergies,

$$\begin{aligned}\text{Increase in exergy} &= \psi_2 - \psi_1 \\ &= [(h_2 - h_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} - T_0(s_2 - s_1)] \\ &= (h_2 - h_1) - T_0(s_2 - s_1)\end{aligned}$$

where

$$\begin{aligned}s_2 - s_1 &= (s_2^{\circ} - s_1^{\circ}) - R\ln\frac{P_2}{P_1} \\ &= (2.0887 - 1.66802)\text{kJ/kg} \cdot \text{K} - (0.287 \text{ kJ/kg} \cdot \text{K})\ln\frac{600 \text{ kPa}}{100 \text{ kPa}} \\ &= -0.09356 \text{ kJ/kg} \cdot \text{K}\end{aligned}$$

Substituting,

$$\begin{aligned}\text{Increase in exergy} &= \psi_2 - \psi_1 \\ &= [(441.61 - 290.16)\text{kJ/kg} - (290 \text{ K})(-0.09356 \text{ kJ/kg} \cdot \text{K})] \\ &= \mathbf{178.6 \text{ kJ/kg}}\end{aligned}$$

Then the reversible power input is

$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = (2.1/60 \text{ kg/s})(178.6 \text{ kJ/kg}) = 6.25 \text{ kW}$$

(*b*) The rate of exergy destruction (or irreversibility) is determined from its definition,

$$\dot{X}_{\text{destroyed}} = \dot{W}_{\text{in}} - \dot{W}_{\text{rev,in}} = 8 - 6.25 = \mathbf{1.75 \text{ kW}}$$

Discussion Note that 1.75 kW of power input is wasted during this compression process.

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8-55 Problem 8-54 is reconsidered. The problem is to be solved and the actual heat transfer, its direction, the minimum power input, and the compressor second-law efficiency are to be determined.

Analysis The problem is solved using EES, and the solution is given below.

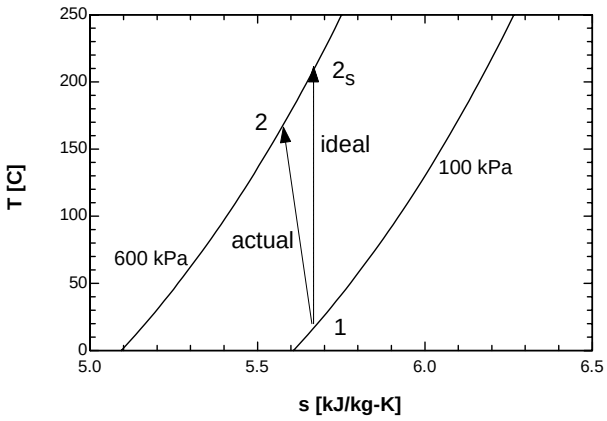
```
Function Direction$(Q)
If Q<0 then Direction$='out' else Direction$='in'
end
Function Violation$(eta)
If eta>1 then Violation$='You have violated the 2nd Law!!!!!' else Violation$=''
end

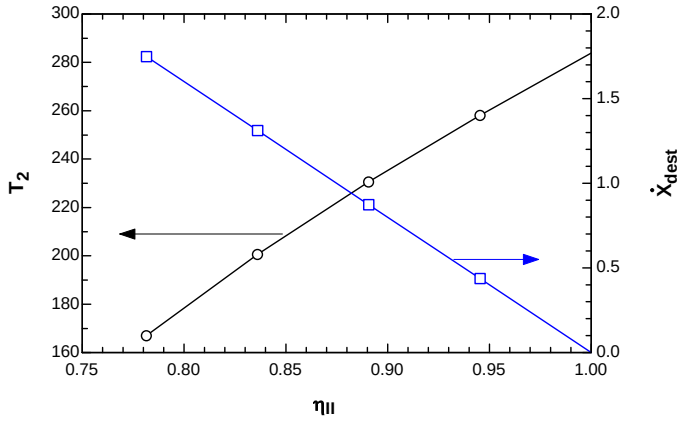
{"Input Data from the Diagram Window"
T_1=17 [C]
P_1=100 [kPa]
W_dot_c = 8 [kW]
P_2=600 [kPa]
S_dot_gen=0
Q_dot_net=0}
{"Special cases"
T_2=167 [C]
m_dot=2.1 [kg/min]}
T_o=T_1
P_o=P_1
m_dot_in=m_dot*Convert(kg/min, kg/s)
"Steady-flow conservation of mass"
m_dot_in = m_dot_out
"Conservation of energy for steady-flow is:"
E_dot_in - E_dot_out = DELTAE_dot
DELTAE_dot = 0
E_dot_in=Q_dot_net + m_dot_in*h_1 +W_dot_c
"If Q_dot_net < 0, heat is transferred from the compressor"
E_dot_out= m_dot_out*h_2
h_1 =enthalpy(air,T=T_1)
h_2 = enthalpy(air, T=T_2)
W_dot_net=-W_dot_c
W_dot_rev=-m_dot_in*(h_2 - h_1 -(T_1+273.15)*(s_2-s_1))
"Irreversibility, entropy generated, second law efficiency, and exergy destroyed:"
s_1=entropy(air, T=T_1,P=P_1)
s_2=entropy(air,T=T_2,P=P_2)
s_2s=entropy(air,T=T_2s,P=P_2)
s_2s=s_1"This yields the isentropic T_2s for an isentropic process bewteen T_1, P_1 and
P_2"I_dot=(T_o+273.15)*S_dot_gen"Irreversiblility for the Process, KW"
S_dot_gen=(-Q_dot_net/(T_o+273.15) +m_dot_in*(s_2-s_1)) "Entropy generated, kW"
Eta_II=W_dot_rev/W_dot_net"Definition of compressor second law efficiency, Eq. 7_6"
h_o=enthalpy(air,T=T_o)
s_o=entropy(air,T=T_o,P=P_o)
Psi_in=h_1-h_o-(T_o+273.15)*(s_1-s_o)  "availability function at state 1"
Psi_out=h_2-h_o-(T_o+273.15)*(s_2-s_o) "availability function at state 2"
X_dot_in=Psi_in*m_dot_in
X_dot_out=Psi_out*m_dot_out
DELTAX_dot=X_dot_in-X_dot_out
"General Exergy balance for a steady-flow system, Eq. 7-47"
(1-(T_o+273.15)/(T_o+273.15))*Q_dot_net-W_dot_net+m_dot_in*Psi_in - m_dot_out*Psi_out =X_dot_dest
"For the Diagram Window"
Text$=Direction$(Q_dot_net)
Text2$=Violation$(Eta_II)
```

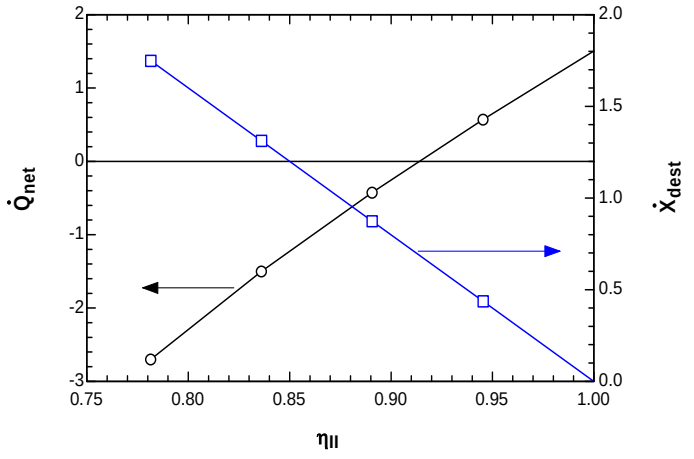
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η_{II}	I [kW]	X_{dest} [kW]	T_{2s} [C]	T_2 [C]	Q_{net} [kW]
0.7815	1.748	1.748	209.308	167	-2.7
0.8361	1.311	1.311	209.308	200.6	-1.501
0.8908	0.874	0.874	209.308	230.5	-0.4252
0.9454	0.437	0.437	209.308	258.1	0.5698
1	1.425E-13	5.407E-15	209.308	283.9	1.506

How can entropy decrease?







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8-56 Steam is decelerated in a diffuser. The second law efficiency of the diffuser is to be determined.

Assumptions **1** The diffuser operates steadily. **2** The changes in potential energies are negligible.

Properties The properties of steam at the inlet and the exit of the diffuser are (Tables A-4 through A-6)

$$\left.\begin{aligned} P_1 &= 500\ \text{kPa} \\ T_1 &= 200°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 2855.8\ \text{kJ/kg} \\ s_1 &= 7.0610\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 200\ \text{kPa} \\ x_2 &= 1\ \text{(sat. vapor)} \end{aligned}\right\} \begin{aligned} h_2 &= 2706.3\ \text{kJ/kg} \\ s_2 &= 7.1270\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

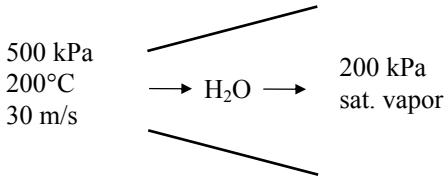
Analysis We take the diffuser to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$\frac{V_2^2 - V_1^2}{2} = h_1 - h_2 = \Delta \text{ke}_{\text{actual}}$$



Substituting,

$$\Delta \text{ke}_{\text{actual}} = h_1 - h_2 = 2855.8 - 2706.3 = 149.5\ \text{kJ/kg}$$

An exergy balance on the diffuser gives

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 = \dot{m}\psi_2$$

$$h_1 - h_0 + \frac{V_1^2}{2} - T_0(s_1 - s_0) = h_2 - h_0 + \frac{V_2^2}{2} - T_0(s_2 - s_0)$$

$$\frac{V_2^2 - V_1^2}{2} = h_1 - h_2 - T_0(s_1 - s_2)$$

$$\Delta \text{ke}_{\text{rev}} = h_1 - h_2 - T_0(s_1 - s_2)$$

Substituting,

$$\begin{aligned} \Delta \text{ke}_{\text{rev}} &= h_1 - h_2 - T_0(s_1 - s_2) \\ &= (2855.8 - 2706.3)\text{kJ/kg} - (298\ \text{K})(7.0610 - 7.1270)\ \text{kJ/kg}\cdot\text{K} \\ &= 169.2\ \text{kJ/kg} \end{aligned}$$

The second law efficiency is then

$$\eta_{\text{II}} = \frac{\Delta \text{ke}_{\text{actual}}}{\Delta \text{ke}_{\text{rev}}} = \frac{149.5\ \text{kJ/kg}}{169.2\ \text{kJ/kg}} = \mathbf{0.884}$$

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8-57 Air is accelerated in a nozzle while losing some heat to the surroundings. The exit temperature of air and the exergy destroyed during the process are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The nozzle operates steadily.

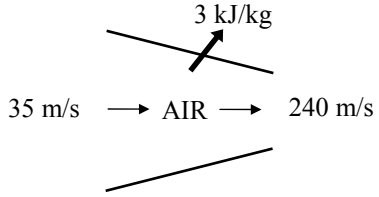
Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1). The properties of air at the nozzle inlet are (Table A-17)

$$T_1 = 338\ \text{K} \longrightarrow h_1 = 338.40\ \text{kJ/kg}$$
$$s_1^{\text{o}} = 1.8219\ \text{kJ/kg}\cdot\text{K}$$

Analysis (a) We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\nearrow 0\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + \text{V}_2^2/2) + \dot{Q}_{\text{out}}$$



or

$$0 = q_{\text{out}} + h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Therefore,

$$h_2 = h_1 - q_{\text{out}} - \frac{V_2^2 - V_1^2}{2} = 338.40 - 3 - \frac{(240\ \text{m/s})^2 - (35\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 307.21\ \text{kJ/kg}$$

At this h_2 value we read, from Table A-17, $T_2 = 307.0$ K **= 34.0°C** and $s_2^{\text{o}} = 1.7251\ \text{kJ/kg}\cdot\text{K}$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation S_{gen} is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is T_{surr} at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\not\nearrow 0}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$ms_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}}$$

where

$$\Delta s_{\text{air}} = s_2^{\text{o}} - s_1^{\text{o}} - R\ln\frac{P_2}{P_1} = (1.7251 - 1.8219)\text{kJ/kg}\cdot\text{K} - (0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{95\ \text{kPa}}{200\ \text{kPa}} = 0.1169\ \text{kJ/kg}\cdot\text{K}$$

Substituting, the entropy generation and exergy destruction per unit mass of air are determined to be

$$x_{\text{destroyed}} = T_0 s_{gen} = T_{surr} s_{gen}$$
$$= T_0\left(s_2 - s_1 + \frac{q_{surr}}{T_{surr}}\right) = (290\ \text{K})\left(0.1169\ \text{kJ/kg}\cdot\text{K} + \frac{3\ \text{kJ/kg}}{290\ \text{K}}\right) = \mathbf{36.9\ kJ/kg}$$

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Alternative solution The exergy destroyed during a process can be determined from an exergy balance applied on the *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is environment temperature T_0 (or T_{surr}) at all times. Noting that exergy transfer with heat is zero when the temperature at the point of transfer is the environment temperature, the exergy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0 \rightarrow \dot{X}_{\text{destroyed}} = \dot{X}_{in} - \dot{X}_{out} = \dot{m}\psi_1 - \dot{m}\psi_2 = \dot{m}(\psi_1 - \psi_2)$$

$$= \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta ke - \Delta pe^{\nearrow 0}] = \dot{m}[T_0(s_2 - s_1) - (h_2 - h_1 + \Delta ke)]$$

$$= \dot{m}[T_0(s_2 - s_1) + q_{\text{out}}] \quad \text{since, from energy balance, } -q_{\text{out}} = h_2 - h_1 + \Delta ke$$

$$= T_0\left(\dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0}\right) = T_0\dot{S}_{\text{gen}}$$

Therefore, the two approaches for the determination of exergy destruction are identical.

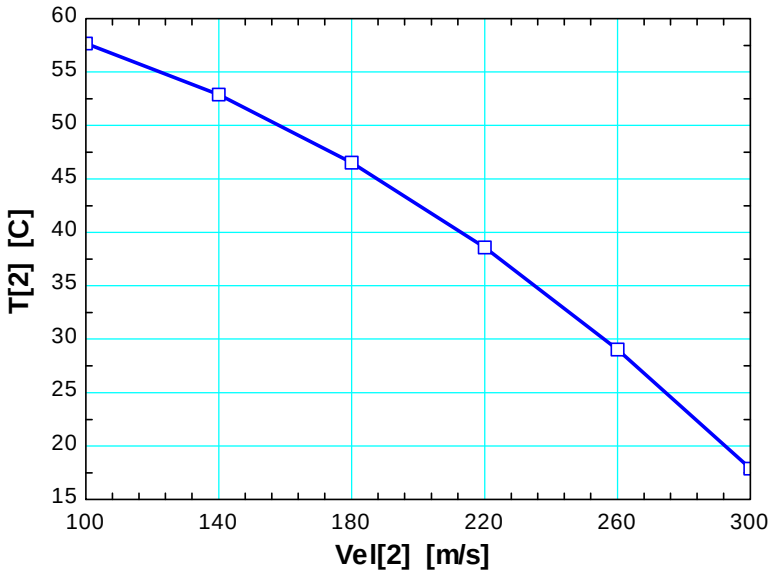
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8-58 Problem 8-57 is reconsidered. The effect of varying the nozzle exit velocity on the exit temperature and exergy destroyed is to be investigated.

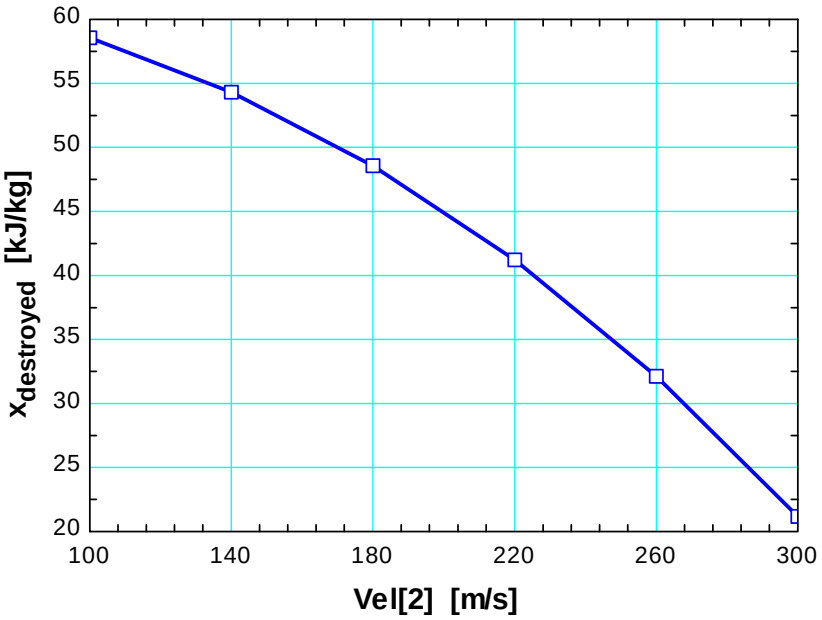
Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns:"
WorkFluid$ = 'Air'
P[1] = 200  [kPa]
T[1] =65 [C]
P[2] = 95  [kPa]
Vel[1] = 35 [m/s]
{Vel[2] = 240 [m/s]}
T_o = 17 [C]
T_surr = T_o
q_loss = 3 [kJ/kg]
"Conservation of Energy - SSSF energy balance for nozzle -- neglecting the change in potential energy:"
h[1]=enthalpy(WorkFluid$,T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
ke[1] = Vel[1]^2/2
ke[2]=Vel[2]^2/2
h[1]+ke[1]*convert(m^2/s^2,kJ/kg) = h[2] + ke[2]*convert(m^2/s^2,kJ/kg)+q_loss
T[2]=temperature(WorkFluid$,h=h[2])
s[2]=entropy(WorkFluid$,P=P[2],h=h[2])
"The entropy generated is detemined from the entropy balance:"
s[1] - s[2] - q_loss/(T_surr+273) + s_gen = 0
x_destroyed = (T_o+273)*s_gen
```

Vel_2 [m/s]	T_2 [C]	$x_{destroyed}$ [kJ/kg]
100	57.66	58.56
140	52.89	54.32
180	46.53	48.56
220	38.58	41.2
260	29.02	32.12
300	17.87	21.16



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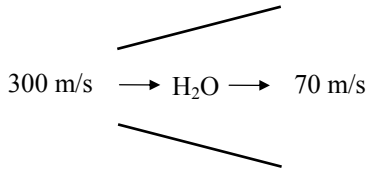
8-59 Steam is decelerated in a diffuser. The mass flow rate of steam and the wasted work potential during the process are to be determined.

Assumptions **1** The diffuser operates steadily. **2** The changes in potential energies are negligible.

Properties The properties of steam at the inlet and the exit of the diffuser are (Tables A-4 through A-6)

$$\left.\begin{matrix} P_1 = 10\,\text{kPa} \\ T_1 = 50°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 2592.0\,\text{kJ/kg} \\ s_1 = 8.1741\,\text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} T_2 = 50°\text{C} \\ \text{sat.vapor} \end{matrix}\right\} \begin{matrix} h_2 = 2591.3\,\text{kJ/kg} \\ s_2 = 8.0748\,\text{kJ/kg}\cdot\text{K} \\ \upsilon_2 = 12.026\,\text{m}^3/\text{kg} \end{matrix}$$



Analysis (a) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\upsilon_2} A_2 V_2 = \frac{1}{12.026\,\text{m}^3/\text{kg}} (3\,\text{m}^2)(70\,\text{m/s}) = \mathbf{17.46\,kg/s}$$

(*b*) We take the diffuser to be the system, which is a control volume. Assuming the direction of heat transfer to be from the stem, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\not\to 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + \text{V}_2^2/2) + \dot{Q}_{\text{out}}$$

$$\dot{Q}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting,

$$\dot{Q}_{\text{out}} = -(17.46\,\text{kg/s})\left[2591.3 - 2592.0 + \frac{(70\,\text{m/s})^2 - (300\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right)\right] = 754.8\,\text{kJ/s}$$

The wasted work potential is equivalent to exergy destruction. The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation S_{gen} is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is T_{surr} at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\to 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0\left(\dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0}\right)$$

$$= (298\,\text{K})\left((17.46\,\text{kg/s})(8.0748 - 8.1741)\text{kJ/kg}\cdot\text{K} + \frac{754.8\,\text{kW}}{298\,\text{K}}\right)$$

$$= \mathbf{238.3\,kW}$$

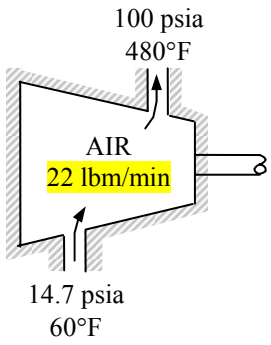
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8-60E Air is compressed steadily by a compressor from a specified state to another specified state. The minimum power input required for the compressor is to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is $R = 0.06855$ Btu/lbm.R (Table A-1E). From the air table (Table A-17E)

$$T_1 = 520\ \text{R} \longrightarrow h_1 = 124.27\ \text{Btu/lbm}$$
$$s_1^{\text{o}} = 0.59173\ \text{Btu/lbm}\cdot\text{R}$$
$$T_2 = 940\ \text{R} \longrightarrow h_2 = 226.11\ \text{Btu/lbm}$$
$$s_2^{\text{o}} = 0.73509\ \text{Btu/lbm}\cdot\text{R}$$



Analysis The reversible (or minimum) power input is determined from the rate form of the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} \quad \underbrace{-\dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$
$$\dot{m}\psi_1 + \dot{W}_{\text{rev,in}} = \dot{m}\psi_2$$
$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}]$$

where

$$\Delta s_{\text{air}} = s_2^{\text{o}} - s_1^{\text{o}} - R\ln\frac{P_2}{P_1}$$
$$= (0.73509 - 0.59173)\text{Btu/lbm}\cdot\text{R} - (0.06855\ \text{Btu/lbm}\cdot\text{R})\ln\frac{100\ \text{psia}}{14.7\ \text{psia}}$$
$$= 0.01193\ \text{Btu/lbm}\cdot\text{R}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (22/60\ \text{lbm/s})\left[(226.11 - 124.27)\text{Btu/lbm} - (520\ \text{R})(0.01193\ \text{Btu/lbm}\cdot\text{R})\right]$$
$$= 35.1\ \text{Btu/s} = \mathbf{49.6\ hp}$$

Discussion Note that this is the minimum power input needed for this compressor.

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8-61 Steam expands in a turbine from a specified state to another specified state. The actual power output of the turbine is given. The reversible power output and the second-law efficiency are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** The temperature of the surroundings is given to be 25°C.

Properties From the steam tables (Tables A-4 through A-6)

$$\left.\begin{matrix} P_1 = 6\ \text{MPa} \\ T_1 = 600°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3658.8\ \text{kJ/kg} \\ s_1 = 7.1693\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 50\ \text{kPa} \\ T_2 = 100°\text{C} \end{matrix}\right\} \begin{matrix} h_2 = 2682.4\ \text{kJ/kg} \\ s_2 = 7.6953\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

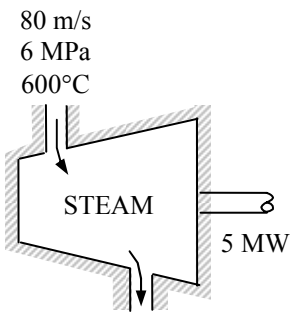
Analysis (*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2)$$

$$\dot{W}_{\text{out}} = \dot{m}\left[h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right]$$



Substituting,

$$5000\ \text{kJ/s} = \dot{m}\left(3658.8 - 2682.4 + \frac{(80\ \text{m/s})^2 - (140\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = 5.156\ \text{kg/s}$$

The reversible (or maximum) power output is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 = \dot{W}_{\text{rev,out}} + \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,out}} = \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta\text{ke} - \Delta\text{pe}^{\nearrow 0}]$$

Substituting,

$$\dot{W}_{\text{rev,out}} = \dot{W}_{\text{out}} - \dot{m}T_0(s_1 - s_2)$$
$$= 5000\ \text{kW} - (5.156\ \text{kg/s})(298\ \text{K})(7.1693 - 7.6953)\ \text{kJ/kg}\cdot\text{K} = \mathbf{5808\ kW}$$

(*b*) The second-law efficiency of a turbine is the ratio of the actual work output to the reversible work,

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{out}}}{\dot{W}_{\text{rev,out}}} = \frac{5\ \text{MW}}{5.808\ \text{MW}} = \mathbf{86.1\%}$$

Discussion Note that 13.9% percent of the work potential of the steam is wasted as it flows through the turbine during this process.

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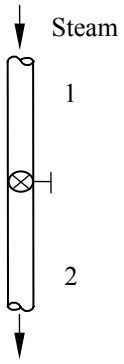
8-62 Steam is throttled from a specified state to a specified pressure. The decrease in the exergy of the steam during this throttling process is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C. **4** Heat transfer is negligible.

Properties The properties of steam before and after throttling are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 6\ \text{MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3178.3\ \text{kJ/kg} \\ s_1 = 6.5432\ \text{kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2\ \text{MPa} \\ h_2 = h_1 \end{array}\right\} s_2 = 7.0225\ \text{kJ/kg} \cdot \text{K}$$



Analysis The decrease in exergy is of the steam is the difference between the inlet and exit flow exergies,

$$\begin{aligned} \text{Decrease in exergy} &= \psi_1 - \psi_2 = -[\Delta h^{\nearrow^0} - \Delta ke^{\nearrow^0} - \Delta pe^{\nearrow^0} - T_0(s_1 - s_2)] = T_0(s_2 - s_1) \\ &= (298\ \text{K})(7.0225 - 6.5432)\text{kJ/kg} \cdot \text{K} \\ &= \mathbf{143\ kJ/kg} \end{aligned}$$

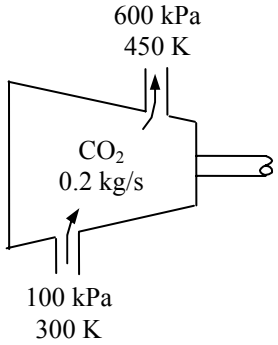
Discussion Note that 143 kJ/kg of work potential is wasted during this throttling process.

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8-63 CO_2 gas is compressed steadily by a compressor from a specified state to another specified state. The power input to the compressor if the process involved no irreversibilities is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** CO_2 is an ideal gas with constant specific heats.

Properties At the average temperature of (300 + 450)/2 = 375 K, the constant pressure specific heat and the specific heat ratio of CO_2 are $c_p = 0.917$ kJ/kg.K and $k = 1.261$ (Table A-2b). Also, $c_p = 0.1889$ kJ/kg.K (Table A-2a).



Analysis The reversible (or minimum) power input is determined from the exergy balance applied on the compressor, and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 + \dot{W}_{\text{rev,in}} = \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1)$$

$$= \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta\text{ke}^{\nearrow 0} + \Delta\text{pe}^{\nearrow 0}]$$

where

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (0.917\ \text{kJ/kg}\cdot\text{K})\ln\frac{450\ \text{K}}{300\ \text{K}} - (0.1889\ \text{kJ/kg}\cdot\text{K})\ln\frac{600\ \text{kPa}}{100\ \text{kPa}}$$

$$= 0.03335\ \text{kJ/kg}\cdot\text{K}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (0.2\ \text{kg/s})\big[(0.917\ \text{kJ/kg}\cdot\text{K})(450 - 300)\text{K} - (298\ \text{K})(0.03335\ \text{kJ/kg}\cdot\text{K})\big] = \mathbf{25.5\ kW}$$

Discussion Note that a minimum of 25.5 kW of power input is needed for this compressor.

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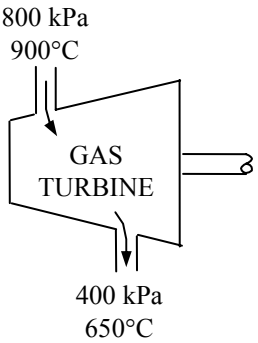
8-64 Combustion gases expand in a turbine from a specified state to another specified state. The exergy of the gases at the inlet and the reversible work output of the turbine are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C. **4** The combustion gases are ideal gases with constant specific heats.

Properties The constant pressure specific heat and the specific heat ratio are given to be c_p = 1.15 kJ/kg.K and k = 1.3. The gas constant R is determined from

$$R = c_p - c_v = c_p - c_p / k = c_p(1-1/k) = (1.15\,\text{kJ/kg}\cdot\text{K})(1-1/1.3) = 0.265\,\text{kJ/kg}\cdot\text{K}$$

Analysis (*a*) The exergy of the gases at the turbine inlet is simply the flow exergy,



$$\psi_1 = h_1 - h_0 - T_0(s_1 - s_0) + \frac{V_1^2}{2} + gz_1^{\nearrow 0}$$

where

$$\begin{aligned} s_1 - s_0 &= c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0} \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})\ln\frac{1173\,\text{K}}{298\,\text{K}} - (0.265\,\text{kJ/kg}\cdot\text{K})\ln\frac{800\,\text{kPa}}{100\,\text{kPa}} \\ &= 1.025\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

Thus,

$$\psi_1 = (1.15\,\text{kJ/kg.K})(900-25)°\text{C} - (298\,\text{K})(1.025\,\text{kJ/kg}\cdot\text{K}) + \frac{(100\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{705.8\,kJ/kg}$$

(*b*) The reversible (or maximum) work output is determined from an exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\begin{aligned} \dot{X}_{\text{in}} &= \dot{X}_{\text{out}} \\ \dot{m}\psi_1 &= \dot{W}_{\text{rev,out}} + \dot{m}\psi_2 \\ \dot{W}_{\text{rev,out}} &= \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta\text{ke} - \Delta\text{pe}^{\nearrow 0}] \end{aligned}$$

where

$$\Delta\text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(220\,\text{m/s})^2 - (100\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 19.2\,\text{kJ/kg}$$

and

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})\ln\frac{923\,\text{K}}{1173\,\text{K}} - (0.265\,\text{kJ/kg}\cdot\text{K})\ln\frac{400\,\text{kPa}}{800\,\text{kPa}} \\ &= -0.09196\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

Then the reversible work output on a unit mass basis becomes

$$\begin{aligned} w_{\text{rev,out}} &= h_1 - h_2 + T_0(s_2 - s_1) - \Delta\text{ke} = c_p(T_1 - T_2) + T_0(s_2 - s_1) - \Delta\text{ke} \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})(900-650)°\text{C} + (298\,\text{K})(-0.09196\,\text{kJ/kg}\cdot\text{K}) - 19.2\,\text{kJ/kg} \\ &= \mathbf{240.9\,kJ/kg} \end{aligned}$$

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8-65E Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The actual power input and the second-law efficiency to the compressor are to be determined.

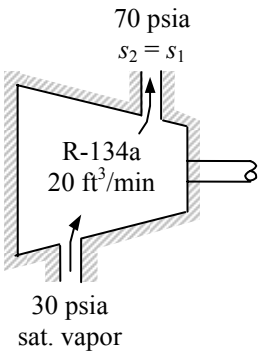
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the refrigerant tables (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 30 \text{ psia} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@30\text{ psia}} = 105.32 \text{ Btu/lbm} \\ s_1 = s_{g@30\text{ psia}} = 0.2238 \text{ Btu/lbm}\cdot\text{R} \\ \nu_1 = \nu_{g@30\text{ psia}} = 1.5492 \text{ ft}^3\text{/lbm} \end{array} \qquad \left.\begin{array}{l} P_2 = 70 \text{ psia} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 112.80 \text{ Btu/lbm}$$

Analysis From the isentropic efficiency relation,

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_c$$
$$= 105.32 + (112.80 - 105.32)/0.80$$
$$= 114.67 \text{ Btu/lbm}$$



Then,

$$\left.\begin{array}{l} P_2 = 70 \text{ psia} \\ h_{2a} = 114.67 \end{array}\right\} s_2 = 0.2274 \text{ Btu/lbm}$$

Also, $$\dot{m} = \frac{\dot{\nu}_1}{\nu_1} = \frac{20/60 \text{ ft}^3/\text{s}}{1.5492 \text{ ft}^3/\text{lbm}} = 0.2152 \text{ lbm/s}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{a,\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{a,\text{in}} = \dot{m}(h_2 - h_1)$$

Substituting, the actual power input to the compressor becomes

$$\dot{W}_{\text{a,in}} = (0.2152 \text{ lbm/s})(114.67 - 105.32) \text{ Btu/lbm}\left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}}\right) = \mathbf{2.85 \text{ hp}}$$

(*b*) The reversible (or minimum) power input is determined from the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0 \text{ (reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{W}_{\text{rev,in}} + \dot{m}\psi_1 = \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta \text{ke}^{\nearrow 0} + \Delta \text{pe}^{\nearrow 0}]$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (0.2152 \text{ lbm/s})\left[(114.67 - 105.32)\text{Btu/lbm} - (535 \text{ R})(0.2274 - 0.2238)\text{Btu/lbm}\cdot\text{R}\right]$$
$$= 1.606 \text{ Btu/s} = 2.27 \text{ hp} \qquad (\text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s})$$

Thus, $$\eta_{\text{II}} = \frac{\dot{W}_{\text{rev,in}}}{\dot{W}_{\text{act,in}}} = \frac{2.27 \text{ hp}}{2.85 \text{ hp}} = \mathbf{79.8\%}$$

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8-66 Refrigerant-134a enters an adiabatic compressor at a specified state with a specified volume flow rate, and leaves at a specified state. The power input, the isentropic efficiency, the rate of exergy destruction, and the second-law efficiency are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Analysis (a) The properties of refrigerant at the inlet and exit states of the compressor are obtained from R-134a tables:

$$\left.\begin{array}{l} T_1 = -26°\text{C} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 234.68\ \text{kJ/kg} \\ s_1 = 0.9514\ \text{kJ/kg}\cdot\text{K} \\ v_1 = 0.18946\ \text{m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ T_2 = 50°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 286.69\ \text{kJ/kg} \\ s_2 = 0.9802\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ s_2 = s_1 = 0.9514\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} h_{2s} = 277.53\ \text{kJ/kg}$$

The mass flow rate of the refrigerant and the actual power input are

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{(0.45/60)\ \text{m}^3/\text{s}}{0.18946\ \text{m}^3/\text{kg}} = 0.03959\ \text{kg/s}$$

$$\dot{W}_{\text{act}} = \dot{m}(h_2 - h_1) = (0.03959\ \text{kg/s})(286.69 - 234.68)\text{kJ/kg} = \mathbf{2.059\ kW}$$

(b) The power input for the isentropic case and the isentropic efficiency are

$$\dot{W}_{\text{isen}} = \dot{m}(h_{2s} - h_1) = (0.03959\ \text{kg/s})(277.53 - 234.68)\text{kJ/kg} = 1.696\ \text{kW}$$

$$\eta_{\text{Comp,isen}} = \frac{\dot{W}_{\text{isen}}}{\dot{W}_{\text{act}}} = \frac{1.696\ \text{kW}}{2.059\ \text{kW}} = 0.8238 = \mathbf{82.4\%}$$

(c) The exergy destruction is

$$\dot{X}_{\text{dest}} = \dot{m}T_0(s_2 - s_1) = (0.03959\ \text{kg/s})(300\ \text{K})(0.9802 - 0.9514)\text{kJ/kg}\cdot\text{K} = \mathbf{0.3417\ kW}$$

The reversible power and the second-law efficiency are

$$\dot{W}_{\text{rev}} = \dot{W}_{\text{act}} - \dot{X}_{\text{dest}} = 2.059 - 0.3417 = 1.717\ \text{kW}$$

$$\eta_{\text{Comp,II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{act}}} = \frac{1.717\ \text{kW}}{2.059\ \text{kW}} = 0.8341 = \mathbf{83.4\%}$$

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8-67 Refrigerant-134a is condensed in a refrigeration system by rejecting heat to ambient air. The rate of heat rejected, the COP of the refrigeration cycle, and the rate of exergy destruction are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The properties of refrigerant at the inlet and exit states of the condenser are (from R134a tables)

$$\left.\begin{array}{l} P_1 = 700\ \text{kPa} \\ T_1 = 50°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 288.53\ \text{kJ/kg} \\ s_1 = 0.9954\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 700\ \text{kPa} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} h_2 = 88.82\ \text{kJ/kg} \\ s_2 = 0.3323\ \text{kJ/kg}\cdot\text{K} \end{array}$$

The rate of heat rejected in the condenser is

$$\dot{Q}_H = \dot{m}_R(h_1 - h_2) = (0.05\ \text{kg/s})(288.53 - 88.82)\text{kJ/kg} = \mathbf{9.985\ kW}$$

(b) From the definition of COP for a refrigerator,

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{\dot{Q}_L}{\dot{Q}_H - \dot{Q}_L} = \frac{6\ \text{kW}}{(9.985 - 6)\ \text{kW}} = \mathbf{1.506}$$

(c) The entropy generation and the exergy destruction in the condenser are

$$\dot{S}_{\text{gen}} = \dot{m}_R(s_2 - s_1) + \frac{\dot{Q}_H}{T_H}$$

$$= (0.05\ \text{kg/s})(0.3323 - 0.9954)\ \text{kJ/kg}\cdot\text{K} + \frac{9.985\ \text{kW}}{298\ \text{K}} = 0.0003516\ \text{kW/K}$$

$$\dot{X}_{\text{dest}} = T_0\dot{S}_{\text{gen}} = (298\ \text{K})(0.0003516\ \text{kJ/kg}\cdot\text{K}) = \mathbf{0.1048\ kW}$$

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8-68E Refrigerant-134a is evaporated in the evaporator of a refrigeration system. the rate of cooling provided, the rate of exergy destruction, and the second-law efficiency of the evaporator are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The rate of cooling provided is

$$\dot{Q}_L = \dot{m}(h_2 - h_1) = (0.08\,\text{lbm/s})(172.1 - 107.5)\text{Btu/lbm} = 5.162\,\text{Btu/s} = \mathbf{18{,}580\,Btu/h}$$

(b) The entropy generation and the exergy destruction are

$$\begin{aligned}\dot{S}_{\text{gen}} &= \dot{m}(s_2 - s_1) - \frac{\dot{Q}_L}{T_L} \\ &= (0.08\,\text{lbm/s})(0.4225 - 0.2851)\,\text{Btu/lbm}\cdot\text{R} - \frac{5.162\,\text{Btu/s}}{(50+460)\,\text{R}} \\ &= 0.0008691\,\text{Btu/s}\cdot\text{R}\end{aligned}$$

$$\dot{X}_{\text{dest}} = T_0\dot{S}_{\text{gen}} = (537\,\text{R})(0.0008691\,\text{Btu/s}\cdot\text{R}) = \mathbf{0.4667\,Btu/s}$$

(c) The exergy supplied (or expended) during this cooling process is the exergy decrease of the refrigerant as it evaporates in the evaporator:

$$\begin{aligned}\dot{X}_1 - \dot{X}_2 &= \dot{m}(h_1 - h_2) - \dot{m}T_0(s_1 - s_2) \\ &= -5.162 - (0.08\,\text{lbm/s})(537\,\text{R})(0.2851 - 0.4225)\,\text{Btu/lbm}\cdot\text{R} \\ &= 0.7400\,\text{Btu/s}\end{aligned}$$

The exergy efficiency is then

$$\eta_{\text{II,Evap}} = 1 - \frac{\dot{X}_{\text{dest}}}{\dot{X}_1 - \dot{X}_2} = 1 - \frac{0.4667}{0.7400} = 0.3693 = \mathbf{36.9\%}$$

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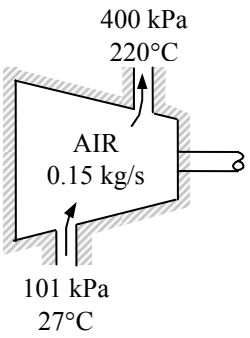
8-69 Air is compressed steadily by a compressor from a specified state to another specified state. The reversible power is to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1). From the air table (Table A-17)

$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.19\text{ kJ/kg}$$
$$s_1^o = 1.702\text{ kJ/kg}\cdot\text{K}$$

$$T_2 = 493\text{ K} \longrightarrow h_2 = 495.82\text{ kJ/kg}$$
$$s_2^o = 2.20499\text{ kJ/kg}\cdot\text{K}$$



Analysis The reversible (or minimum) power input is determined from the rate form of the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$
$$\dot{m}\psi_1 + \dot{W}_{\text{rev,in}} = \dot{m}\psi_2$$
$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta\text{ke}^{\nearrow 0} + \Delta\text{pe}^{\nearrow 0}]$$

where

$$s_2 - s_1 = s_2^o - s_1^o - R\ln\frac{P_2}{P_1}$$
$$= (2.205 - 1.702)\text{kJ/kg}\cdot\text{K} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{400\text{ kPa}}{101\text{ kPa}}$$
$$= 0.1080\text{ kJ/kg}\cdot\text{K}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (0.15\text{ kg/s})\big[(495.82 - 300.19)\text{kJ/kg} - (298\text{ K})(0.1080\text{ kJ/kg}\cdot\text{K})\big] = \mathbf{24.5\ kW}$$

Discussion Note that a minimum of 24.5 kW of power input is needed for this compression process.

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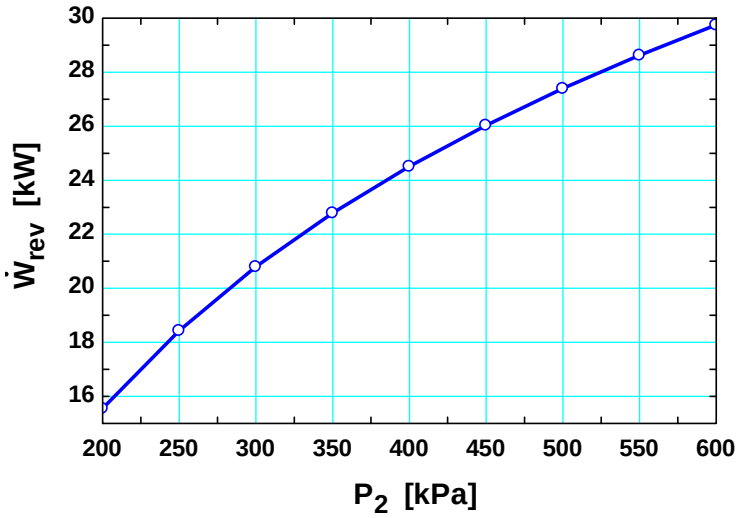
8-70 Problem 8-69 is reconsidered. The effect of compressor exit pressure on reversible power is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
T_1=27 [C]
P_1=101 [kPa]
m_dot = 0.15 [kg/s]
{P_2=400 [kPa]}
T_2=220 [C]
T_o=25 [C]
P_o=100 [kPa]
m_dot_in=m_dot

"Steady-flow conservation of mass"
m_dot_in = m_dot_out
h_1 =enthalpy(air,T=T_1)
h_2 = enthalpy(air, T=T_2)
W_dot_rev=m_dot_in*(h_2 - h_1 -(T_1+273.15)*(s_2-s_1))
s_1=entropy(air, T=T_1,P=P_1)
s_2=entropy(air,T=T_2,P=P_2)
```

P_2 [kPa]	W_{rev} [kW]
200	15.55
250	18.44
300	20.79
350	22.79
400	24.51
450	26.03
500	27.4
550	28.63
600	29.75



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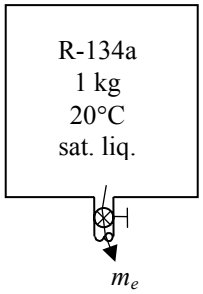
8-71 A rigid tank initially contains saturated liquid of refrigerant-134a. R-134a is released from the vessel until no liquid is left in the vessel. The exergy destruction associated with this process is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process. It can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Properties The properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} T_1 = 20°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \nu_1 = \nu_{f@20°\text{C}} = 0.0008161\ \text{m}^3/\text{kg} \\ u_1 = u_{f@20°\text{C}} = 78.86\ \text{kJ/kg} \\ s_1 = s_{f@20°\text{C}} = 0.30063\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 20°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \nu_2 = \nu_{g@20°\text{C}} = 0.035969\ \text{m}^3/\text{kg} \\ u_2 = u_{g@20°\text{C}} = 241.02\ \text{kJ/kg} \\ s_2 = s_e = s_{g@20°\text{C}} = 0.92234\ \text{kJ/kg}\cdot\text{K} \\ h_e = h_{g@20°\text{C}} = 261.59\ \text{kJ/kg} \end{array}$$



Analysis The volume of the container is

$$V = m_1\nu_1 = (1\,\text{kg})(0.0008161\,\text{m}^3/\text{kg}) = 0.0008161\ \text{m}^3$$

The mass in the container at the final state is

$$m_2 = \frac{V}{\nu_2} = \frac{0.0008161\,\text{m}^3}{0.035969\,\text{m}^3/\text{kg}} = 0.02269\ \text{kg}$$

The amount of mass leaving the container is

$$m_e = m_1 - m_2 = 1 - 0.02269 = 0.9773\ \text{kg}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on the system:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 + m_e s_e$$

Substituting,

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 (m_2 s_2 - m_1 s_1 + m_e s_e) \\ &= (293\ \text{K})(0.02269 \times 0.92234 - 1 \times 0.30063 + 0.9773 \times 0.92234) \\ &= \mathbf{182.2\ kJ} \end{aligned}$$

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8-72E An adiabatic rigid tank that is initially evacuated is filled by air from a supply line. The work potential associated with this process is to be determined.

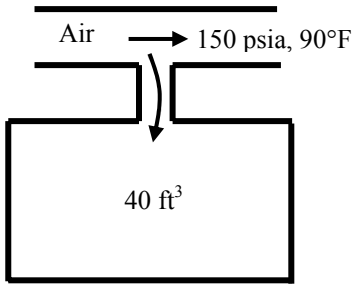
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process. It can be analyzed as a uniform-flow process since the state of fluid entering the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R, k = 1.4, and R = 0.06855 Btu/lbm·R = 0.3704 kPa·m^3/lbm·R (Table A-2Ea).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = m_2 u_2$$

Combining the two balances:

$$h_i = u_2 \longrightarrow c_p T_i = c_v T_2 \longrightarrow T_2 = \frac{c_p}{c_v} T_i = kT_i$$

Substituting,

$$T_2 = kT_i = (1.4)(550\ \text{R}) = 770\ \text{R}$$

The final mass in the tank is

$$m_2 = m_i = \frac{P\boldsymbol{V}}{RT_2} = \frac{(150\ \text{psia})(40\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(770\ \text{R})} = 21.04\ \text{lbm}$$

The work potential associated with this process is equal to the exergy destroyed during the process. The exergy destruction during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on the system:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2$$

$$S_{\text{gen}} = m_2 s_2 - m_i s_i$$

$$S_{\text{gen}} = m_2 (s_2 - s_i)$$

Substituting,

$$W_{\text{rev}} = X_{\text{destyroyed}} = m_2 T_0 (s_2 - s_i) = m_2 T_0 \left(c_p \ln \frac{T_2}{T_i} \right)$$

$$= (21.04\ \text{lbm})(540\ \text{R}) \left[(0.240\ \text{Btu/lbm}\cdot\text{R}) \ln \frac{770\ \text{R}}{550\ \text{R}} \right]$$

$$= \mathbf{917\ Btu}$$

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8-73E An rigid tank that is initially evacuated is filled by air from a supply line. The work potential associated with this process is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process. It can be analyzed as a uniform-flow process since the state of fluid entering the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and R = 0.06855 Btu/lbm·R = 0.3704 kPa·m^3/lbm·R (Table A-2Ea).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

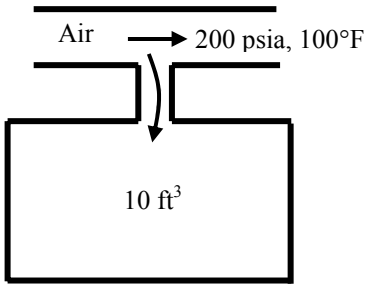
Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i - Q_{\text{out}} = m_2 u_2$$

$$Q_{\text{out}} = m_i h_i - m_2 u_2$$



Combining the two balances:

$$Q_{\text{out}} = m_2 (h_i - u_2)$$

The final mass in the tank is

$$m_2 = m_i = \frac{P\boldsymbol{V}}{RT_2} = \frac{(150 \text{ psia})(40 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(550 \text{ R})} = 29.45 \text{ lbm}$$

Substituting,

$$\begin{aligned} Q_{\text{out}} &= m_2 (h_i - u_2) = m_2 (c_p T_i - c_v T_i) = m_2 T_i (c_p - c_v) = m_2 T_i R \\ &= (29.45 \text{ lbm})(550 \text{ R})(0.06855 \text{ Btu/lbm} \cdot \text{R}) \\ &= 1110 \text{ Btu} \end{aligned}$$

The work potential associated with this process is equal to the exergy destroyed during the process. The exergy destruction during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on the system:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$m_i s_i - \frac{Q_{\text{out}}}{T_0} + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2$$

$$S_{\text{gen}} = m_2 s_2 - m_i s_i + \frac{Q_{\text{out}}}{T_0}$$

$$S_{\text{gen}} = m_2 (s_2 - s_i) + \frac{Q_{\text{out}}}{T_0}$$

Noting that both the temperature and pressure in the tank is same as those in the supply line at the final state, substituting gives,

$$\begin{aligned} W_{\text{rev}} &= X_{\text{destroyed}} = T_0 \left[m_2 (s_2 - s_i) + \frac{Q_{\text{out}}}{T_0} \right] \\ &= T_0 \left(0 + \frac{Q_{\text{out}}}{T_0} \right) = T_0 \left(\frac{Q_{\text{out}}}{T_0} \right) = Q_{\text{out}} = \mathbf{1110 \text{ Btu}} \end{aligned}$$

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8-74 Steam expands in a turbine steadily at a specified rate from a specified state to another specified state. The power potential of the steam at the inlet conditions and the reversible power output are to be determined.

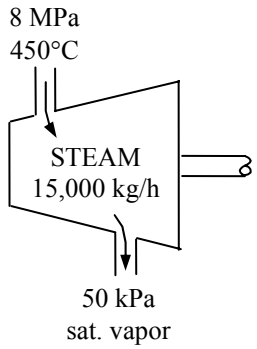
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C.

Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{matrix} P_1 = 8\ \text{MPa} \\ T_1 = 450°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3273.3\ \text{kJ/kg} \\ s_1 = 6.5579\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 50\ \text{kPa} \\ \text{sat. vapor} \end{matrix}\right\} \begin{matrix} h_2 = 2645.2\ \text{kJ/kg} \\ s_2 = 7.5931\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_0 = 100\ \text{kPa} \\ T_0 = 25°\text{C} \end{matrix}\right\} \begin{matrix} h_0 \cong h_{f@25°\text{C}} = 104.83\ \text{kJ/kg} \\ s_0 \cong s_{f@25°\text{C}} = 0.36723\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$



Analysis (*a*) The power potential of the steam at the inlet conditions is equivalent to its exergy at the inlet state,

$$\dot{\Psi} = \dot{m}\psi_1 = \dot{m}\left(h_1 - h_0 - T_0(s_1 - s_0) + \frac{V_1^{2\,\cancel{\nearrow 0}}}{2} + gz_1^{\cancel{\nearrow 0}} \right) = \dot{m}\big(h_1 - h_0 - T_0(s_1 - s_0)\big)$$

$$= (15{,}000/3600\ \text{kg/s})\big[(3273.3 - 104.83)\text{kJ/kg} - (298\ \text{K})(6.5579 - 0.36723)\text{kJ/kg}\cdot\text{K}\big]$$

$$= \mathbf{5515\ kW}$$

(*b*) The power output of the turbine if there were no irreversibilities is the reversible power, is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\,\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\,\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 = \dot{W}_{\text{rev,out}} + \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,out}} = \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta\text{ke}^{\nearrow 0} - \Delta\text{pe}^{\nearrow 0}]$$

Substituting,

$$\dot{W}_{\text{rev,out}} = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2)]$$

$$= (15{,}000/3600\ \text{kg/s})\big[(3273.3 - 2645.2)\ \text{kJ/kg} - (298\ \text{K})(6.5579 - 7.5931)\ \text{kJ/kg}\cdot\text{K}\big]$$

$$= \mathbf{3902\ kW}$$

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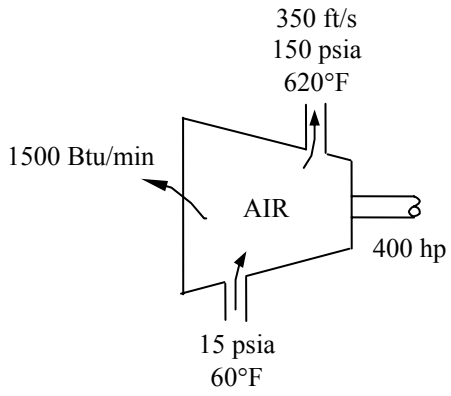
8-75E Air is compressed steadily by a 400-hp compressor from a specified state to another specified state while being cooled by the ambient air. The mass flow rate of air and the part of input power that is used to just overcome the irreversibilities are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** Potential energy changes are negligible. **3** The temperature of the surroundings is given to be 60°F.

Properties The gas constant of air is $R = 0.06855$ Btu/lbm.R (Table A-1E). From the air table (Table A-17E)

$$T_1 = 520\ \text{R} \ \Big\} \ \begin{matrix} h_1 = 124.27\ \text{Btu/lbm} \\ s_1^{\text{o}} = 0.59173\ \text{Btu/lbm}\cdot\text{R} \end{matrix}$$

$$T_2 = 1080\ \text{R} \ \Big\} \ \begin{matrix} h_2 = 260.97\ \text{Btu/lbm} \\ s_2^{\text{o}} = 0.76964\ \text{Btu/lbm}\cdot\text{R} \end{matrix}$$



Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{a,\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) + \dot{Q}_{\text{out}} \ \rightarrow \ \dot{W}_{a,\text{in}} - \dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate of the refrigerant becomes

$$(400\ \text{hp})\left(\frac{0.7068\ \text{Btu/s}}{1\ \text{hp}}\right) - (1500/60\ \text{Btu/s}) = \dot{m}\left(260.97 - 124.27 + \frac{(350\ \text{ft/s})^2 - 0}{2}\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right)$$

It yields $\dot{m} = \mathbf{1.852\ lbm/s}$

(*b*) The portion of the power output that is used just to overcome the irreversibilities is equivalent to exergy destruction, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \ \rightarrow \ \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0}$$

where

$$s_2 - s_1 = s_2^0 - s_1^0 - R\ln\frac{P_2}{P_1} = (0.76964 - 0.59173)\ \text{Btu/lbm} - (0.06855\ \text{Btu/lbm.R})\ln\frac{150\ \text{psia}}{15\ \text{psia}} = 0.02007\ \text{Btu/lbm.R}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0\left(\dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0}\right)$$

$$= (520\ \text{R})\left((1.852\ \text{lbm/s})(0.02007\ \text{Btu/lbm}\cdot\text{R}) + \frac{1500/60\ \text{Btu/s}}{520\ \text{R}}\right)\left(\frac{1\ \text{hp}}{0.7068\ \text{Btu/s}}\right) = \mathbf{62.72\ hp}$$

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8-76 Hot combustion gases are accelerated in an adiabatic nozzle. The exit velocity and the decrease in the exergy of the gases are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** The combustion gases are ideal gases with constant specific heats.

Properties The constant pressure specific heat and the specific heat ratio are given to be $c_p = 1.15$ kJ/kg.K and $k = 1.3$. The gas constant R is determined from

$$R = c_p - c_v = c_p - c_p / k = c_p(1-1/k) = (1.15\,\text{kJ/kg}\cdot\text{K})(1-1/1.3) = 0.2654\,\text{kJ/kg}\cdot\text{K}$$

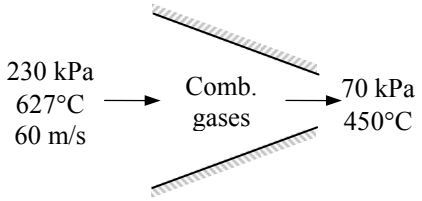
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2}$$



Then the exit velocity becomes

$$V_2 = \sqrt{2c_p(T_1 - T_2) + V_1^2}$$

$$= \sqrt{2(1.15\,\text{kJ/kg}\cdot\text{K})(627-450)\text{K}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right) + (60\,\text{m/s})^2}$$

$$= \mathbf{641\,m/s}$$

(*b*) The decrease in exergy of combustion gases is simply the difference between the initial and final values of flow exergy, and is determined to be

$$\psi_1 - \psi_2 = w_{\text{rev}} = h_1 - h_2 - \Delta\text{ke} - \Delta\text{pe}^{\nearrow 0} + T_0(s_2 - s_1) = c_p(T_1 - T_2) + T_0(s_2 - s_1) - \Delta\text{ke}$$

where

$$\Delta\text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(641\,\text{m/s})^2 - (60\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 203.6\,\text{kJ/kg}$$

and

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (1.15\,\text{kJ/kg}\cdot\text{K})\ln\frac{723\,\text{K}}{900\,\text{K}} - (0.2654\,\text{kJ/kg}\cdot\text{K})\ln\frac{70\,\text{kPa}}{230\,\text{kPa}}$$

$$= 0.06386\,\text{kJ/kg}\cdot\text{K}$$

Substituting,

$$\text{Decrease in exergy} = \psi_1 - \psi_2$$

$$= (1.15\,\text{kJ/kg}\cdot\text{K})(627-450)°\text{C} + (293\,\text{K})(0.06386\,\text{kJ/kg}\cdot\text{K}) - 203.6\,\text{kJ/kg}$$

$$= \mathbf{18.7\,kJ/kg}$$

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8-77 Steam is accelerated in an adiabatic nozzle. The exit velocity of the steam, the isentropic efficiency, and the exergy destroyed within the nozzle are to be determined.

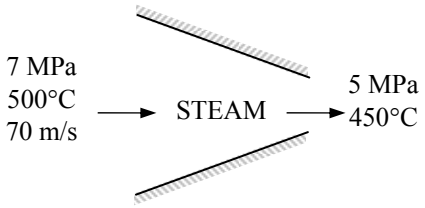
Assumptions **1** The nozzle operates steadily. **2** The changes in potential energies are negligible.

Properties The properties of steam at the inlet and the exit of the nozzle are (Tables A-4 through A-6)

$$\left.\begin{matrix} P_1 = 7\ \text{MPa} \\ T_1 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3411.4\ \text{kJ/kg} \\ s_1 = 6.8000\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 5\ \text{MPa} \\ T_2 = 450°\text{C} \end{matrix}\right\} \begin{matrix} h_2 = 3317.2\ \text{kJ/kg} \\ s_2 = 6.8210\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_{2s} = 5\ \text{MPa} \\ s_{2s} = s_1 \end{matrix}\right\} h_{2s} = 3302.0\ \text{kJ/kg}$$



Analysis (*a*) We take the nozzle to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\begin{matrix}\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}\end{matrix}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\begin{matrix}\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}\end{matrix}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + \text{V}_2^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Then the exit velocity becomes

$$V_2 = \sqrt{2(h_1 - h_2) + \mathbf{V}_1^2} = \sqrt{2(3411.4 - 3317.2)\ \text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right) + (70\ \text{m/s})^2} = \mathbf{439.6\ m/s}$$

(*b*) The exit velocity for the isentropic case is determined from

$$V_{2s} = \sqrt{2(h_1 - h_{2s}) + \mathbf{V}_1^2} = \sqrt{2(3411.4 - 3302.0)\ \text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right) + (70\ \text{m/s})^2} = 472.9\ \text{m/s}$$

Thus,

$$\eta_N = \frac{V_2^2/2}{V_{2s}^2/2} = \frac{(439.6\ \text{m/s})^2/2}{(472.9\ \text{m/s})^2/2} = \mathbf{86.4\%}$$

(*c*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation S_{gen} is determined from an entropy balance on the actual nozzle. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\begin{matrix}\text{Rate of net entropy transfer}\\ \text{by heat and mass}\end{matrix}} + \underbrace{\dot{S}_{\text{gen}}}_{\begin{matrix}\text{Rate of entropy}\\ \text{generation}\end{matrix}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0}}_{\begin{matrix}\text{Rate of change}\\ \text{of entropy}\end{matrix}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) \text{ or } \quad s_{\text{gen}} = s_2 - s_1$$

Substituting, the exergy destruction in the nozzle on a unit mass basis is determined to be

$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0(s_2 - s_1) = (298\ \text{K})(6.8210 - 6.8000)\text{kJ/kg}\cdot\text{K} = \mathbf{6.28\ kJ/kg}$$

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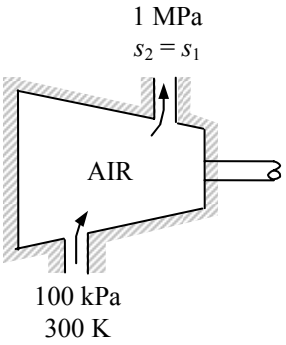
8-78 Air is compressed in a steady-flow device isentropically. The work done, the exit exergy of compressed air, and the exergy of compressed air after it is cooled to ambient temperature are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply. **3** The environment temperature and pressure are given to be 300 K and 100 kPa. **4** The kinetic and potential energies are negligible.

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1). The constant pressure specific heat and specific heat ratio of air at room temperature are c_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2).

Analysis (a) From the constant specific heats ideal gas isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\ \text{K})\left(\frac{1000\ \text{kPa}}{100\ \text{kPa}}\right)^{0.4/1.4} = 579.2\ \text{K}$$



For a steady-flow isentropic compression process, the work input is determined from

$$\begin{aligned} w_{\text{comp,in}} &= \frac{kRT_1}{k-1}\left\{(P_2/P_1)^{(k-1)/k} - 1\right\} \\ &= \frac{(1.4)(0.287\text{kJ/kg}\cdot\text{K})(300\text{K})}{1.4-1}\left\{(1000/100)^{0.4/1.4} - 1\right\} \\ &= \mathbf{280.5\ kJ/kg} \end{aligned}$$

(*b*) The exergy of air at the compressor exit is simply the flow exergy at the exit state,

$$\begin{aligned} \psi_2 &= h_2 - h_0 - T_0(s_2 - s_0)^{\nearrow^0} + \frac{V_2^2}{2}^{\nearrow^0} + gz_2^{\nearrow^0} \quad \text{(since the proccess 0 - 2 is isentropic)} \\ &= c_p(T_2 - T_0) \\ &= (1.005\ \text{kJ/kg.K})(579.2 - 300)\text{K} = \mathbf{280.6\ kJ/kg} \end{aligned}$$

which is the same as the compressor work input. This is not surprising since the compression process is reversible.

(*c*) The exergy of compressed air at 1 MPa after it is cooled to 300 K is again the flow exergy at that state,

$$\begin{aligned} \psi_3 &= h_3 - h_0 - T_0(s_3 - s_0) + \frac{V_3^2}{2}^{\nearrow^0} + gz_3^{\nearrow^0} \\ &= c_p(T_3 - T_0)^{\nearrow^0} - T_0(s_3 - s_0) \quad (\text{since } T_3 = T_0 = 300\ \text{K}) \\ &= -T_0(s_3 - s_0) \end{aligned}$$

where

$$s_3 - s_0 = c_p \ln\frac{T_3}{T_0}^{\nearrow^0} - R\ln\frac{P_3}{P_0} = -R\ln\frac{P_3}{P_0} = -(0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{1000\ \text{kPa}}{100\ \text{kPa}} = -0.661\ \text{kJ/kg.K}$$

Substituting,

$$\psi_3 = -(300\ \text{K})(-0.661\ \text{kJ/kg.K}) = \mathbf{198\ kJ/kg}$$

Note that the exergy of compressed air decreases from 280.6 to 198 as it is cooled to ambient temperature.

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8-79 A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered the tank and the exergy destroyed during this process are to be determined.

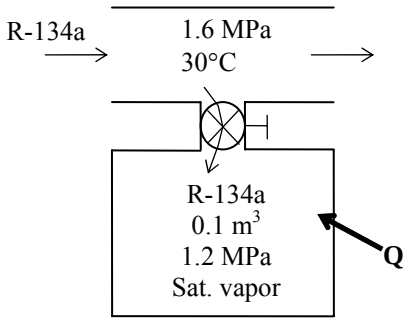
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1.2\ \text{MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@1.2\ \text{MPa}} = 0.01672\ \text{m}^3/\text{kg} \\ u_1 = u_{g@1.2\ \text{MPa}} = 253.81\ \text{kJ/kg} \\ s_1 = s_{g@1.2\ \text{MPa}} = 0.91303\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 1.4\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_2 = v_{f@1.4\ \text{MPa}} = 0.0009166\ \text{m}^3/\text{kg} \\ u_2 = u_{f@1.4\ \text{MPa}} = 125.94\ \text{kJ/kg} \\ s_2 = s_{f@1.4\ \text{MPa}} = 0.45315\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.6\ \text{MPa} \\ T_i = 30°\text{C} \end{array}\right\} \begin{array}{l} h_i = 93.56\ \text{kJ/kg} \\ s_i = 0.34554\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis We take the tank as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

(*a*) The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.1\ \text{m}^3}{0.01672\ \text{m}^3/\text{kg}} = 5.983\ \text{kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.1\ \text{m}^3}{0.0009166\ \text{m}^3/\text{kg}} = 109.10\ \text{kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 109.10 - 5.983 = \mathbf{103.11\ kg}$$

The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(103.11\ \text{kg})(93.56\ \text{kJ/kg}) + (109.10)(125.94\ \text{kJ/kg}) - (5.983\ \text{kg})(253.81\ \text{kJ/kg}) \\ &= 2573\ \text{kJ} \end{aligned}$$

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(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on an *extended system* that includes the tank and its immediate surroundings so that the boundary temperature of the extended system is the surroundings temperature T_{surr} at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} + m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}} \quad \text{Substituting, the exergy destruction}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i - \frac{Q_{\text{in}}}{T_0}$$

is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 \left[m_2 s_2 - m_1 s_1 - m_i s_i - \frac{Q_{\text{in}}}{T_0} \right] \\ &= (318\ \text{K})\left[109.10 \times 0.45315 - 5.983 \times 0.91303 - 103.11 \times 0.34554 - (2573\ \text{kJ})/(318\ \text{K})\right] \\ &= \mathbf{80.3\ kJ} \end{aligned}$$

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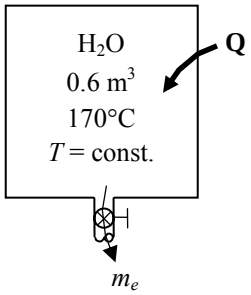
8-80 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer, the reversible work, and the exergy destruction during this process are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 170°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@170°\text{C}} = 0.001114\ \text{m}^3/\text{kg} \\ u_1 = u_{f@170°\text{C}} = 718.20\ \text{kJ/kg} \\ s_1 = s_{f@170°\text{C}} = 2.0417\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_e = 170°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_e = h_{f@170°\text{C}} = 719.08\ \text{kJ/kg} \\ s_e = s_{f@170°\text{C}} = 2.0417\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{0.6\ \text{m}^3}{0.001114\ \text{m}^3/\text{kg}} = 538.47\ \text{kg}$$

$$m_2 = \frac{1}{2} m_1 = \frac{1}{2}(538.47\ \text{kg}) = 269.24\ \text{kg} = m_e$$

Now we determine the final internal energy and entropy,

$$v_2 = \frac{V}{m_2} = \frac{0.6\ \text{m}^3}{269.24\ \text{kg}} = 0.002229\ \text{m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002229 - 0.001114}{0.24260 - 0.001114} = 0.004614$$

$$\left.\begin{array}{l} T_2 = 170°\text{C} \\ x_2 = 0.004614 \end{array}\right\} \begin{array}{l} u_2 = u_f + x_2 u_{fg} = 718.20 + (0.004614)(1857.5) = 726.77\ \text{kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 2.0417 + (0.004614)(4.6233) = 2.0630\ \text{kJ/kg}\cdot\text{K} \end{array}$$

The heat transfer during this process is determined by substituting these values into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (269.24\ \text{kg})(719.08\ \text{kJ/kg}) + (269.24\ \text{kg})(726.77\ \text{kJ/kg}) - (538.47\ \text{kg})(718.20\ \text{kJ/kg}) \\ &= \mathbf{2545\ kJ} \end{aligned}$$

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(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on an *extended system* that includes the tank and the region between the tank and the source so that the boundary temperature of the extended system at the location of heat transfer is the source temperature T_{source} at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} - m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 \left[m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_{\text{source}}} \right] \\ &= (298\ \text{K})\left[269.24 \times 2.0630 - 538.47 \times 2.0417 + 269.24 \times 2.0417 - (2545\ \text{kJ})/(523\ \text{K})\right] \\ &= \mathbf{141.2\ kJ} \end{aligned}$$

For processes that involve no actual work, the reversible work output and exergy destruction are identical. Therefore,

$$X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act,out}} \ \rightarrow \ W_{\text{rev,out}} = X_{\text{destroyed}} = \mathbf{141.2\,kJ}$$

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8-81E An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to 20 psia. The amount of electrical work done and the exergy destroys are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats. **5** The environment temperature is given to be 70°F.

Properties The gas constant of air is $R = 0.3704$ psia.ft^3/lbm.R (Table A-1E). The properties of air are (Table A-17E)

$$T_e = 640\ \text{R} \longrightarrow h_e = 153.09\ \text{Btu/lbm}$$

$$T_1 = 640\ \text{R} \longrightarrow u_1 = 109.21\ \text{Btu/lbm} \qquad T_2 = 640\ \text{R} \longrightarrow u_2 = 109.21\ \text{Btu/lbm}$$

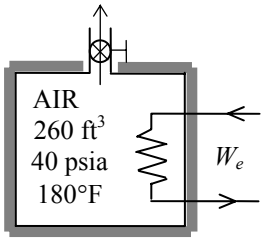
Analysis We take the tank as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \ \ (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$



The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(40\ \text{psia})(260\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(640\ \text{R})} = 43.86\ \text{lbm}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(20\ \text{psia})(260\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(640\ \text{R})} = 21.93\ \text{lbm}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 43.86 - 21.93 = 21.93\ \text{lbm}$$

$$\begin{aligned} W_{\text{e,in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (21.93\ \text{lbm})(153.09\ \text{Btu/lbm}) + (21.93\ \text{lbm})(109.21\ \text{Btu/lbm}) - (43.86\ \text{lbm})(109.21\ \text{Btu/lbm}) \\ &= \mathbf{962\ Btu} \end{aligned}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} is determined from an entropy balance on the insulated tank. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\begin{aligned} -m_e s_e + S_{\text{gen}} &= \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}} \\ S_{\text{gen}} &= m_2 s_2 - m_1 s_1 + m_e s_e \\ &= m_2 s_2 - m_1 s_1 + (m_1 - m_2) s_e \\ &= m_2 (s_2 - s_e) - m_1 (s_1 - s_e) \end{aligned}$$

Assuming a constant average pressure of (40 + 20) / 2 = 30 psia for the exit stream, the entropy changes are determined to be

$$s_2 - s_e = c_p \ln \frac{\cancelto{0}{T_2}}{T_e} - R \ln \frac{P_2}{P_e} = -(0.06855\ \text{Btu/lbm}\cdot\text{R}) \ln \frac{20\ \text{psia}}{30\ \text{psia}} = 0.02779\ \text{Btu/lbm}\cdot\text{R}$$

$$s_1 - s_e = c_p \ln \frac{\cancelto{0}{T_1}}{T_e} - R \ln \frac{P_1}{P_e} = -(0.06855\ \text{Btu/lbm}\cdot\text{R}) \ln \frac{40\ \text{psia}}{30\ \text{psia}} = -0.01972\ \text{Btu/lbm}\cdot\text{R}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 [m_2 (s_2 - s_e) - m_1 (s_1 - s_e)] \\ &= (530\ \text{R})\big[(21.93\ \text{lbm})(0.02779\ \text{Btu/lbm}\cdot\text{R}) - (43.86\ \text{lbm})(-0.01972\ \text{Btu/lbm}\cdot\text{R})\big] = \mathbf{782\ Btu} \end{aligned}$$

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8-82 A cylinder initially contains helium gas at a specified pressure and temperature. A valve is opened, and helium is allowed to escape until its volume decreases by half. The work potential of the helium at the initial state and the exergy destroyed during the process are to be determined.

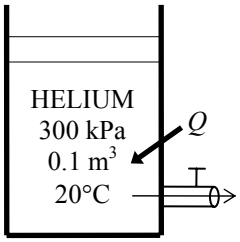
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The tank is insulated and thus heat transfer is negligible. **5** Helium is an ideal gas with constant specific heats.

Properties The gas constant of helium is $R = 2.0769$ kPa.m^3/kg.K = 2.0769 kJ/kg.K. The specific heats of helium are c_p = 5.1926 kJ/kg.K and c_v = 3.1156 kJ/kg.K (Table A-2).

Analysis (*a*) From the ideal gas relation, the initial and the final masses in the cylinder are determined to be

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(300\,\text{kPa})(0.1\,\text{m}^3)}{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})} = 0.0493\,\text{kg}$$

$$m_e = m_2 = m_1/2 = 0.0493/2 = 0.0247\,\text{kg}$$



The work potential of helium at the initial state is simply the initial exergy of helium, and is determined from the closed-system exergy relation,

$$\Phi_1 = m_1\phi = m_1[(u_1 - u_0) - T_0(s_1 - s_0) + P_0(v_1 - v_0)]$$

where

$$v_1 = \frac{RT_1}{P_1} = \frac{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{300\,\text{kPa}} = 2.0284\,\text{m}^3/\text{kg}$$

$$v_0 = \frac{RT_0}{P_0} = \frac{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{95\,\text{kPa}} = 6.405\,\text{m}^3/\text{kg}$$

and

$$\begin{aligned} s_1 - s_0 &= c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0} \\ &= (5.1926\,\text{kJ/kg}\cdot\text{K})\ln\frac{293\,\text{K}}{293\,\text{K}} - (2.0769\,\text{kJ/kg}\cdot\text{K})\ln\frac{300\,\text{kPa}}{95\,\text{kPa}} \\ &= -2.388\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Thus,

$$\begin{aligned} \Phi_1 &= (0.0493\,\text{kg})\{(3.1156\,\text{kJ/kg}\cdot\text{K})(20-20)°\text{C} - (293\,\text{K})(-2.388\,\text{kJ/kg}\cdot\text{K}) \\ &\quad + (95\,\text{kPa})(2.0284-6.405)\text{m}^3/\text{kg}[\text{kJ/kPa}\cdot\text{m}^3]\} \\ &= \mathbf{14.0\,kJ} \end{aligned}$$

(*b*) We take the cylinder as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - m_e h_e + W_{\text{b,in}} = m_2 u_2 - m_1 u_1$$

Combining the two relations gives

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$$\begin{aligned} Q_{\text{in}} &= (m_1 - m_2)h_e + m_2u_2 - m_1u_1 - W_{\text{b,in}} \\ &= (m_1 - m_2)h_e + m_2h_2 - m_1h_1 \\ &= (m_1 - m_2 + m_2 - m_1)h_1 \\ &= 0 \end{aligned}$$

since the boundary work and ΔU combine into ΔH for constant pressure expansion and compression processes.

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} can be determined from an entropy balance on the cylinder. Noting that the pressure and temperature of helium in the cylinder are maintained constant during this process and heat transfer is zero, it gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$\begin{aligned} -m_e s_e + S_{\text{gen}} &= \Delta S_{\text{cylinder}} = (m_2 s_2 - m_1 s_1)_{\text{cylinder}} \\ S_{\text{gen}} &= m_2 s_2 - m_1 s_1 + m_e s_e \\ &= m_2 s_2 - m_1 s_1 + (m_1 - m_2)s_e \\ &= (m_2 - m_1 + m_1 - m_2)s_1 \\ &= 0 \end{aligned}$$

since the initial, final, and the exit states are identical and thus $s_e = s_2 = s_1$. Therefore, this discharge process is reversible, and

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = \mathbf{0}$$

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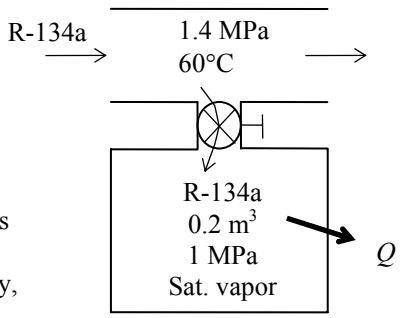
8-83 A rigid tank initially contains saturated R-134a vapor at a specified pressure. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The amount of heat transfer with the surroundings and the exergy destruction are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is from the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1\,\text{MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} u_1 = u_{g@1\,\text{MPa}} = 250.68\,\text{kJ/kg} \\ s_1 = s_{g@1\,\text{MPa}} = 0.91558\,\text{kJ/kg}\cdot\text{K} \\ \nu_1 = \nu_{g@1\,\text{MPa}} = 0.020313\,\text{m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.4\,\text{MPa} \\ T_i = 60°\text{C} \end{array}\right\} \begin{array}{l} h_i = 285.47\,\text{kJ/kg} \\ s_i = 0.93889\,\text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i - Q_{\text{out}} = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{V}{\nu_1} = \frac{0.2\,\text{m}^3}{0.020313\,\text{m}^3/\text{kg}} = 9.846\,\text{kg}$$

$$m_2 = m_f + m_g = \frac{V_f}{\nu_f} + \frac{V_g}{\nu_g} = \frac{0.1\,\text{m}^3}{0.0008934\,\text{m}^3/\text{kg}} + \frac{0.1\,\text{m}^3}{0.016715\,\text{m}^3/\text{kg}} = 111.93 + 5.983 = 117.91\,\text{kg}$$

$$U_2 = m_2 u_2 = m_f u_f + m_g u_g = 111.93 \times 116.70 + 5.983 \times 253.81 = 14{,}581\,\text{kJ}$$

$$S_2 = m_2 s_2 = m_f s_f + m_g s_g = 111.93 \times 0.42441 + 5.983 \times 0.91303 = 52.967\,\text{kJ/K}$$

Then from the mass and energy balances,

$$m_i = m_2 - m_1 = 117.91 - 9.846 = 108.06\,\text{kg}$$

The heat transfer during this process is determined from the energy balance to be

$$Q_{\text{out}} = m_i h_i - m_2 u_2 + m_1 u_1 = 108.06 \times 285.47 - 14{,}581 + 9.846 \times 250.68 = \mathbf{18{,}737\ kJ}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on an *extended system* that includes the cylinder and its immediate surroundings so that the boundary temperature of the extended system is the surroundings temperature T_{surr} at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i + \frac{Q_{\text{out}}}{T_0}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 \left[m_2 s_2 - m_1 s_1 - m_i s_i + \frac{Q_{\text{out}}}{T_0} \right] \\ &= (298\,\text{K})[52.967 - 9.846 \times 0.91558 - 108.06 \times 0.93889 + 18{,}737/298] \\ &= \mathbf{1599\ kJ} \end{aligned}$$

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8-84 An insulated cylinder initially contains saturated liquid-vapor mixture of water. The cylinder is connected to a supply line, and the steam is allowed to enter the cylinder until all the liquid is vaporized. The amount of steam that entered the cylinder and the exergy destroyed are to be determined.

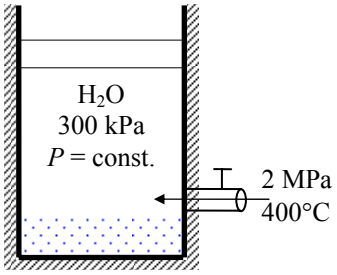
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **4** The device is insulated and thus heat transfer is negligible.

Properties The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 300\ \text{kPa} \\ x_1 = 13/15 = 0.8667 \end{array}\right\} \begin{array}{l} h_1 = h_f + x_1 h_{fg} = 561.43 + 0.8667 \times 2163.5 = 2436.5\ \text{kJ/kg} \\ s_1 = s_f + x_1 s_{fg} = 1.6716 + 0.8667 \times 5.3200 = 6.2824\ \text{kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 300\ \text{kPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_2 = h_{g@300\ \text{kPa}} = 2724.9\ \text{kJ/kg} \\ s_2 = s_{g@300\ \text{kPa}} = 6.9917\ \text{kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_i = 2\ \text{MPa} \\ T_i = 400°\text{C} \end{array}\right\} \begin{array}{l} h_i = 3248.4\ \text{kJ/kg} \\ s_i = 7.1292\ \text{kJ/kg} \cdot \text{K} \end{array}$$



Analysis (*a*) We take the cylinder as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this unsteady-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

Combining the two relations gives $0 = W_{\text{b,out}} - (m_2 - m_1) h_i + m_2 u_2 - m_1 u_1$

or, $$0 = -(m_2 - m_1) h_i + m_2 h_2 - m_1 h_1$$

since the boundary work and ΔU combine into ΔH for constant pressure expansion and compression processes. Solving for m_2 and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3248.4 - 2436.5)\text{kJ/kg}}{(3248.4 - 2724.9)\text{kJ/kg}} (15\ \text{kg}) = 23.27\ \text{kg}$$

Thus, $m_i = m_2 - m_1 = 23.27 - 15 = \mathbf{8.27\ kg}$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} is determined from an entropy balance on the insulated cylinder,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$m_i s_i + S_{\text{gen}} = \Delta S_{\text{system}} = m_2 s_2 - m_1 s_1$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 [m_2 s_2 - m_1 s_1 - m_i s_i] \\ &= (298\ \text{K})(23.27 \times 6.9917 - 15 \times 6.2824 - 8.27 \times 7.1292) \\ &= \mathbf{2832\ kJ} \end{aligned}$$

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8-85 Each member of a family of four takes a shower every day. The amount of exergy destroyed by this family per year is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The kinetic and potential energies are negligible. **3** Heat losses from the pipes, mixing section are negligible and thus $\dot{Q} \cong 0$. **4** Showers operate at maximum flow conditions during the entire shower. **5** Each member of the household takes a shower every day. **6** Water is an incompressible substance with constant properties at room temperature. **7** The efficiency of the electric water heater is 100%.

Properties The density and specific heat of water are at room temperature are $\rho = 997$ kg/m^3 and $c = 4.18$ kJ/kg.°C (Table A-3).

Analysis The mass flow rate of water at the shower head is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (0.997\,\text{kg/L})(10\,\text{L/min}) = 9.97\,\text{kg/min}$$

The mass balance for the mixing chamber can be expressed in the rate form as

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \ \rightarrow \ \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

where the subscript 1 denotes the cold water stream, 2 the hot water stream, and 3 the mixture.

The rate of entropy generation during this process can be determined by applying the rate form of the entropy balance on a system that includes the electric water heater and the mixing chamber (the T-elbow). Noting that there is no entropy transfer associated with work transfer (electricity) and there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not 0\ \text{(steady)}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} = 0 \quad \text{(since } Q = 0 \text{ and work is entropy free)}$$

$$\dot{S}_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

Noting from mass balance that $\dot{m}_1 + \dot{m}_2 = \dot{m}_3$ and $s_2 = s_1$ since hot water enters the system at the same temperature as the cold water, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_3 s_3 - (\dot{m}_1 + \dot{m}_2) s_1 = \dot{m}_3 (s_3 - s_1) = \dot{m}_3 c_p \ln \frac{T_3}{T_1}$$

$$= (9.97\,\text{kg/min})(4.18\,\text{kJ/kg.K}) \ln \frac{42 + 273}{15 + 273} = 3.735\,\text{kJ/min.K}$$

Noting that 4 people take a 6-min shower every day, the amount of entropy generated per year is

$$S_{\text{gen}} = (\dot{S}_{\text{gen}}) \Delta t (\text{No. of people})(\text{No. of days})$$
$$= (3.735\,\text{kJ/min.K})(6\,\text{min/person} \cdot \text{day})(4\,\text{persons})(365\,\text{days/year})$$
$$= 32{,}715\,\text{kJ/K} \ \text{(per year)}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\,\text{K})(32{,}715\,\text{kJ/K}) = \mathbf{9{,}749{,}000\ kJ}$$

Discussion The value above represents the exergy destroyed within the water heater and the T-elbow in the absence of any heat losses. It does not include the exergy destroyed as the shower water at 42°C is discarded or cooled to the outdoor temperature. Also, an entropy balance on the mixing chamber alone (hot water entering at 55°C instead of 15°C) will exclude the exergy destroyed within the water heater.

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8-86 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of exergy destruction are to be determined.

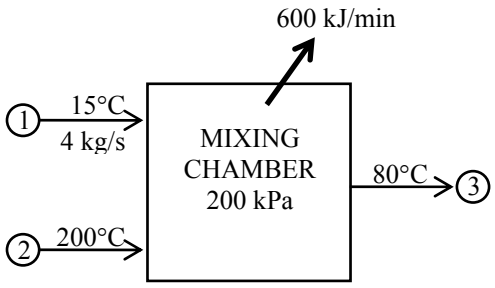
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.23°\text{C}$, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From Tables A-4 through A-6,

$$\left.\begin{matrix} P_1 = 200\ \text{kPa} \\ T_1 = 15°\text{C} \end{matrix}\right\} \begin{matrix} h_1 \cong h_{f@15°\text{C}} = 62.98\ \text{kJ/kg} \\ s_1 \cong s_{f@15°\text{C}} = 0.22447\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 200\ \text{kPa} \\ T_2 = 200°\text{C} \end{matrix}\right\} \begin{matrix} h_2 = 2870.4\ \text{kJ/kg} \\ s_2 = 7.5081\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_3 = 200\ \text{kPa} \\ T_3 = 80°\text{C} \end{matrix}\right\} \begin{matrix} h_3 \cong h_{f@80°\text{C}} = 335.02\ \text{kJ/kg} \\ s_3 \cong s_{f@80°\text{C}} = 1.0756\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$



Analysis (a) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{m}_2 h_2 = \dot{Q}_{\text{out}} + \dot{m}_3 h_3$$

Combining the two relations gives $\dot{Q}_{\text{out}} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2)h_3 = \dot{m}_1(h_1 - h_3) + \dot{m}_2(h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1(h_1 - h_3)}{h_2 - h_3} = \frac{(600/60\ \text{kJ/s}) - (4\ \text{kg/s})(62.98 - 335.02)\text{kJ/kg}}{(2870.4 - 335.02)\text{kJ/kg}} = \mathbf{0.429\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 4 + 0.429 = 4.429\ \text{kg/s}$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} is determined from an entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \ \rightarrow \dot{S}_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{\text{out}}}{T_0}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} \dot{X}_{\text{destroyed}} &= T_0 \dot{S}_{\text{gen}} = T_0\left(\dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{out}}{T_{b,surr}}\right) \\ &= (298\ \text{K})(4.429 \times 1.0756 - 0.429 \times 7.5081 - 4 \times 0.22447 + 10/298)\text{kW/K} \\ &= \mathbf{202\ kW} \end{aligned}$$

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8-87 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the rate of exergy destruction in the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively. The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

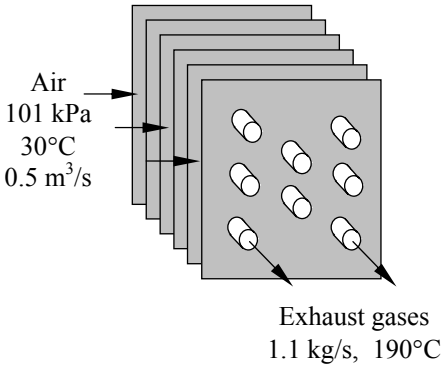
Analysis We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}C_p(T_1 - T_2)$$



Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{gas.}} = (1.1\,\text{kg/s})(1.1\,\text{kJ/kg.°C})(240°\text{C} - 190°\text{C}) = \mathbf{60.5\ kW}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(101\,\text{kPa})(0.5\,\text{m}^3/\text{s})}{(0.287\,\text{kPa.m}^3/\text{kg.K}) \times 303\,\text{K}} = 0.5807\,\text{kg/s}$$

Noting that heat loss by exhaust gases is equal to the heat gain by the air, the air exit temperature becomes

$$\dot{Q} = \left[\dot{m}C_p(T_{\text{out}} - T_{\text{in}})\right]_{\text{air}} \rightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}c_p} = 30°\text{C} + \frac{60.5\,\text{kW}}{(0.5807\,\text{kg/s})(1.005\,\text{kJ/kg.°C})} = 133.7°\text{C}$$

The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\searrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{exhaust}} s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_{\text{exhaust}} s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{exhaust}}(s_2 - s_1) + \dot{m}_{\text{air}}(s_4 - s_3)$$

Noting that the pressure of each fluid remains constant in the heat exchanger, the rate of entropy generation is

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{exhaust}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{air}} c_p \ln\frac{T_4}{T_3}$$

$$= (1.1\,\text{kg/s})(1.1\,\text{kJ/kg.K})\ln\frac{190 + 273}{240 + 273} + (0.5807\,\text{kg/s})(1.005\,\text{kJ/kg.K})\ln\frac{133.7 + 273}{30 + 273}$$

$$= 0.04765\ \text{kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (303\,\text{K})(0.04765\,\text{kW/K}) = \mathbf{14.4\,kW}$$

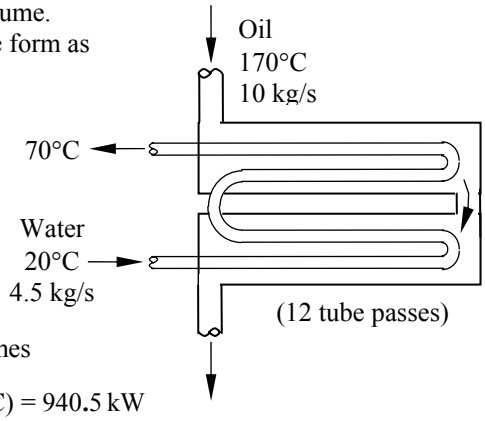
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8-88 Water is heated by hot oil in a heat exchanger. The outlet temperature of the oil and the rate of exergy destruction within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = 940.5 \text{ kW}$$

Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} \rightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{oil}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{oil}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{oil}}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{oil}} c_p \ln\frac{T_4}{T_3}$$

$$= (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.K}) \ln\frac{70 + 273}{20 + 273} + (10 \text{ kg/s})(2.3 \text{ kJ/kg.K}) \ln\frac{129.1 + 273}{170 + 273} = 0.736 \text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298 \text{ K})(0.736 \text{ kW/K}) = \mathbf{219 \text{ kW}}$$

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8-89E Steam is condensed by cooling water in a condenser. The rate of heat transfer and the rate of exergy destruction within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** The temperature of the environment is 77°F.

Properties The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy and entropy of vaporization of water at 120°F are 1025.2 Btu/lbm and s_{fg} = 1.7686 Btu/lbm.R (Table A-4E).

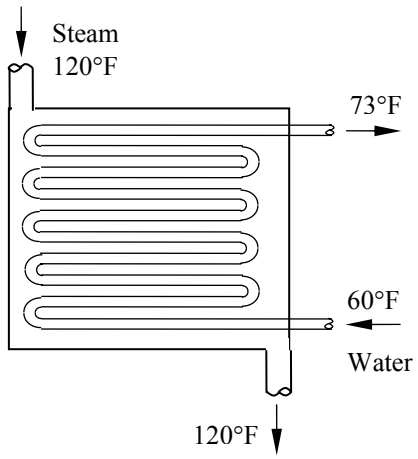
Analysis We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}}$$
$$= (115.3 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73°\text{F} - 60°\text{F}) = \mathbf{1499 \text{ Btu/s}}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1499 \text{ Btu/s}}{1025.2 \text{ Btu/lbm}} = 1.462 \text{ lbm/s}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\searrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{steam}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{steam}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{steam}}(s_f - s_g) = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} - \dot{m}_{\text{steam}} s_{fg}$$

$$= (115.3 \text{ lbm/s})(1.0 \text{ Btu/lbm.R})\ln\frac{73 + 460}{60 + 460} - (1.462 \text{ lbm/s})(1.7686 \text{ Btu/lbm.R}) = 0.2613 \text{ Btu/s.R}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (537 \text{ R})(0.2613 \text{ Btu/s.R}) = \mathbf{140.3 \text{ Btu/s}}$$

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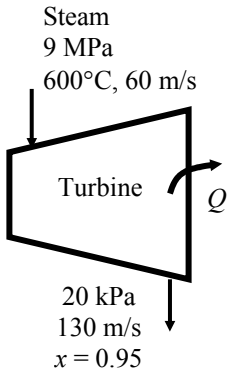
8-90 Steam expands in a turbine, which is not insulated. The reversible power, the exergy destroyed, the second-law efficiency, and the possible increase in the turbine power if the turbine is well insulated are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Potential energy change is negligible.

Analysis (*a*) The properties of the steam at the inlet and exit of the turbine are (Tables A-4 through A-6)

$$\left.\begin{aligned} P_1 &= 9\ \text{MPa} \\ T_1 &= 600°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 3634.1\ \text{kJ/kg} \\ s_1 &= 6.9605\ \text{kJ/kg.K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 20\ \text{kPa} \\ x_2 &= 0.95 \end{aligned}\right\} \begin{aligned} h_2 &= 2491.1\ \text{kJ/kg} \\ s_2 &= 7.5535\ \text{kJ/kg.K} \end{aligned}$$



The enthalpy at the dead state is

$$\left.\begin{aligned} T_0 &= 25°\text{C} \\ x &= 0 \end{aligned}\right\} h_0 = 104.83\ \text{kJ/kg}$$

The mass flow rate of steam may be determined from an energy balance on the turbine

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{\text{out}} + \dot{W}_a$$

$$\dot{m}\left[3634.1\ \text{kJ/kg} + \frac{(60\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right] = \dot{m}\left[2491.1\ \text{kJ/kg} + \frac{(130\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

$$+ 220\ \text{kW} + 4500\ \text{kW} \longrightarrow \dot{m} = 4.137\ \text{kg/s}$$

The reversible power may be determined from

$$\dot{W}_{\text{rev}} = \dot{m}\left[h_1 - h_2 - T_0(s_1 - s_2) + \frac{V_1^2 - V_2^2}{2}\right]$$

$$= (2.693)\left[(3634.1 - 2491.1) - (298)(6.9605 - 7.5535) + \frac{(60\ \text{m/s})^2 - (130\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{5451\ kW}$$

(b) The exergy destroyed in the turbine is

$$\dot{X}_{\text{dest}} = \dot{W}_{\text{rev}} - \dot{W}_{\text{a}} = 5451 - 4500 = \mathbf{951\ kW}$$

(*c*) The second-law efficiency is

$$\eta_{II} = \frac{\dot{W}_{\text{a}}}{\dot{W}_{\text{rev}}} = \frac{4500\ \text{kW}}{5451\ \text{kW}} = \mathbf{0.826}$$

(*d*) The energy of the steam at the turbine inlet in the given dead state is

$$\dot{Q} = \dot{m}(h_1 - h_0) = (4.137\ \text{kg/s})(3634.1 - 104.83)\text{kJ/kg} = 14{,}602\ \text{kW}$$

The fraction of energy at the turbine inlet that is converted to power is

$$f = \frac{\dot{W}_{\text{a}}}{\dot{Q}} = \frac{4500\ \text{kW}}{14{,}602\ \text{kW}} = 0.3082$$

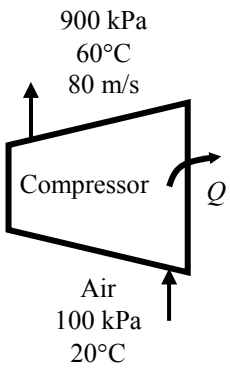
Assuming that the same fraction of heat loss from the turbine could have been converted to work, the possible increase in the power if the turbine is to be well-insulated becomes

$$\dot{W}_{\text{increase}} = f\dot{Q}_{\text{out}} = (0.3082)(220\ \text{kW}) = \mathbf{67.8\ kW}$$

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8-91 Air is compressed in a compressor that is intentionally cooled. The actual and reversible power inputs, the second law efficiency, and the mass flow rate of cooling water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Potential energy change is negligible. **3** Air is an ideal gas with constant specific heats.



Properties The gas constant of air is $R = 0.287$ kJ/kg.K and the specific heat of air at room is $c_p = 1.005$ kJ/kg.K. the specific heat of water at room temperature is $c_w = 4.18$ kJ/kg.K (Tables A-2, A-3).

Analysis (*a*) The mass flow rate of air is

$$\dot{m} = \rho \dot{V}_1 = \frac{P_1}{RT_1}\dot{V}_1 = \frac{(100\,\text{kPa})}{(0.287\,\text{kJ/kg.K})(20+273\,\text{K})}(4.5\,\text{m}^3/\text{s}) = 5.351\,\text{kg/s}$$

The power input for a reversible-isothermal process is given by

$$\dot{W}_{\text{rev}} = \dot{m}RT_1 \ln\frac{P_2}{P_1} = (5.351\,\text{kg/s})(0.287\,\text{kJ/kg.K})(20+273\,\text{K})\ln\left(\frac{900\,\text{kPa}}{100\,\text{kPa}}\right) = \mathbf{988.8\,kW}$$

Given the isothermal efficiency, the actual power may be determined from

$$\dot{W}_{\text{actual}} = \frac{\dot{W}_{\text{rev}}}{\eta_T} = \frac{988.8\,\text{kW}}{0.70} = \mathbf{1413\,kW}$$

(*b*) The given isothermal efficiency is actually the second-law efficiency of the compressor

$$\eta_{\text{II}} = \eta_T = \mathbf{0.70}$$

(*c*) An energy balance on the compressor gives

$$\begin{aligned}\dot{Q}_{\text{out}} &= \dot{m}\left[c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right] + \dot{W}_{\text{actual,in}} \\ &= (5.351\,\text{kg/s})\left[(1.005\,\text{kJ/kg.°C})(20-60)°\text{C} + \frac{0-(80\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right)\right] + 1413\,\text{kW} \\ &= 1181\,\text{kW}\end{aligned}$$

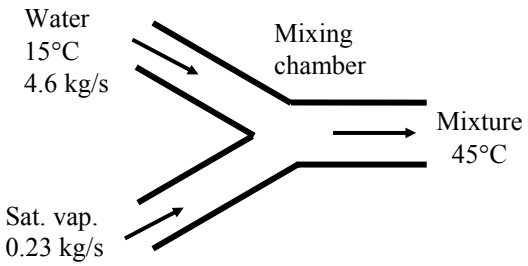
The mass flow rate of the cooling water is

$$\dot{m}_w = \frac{\dot{Q}_{\text{out}}}{c_w \Delta T} = \frac{1181\,\text{kW}}{(4.18\,\text{kJ/kg.°C})(10°\text{C})} = \mathbf{28.25\,kg/s}$$

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8-92 Water is heated in a chamber by mixing it with saturated steam. The temperature of the steam entering the chamber, the exergy destruction, and the second-law efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the chamber is negligible.



Analysis (*a*) The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 15°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = h_0 = 62.98\ \text{kJ/kg} \\ s_1 = s_0 = 0.22447\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_3 = 45°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_3 = 188.44\ \text{kJ/kg} \\ s_3 = 0.63862\ \text{kJ/kg.K} \end{array}$$

An energy balance on the chamber gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$(4.6\ \text{kg/s})(62.98\ \text{kJ/kg}) + (0.23\ \text{kg/s}) h_2 = (4.6 + 0.23\ \text{kg/s})(188.44\ \text{kJ/kg})$$

$$h_2 = 2697.5\ \text{kJ/kg}$$

The remaining properties of the saturated steam are

$$\left.\begin{array}{l} h_2 = 2697.5\ \text{kJ/kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{114.3°C} \\ s_2 = 7.1907\ \text{kJ/kg.K} \end{array}$$

(*b*) The specific exergy of each stream is

$$\psi_1 = 0$$

$$\psi_2 = h_2 - h_0 - T_0(s_2 - s_0)$$
$$= (2697.5 - 62.98)\text{kJ/kg} - (15 + 273\ \text{K})(7.1907 - 0.22447)\text{kJ/kg.K} = 628.28\ \text{kJ/kg}$$

$$\psi_3 = h_3 - h_0 - T_0(s_3 - s_0)$$
$$= (188.44 - 62.98)\text{kJ/kg} - (15 + 273\ \text{K})(0.63862 - 0.22447)\text{kJ/kg.K} = 6.18\ \text{kJ/kg}$$

The exergy destruction is determined from an exergy balance on the chamber to be

$$\dot{X}_{\text{dest}} = \dot{m}_1 \psi_1 + \dot{m}_2 \psi_2 - (\dot{m}_1 + \dot{m}_2) \psi_3$$
$$= 0 + (0.23\ \text{kg/s})(628.28\ \text{kJ/kg}) - (4.6 + 0.23\ \text{kg/s})(6.18\ \text{kJ/kg})$$
$$= \mathbf{114.7\ kW}$$

(*c*) The second-law efficiency for this mixing process may be determined from

$$\eta_{\text{II}} = \frac{(\dot{m}_1 + \dot{m}_2)\psi_3}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} = \frac{(4.6 + 0.23\ \text{kg/s})(6.18\ \text{kJ/kg})}{0 + (0.23\ \text{kg/s})(628.28\ \text{kJ/kg})} = \mathbf{0.207}$$

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8-93 An expression is to be derived for the work potential of the single-phase contents of a rigid adiabatic container when the initially empty container is filled through a single opening from a source of working fluid whose properties remain fixed.

Analysis The conservation of mass principle for this system reduces to

$$\frac{dm_{\mathrm{CV}}}{dt} = \dot{m}_{\mathrm{i}}$$

where the subscript *i* stands for the inlet state. When the entropy generation is set to zero (for calculating work potential) and the combined first and second law is reduced to fit this system, it becomes

$$\dot{W}_{\mathrm{rev}} = -\frac{d(U - T_0 S)}{dt} + (h - T_0 S)_i \dot{m}_i$$

When these are combined, the result is

$$\dot{W}_{\mathrm{rev}} = -\frac{d(U - T_0 S)}{dt} + (h - T_0 S)_i \frac{dm_{\mathrm{CV}}}{dt}$$

Recognizing that there is no initial mass in the system, integration of the above equation produces

$$W_{\mathrm{rev}} = (h - T_0 s)_i m_2 - m_2 (h_2 - T_0 s_2)$$

$$\frac{W_{\mathrm{rev}}}{m_2} = (h_i - h_2) - T_0 (s_i - s_2)$$

where the subscript 2 stands for the final state in the container.

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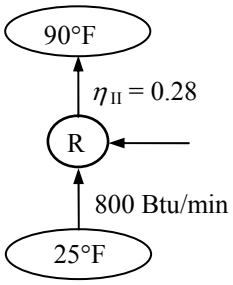
Review Problems

8-94E The 2nd-law efficiency of a refrigerator and the refrigeration rate are given. The power input to the refrigerator is to be determined.

Analysis From the definition of the second law efficiency, the COP of the refrigerator is determined to be

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H / T_L - 1} = \frac{1}{550/485 - 1} = 7.462$$

$$\eta_{II} = \frac{\text{COP}_\text{R}}{\text{COP}_{\text{R,rev}}} \longrightarrow \text{COP}_\text{R} = \eta_{\text{II}}\text{COP}_{\text{R,rev}} = 0.28 \times 7.462 = 2.089$$



Thus the power input is

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{800 \text{ Btu/min}}{2.089}\left(\frac{1 \text{ hp}}{42.41 \text{ Btu/min}}\right) = \mathbf{9.03 \text{ hp}}$$

8-95 Refrigerant-134a is expanded adiabatically in an expansion valve. The work potential of R-134a at the inlet, the exergy destruction, and the second-law efficiency are to be determined.

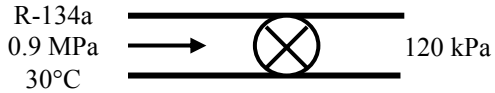
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The properties of the refrigerant at the inlet and exit of the valve and at dead state are (Tables A-11 through A-13)

$$\left.\begin{aligned} P_1 &= 0.9 \text{ MPa} \\ T_1 &= 30°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 93.57 \text{ kJ/kg} \\ s_1 &= 0.34751 \text{ kJ/kg.K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 120 \text{ kPa} \\ h_2 &= h_1 = 93.57 \text{ kJ/kg} \end{aligned}\right\} s_2 = 0.37614 \text{ kJ/kg.K}$$

$$\left.\begin{aligned} P_0 &= 100 \text{ kPa} \\ T_0 &= 20°\text{C} \end{aligned}\right\} \begin{aligned} h_0 &= 272.17 \text{ kJ/kg} \\ s_0 &= 1.0918 \text{ kJ/kg.K} \end{aligned}$$



The specific exergy of the refrigerant at the inlet and exit of the valve are

$$\begin{aligned} \psi_1 &= h_1 - h_0 - T_0(s_1 - s_0) \\ &= (93.57 - 272.17)\text{kJ/kg} - (20 + 273.15 \text{ K})(0.34751 - 1.0918)\text{kJ/kg} \cdot \text{K} \\ &= \mathbf{39.59 \text{ kJ/kg}} \end{aligned}$$

$$\begin{aligned} \psi_2 &= h_2 - h_0 - T_0(s_2 - s_0) \\ &= (93.57 - 272.17)\text{kJ/kg} - (20 + 273.15 \text{ K})(0.37614 - 1.0918 \text{ kJ/kg.K} \\ &= 31.20 \text{ kJ/kg} \end{aligned}$$

(*b*) The exergy destruction is determined to be

$$\begin{aligned} x_{\text{dest}} &= T_0(s_2 - s_1) \\ &= (20 + 273.15 \text{ K})(037614 - 0.34751)\text{kJ/kg} \cdot \text{K} \\ &= \mathbf{8.39 \text{ kJ/kg}} \end{aligned}$$

(*c*) The second-law efficiency for this process may be determined from

$$\eta_{\text{II}} = \frac{\psi_2}{\psi_1} = \frac{31.20 \text{ kJ/kg}}{39.59 \text{ kJ/kg}} = 0.788 = \mathbf{78.8\%}$$

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8-96 Steam is accelerated in an adiabatic nozzle. The exit velocity, the rate of exergy destruction, and the second-law efficiency are to be determined.

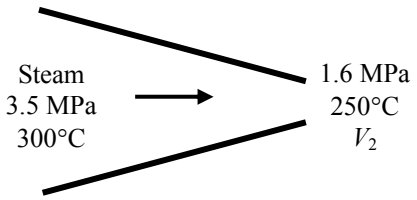
Assumptions **1** Steady operating conditions exist. **2** Potential energy changes are negligible.

Analysis (*a*) The properties of the steam at the inlet and exit of the turbine and at the dead state are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 3.5\ \text{MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2978.4\ \text{kJ/kg} \\ s_1 = 6.4484\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.6\ \text{kPa} \\ T_2 = 250°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 2919.9\ \text{kJ/kg} \\ s_2 = 6.6753\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_0 = 18°\text{C} \\ x = 0 \end{array}\right\} \begin{array}{l} h_0 = 75.54\ \text{kJ/kg} \\ s_0 = 0.2678\ \text{kJ/kg.K} \end{array}$$



The exit velocity is determined from an energy balance on the nozzle

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$2978.4\ \text{kJ/kg} + \frac{(0\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 2919.9\ \text{kJ/kg} + \frac{\text{V}_2^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$V_2 = \mathbf{342.0\ m/s}$$

(*b*) The rate of exergy destruction is the exergy decrease of the steam in the nozzle

$$\dot{X}_{\text{dest}} = \dot{m}\left[h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} - T_0(s_2 - s_1\right]$$

$$= (0.4\ \text{kg/s})\left[\begin{array}{l}(2919.9 - 2978.4)\text{kJ/kg} + \dfrac{(342\ \text{m/s})^2 - 0}{2}\left(\dfrac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) \\ -(291\ \text{K})(6.6753 - 6.4484)\text{kJ/kg.K}\end{array}\right]$$

$$= \mathbf{26.41\ kW}$$

(*c*) The exergy of the refrigerant at the inlet is

$$\dot{X}_1 = \dot{m}\left[h_1 - h_0 + \frac{V_1^2}{2} - T_0(s_1 - s_0\right]$$

$$= (0.4\ \text{kg/s})\left[(2978.4 - 75.54)\ \text{kJ/kg} + 0 - (291\ \text{K})(6.4484 - 0.2678)\text{kJ/kg.K}\right]$$

$$= 441.72\ \text{kW}$$

The second-law efficiency for this device may be defined as the exergy output divided by the exergy input:

$$\eta_{\text{II}} = \frac{\dot{X}_2}{\dot{X}_1} = 1 - \frac{\dot{X}_{\text{dest}}}{\dot{X}_1} = 1 - \frac{26.41\ \text{kW}}{441.72\ \text{kW}} = \mathbf{0.940}$$

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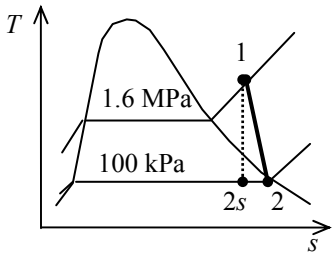
8-97 R-134a is expanded in an adiabatic process with an isentropic efficiency of 0.85. The second law efficiency is to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** The device is adiabatic and thus heat transfer is negligible.

Analysis We take the R-134a as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-W_{\text{out}} = \Delta U = m(u_2 - u_1)$$



From the R-134a tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 1600\ \text{kPa} \\ T_1 = 80°\text{C} \end{array}\right\} \begin{array}{l} \nu_1 = 0.014362\ \text{m}^3/\text{kg} \\ u_1 = 282.09\ \text{kJ/kg} \\ s_1 = 0.9875\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ s_{2s} = s_1 \end{array}\right\} u_{2s} = 223.16\ \text{kJ/kg}$$

The actual work input is

$$w_{a,\text{out}} = \eta_T w_{s,\text{out}} = \eta_T (u_1 - u_{2s}) = (0.85)(282.09 - 223.16)\text{kJ/kg} = 50.09\ \text{kJ/kg}$$

The actual internal energy at the end of the expansion process is

$$w_{a,\text{out}} = (u_1 - u_2) \longrightarrow u_2 = u_1 - w_{a,\text{out}} = 282.09 - 50.09 = 232.00\ \text{kJ/kg}$$

Other actual properties at the final state are (Table A-13)

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ u_2 = 232.00\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} \nu_2 = 0.2139\ \text{m}^3/\text{lbm} \\ s_2 = 1.0251\ \text{kJ/kg}\cdot\text{K} \end{array}$$

The useful work is determined from

$$\begin{aligned} w_u &= w_{a,\text{out}} - w_{\text{surr}} = w_{a,\text{out}} - P_0(\nu_2 - \nu_1) \\ &= 50.09\ \text{kJ/kg} - (100\ \text{kPa})(0.2139 - 0.014362)\ \text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 30.14\ \text{kJ/kg} \end{aligned}$$

The exergy change between initial and final states is

$$\begin{aligned} \phi_1 - \phi_2 &= u_1 - u_2 + P_0(\nu_1 - \nu_2) - T_0(s_1 - s_2) \\ &= (282.09 - 232.00)\text{kJ/kg} + (100\ \text{kPa})(0.014362 - 0.2139)\ \text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &\quad - (298\ \text{K})(0.9875 - 1.0251)\text{kJ/kg}\cdot\text{K} \\ &= 41.34\ \text{kJ/kg} \end{aligned}$$

The second law efficiency is then

$$\eta_{\text{II}} = \frac{w_u}{\Delta\phi} = \frac{30.14\ \text{kJ/kg}}{41.34\ \text{kJ/kg}} = \mathbf{0.729}$$

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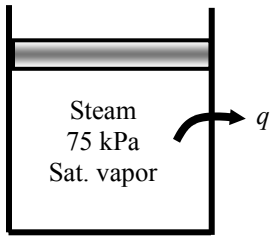
8-98 Steam is condensed in a closed system at a constant pressure from a saturated vapor to a saturated liquid by rejecting heat to a thermal energy reservoir. The second law efficiency is to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible.

Analysis We take the steam as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

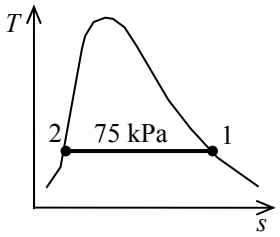
$$W_{b,\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$



From the steam tables (Table A-5),

$$\left.\begin{array}{l} P_1 = 75\ \text{kPa} \\ \text{Sat. vapor} \end{array}\right\} \begin{array}{l} \nu_1 = \nu_g = 2.2172\ \text{m}^3/\text{kg} \\ u_1 = u_g = 2496.1\ \text{kJ/kg} \\ s_1 = s_g = 7.4558\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 75\ \text{kPa} \\ \text{Sat. liquid} \end{array}\right\} \begin{array}{l} \nu_2 = \nu_f = 0.001037\ \text{m}^3/\text{kg} \\ u_2 = u_f = 384.36\ \text{kJ/kg} \\ s_2 = s_f = 1.2132\ \text{kJ/kg}\cdot\text{K} \end{array}$$



The boundary work during this process is

$$w_{b,\text{in}} = P(\nu_1 - \nu_2) = (75\ \text{kPa})(2.2172 - 0.001037)\ \text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 166.2\ \text{kJ/kg}$$

The heat transfer is determined from the energy balance:

$$q_{\text{out}} = w_{b,\text{in}} - (u_2 - u_1) = 166.2\ \text{kJ/kg} - (384.36 - 2496.1)\text{kJ/kg} = 2278\ \text{kJ/kg}$$

The exergy change between initial and final states is

$$\begin{aligned} \phi_1 - \phi_2 &= u_1 - u_2 + P_0(\nu_1 - \nu_2) - T_0(s_1 - s_2) - q_{\text{out}}\left(1 - \frac{T_0}{T_R}\right) \\ &= (2496.1 - 384.36)\text{kJ/kg} + (100\ \text{kPa})(2.2172 - 0.001037)\ \text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &\quad - (298\ \text{K})(7.4558 - 1.2132)\text{kJ/kg}\cdot\text{K} - (2278\ \text{kJ/kg})\left(1 - \frac{298\ \text{K}}{310\ \text{K}}\right) \\ &= 384.9\ \text{kJ/kg} \end{aligned}$$

The second law efficiency is then

$$\eta_{\text{II}} = \frac{w_{b,\text{in}}}{\Delta\phi} = \frac{166.2\ \text{kJ/kg}}{384.9\ \text{kJ/kg}} = 0.432 = \mathbf{43.2\%}$$

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8-99 R-134a is vaporized in a closed system at a constant pressure from a saturated liquid to a saturated vapor by transferring heat from a reservoir at two pressures. The pressure that is more effective from a second-law point of view is to be determined.

***Assumptions* 1** Kinetic and potential energy changes are negligible.

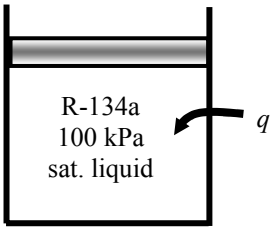
Analysis We take the R-134a as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1)$$

$$Q_{\text{in}} = W_{b,\text{out}} + \Delta U$$

$$Q_{\text{in}} = \Delta H = m(h_2 - h_1)$$



At 100 kPa:

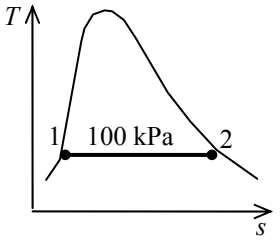
From the R-134a tables (Table A-12),

$$u_{fg@100\,\text{kPa}} = 197.98\,\text{kJ/kg}$$

$$h_{fg@100\,\text{kPa}} = 217.16\,\text{kJ/kg}$$

$$s_{fg@100\,\text{kPa}} = 0.87995\,\text{kJ/kg}\cdot\text{K}$$

$$v_{fg@100\,\text{kPa}} = v_g - v_f = 0.19254 - 0.0007259 = 0.19181\,\text{m}^3/\text{kg}$$



The boundary work during this process is

$$w_{b,\text{out}} = P(v_2 - v_1) = Pv_{fg} = (100\,\text{kPa})(0.19181)\,\text{m}^3/\text{kg}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) = 19.18\,\text{kJ/kg}$$

The useful work is determined from

$$w_u = w_{b,\text{out}} - w_{\text{surr}} = P(v_2 - v_1) - P_0(v_2 - v_1) = 0\,\text{kJ/kg}$$

since $P = P_0 = 100$ kPa. The heat transfer from the energy balance is

$$q_{\text{in}} = h_{fg} = 217.16\,\text{kJ/kg}$$

The exergy change between initial and final states is

$$\phi_1 - \phi_2 = u_1 - u_2 + P_0(v_1 - v_2) - T_0(s_1 - s_2) + q_{\text{in}}\left(1 - \frac{T_0}{T_R}\right)$$

$$= -u_{fg} - P_0 v_{fg} + T_0 s_{fg} + q_{\text{in}}\left(1 - \frac{T_0}{T_R}\right)$$

$$= -197.98\,\text{kJ/kg} - (100\,\text{kPa})(0.19181\,\text{m}^3/\text{kg})\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) + (298\,\text{K})(0.87995\,\text{kJ/kg}\cdot\text{K})$$

$$+ (217.16\,\text{kJ/kg})\left(1 - \frac{298\,\text{K}}{273\,\text{K}}\right)$$

$$= 25.18\,\text{kJ/kg}$$

The second law efficiency is then

$$\eta_{\text{II}} = \frac{w_u}{\Delta\phi} = \frac{0\,\text{kJ/kg}}{25.18\,\text{kJ/kg}} = \mathbf{0}$$

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At 200 kPa:

$$u_{fg@200\,\text{kPa}} = 186.21\,\text{kJ/kg}$$
$$h_{fg@200\,\text{kPa}} = 206.03\,\text{kJ/kg}$$
$$s_{fg@200\,\text{kPa}} = 0.78316\,\text{kJ/kg}\cdot\text{K}$$
$$v_{fg@200\,\text{kPa}} = v_g - v_f = 0.099867 - 0.0007533 = 0.099114\,\text{m}^3/\text{kg}$$

$$w_{b,\text{out}} = P(v_2 - v_1) = Pv_{fg} = (200\,\text{kPa})(0.099114)\,\text{m}^3/\text{kg}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) = 19.82\,\text{kJ/kg}$$

$$w_u = w_{b,\text{out}} - w_{\text{surr}} = P(v_2 - v_1) - P_0(v_2 - v_1)$$
$$= (P - P_0)v_{fg} = (200-100)\,\text{kPa}(0.099114)\,\text{m}^3/\text{kg}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) = 9.911\,\text{kJ/kg}$$

$$q_{\text{in}} = h_{fg} = 206.03\,\text{kJ/kg}$$

$$\phi_1 - \phi_2 = u_1 - u_2 + P_0(v_1 - v_2) - T_0(s_1 - s_2) + q_{\text{in}}\left(1 - \frac{T_0}{T_R}\right)$$
$$= -u_{fg} - P_0 v_{fg} + T_0 s_{fg} + q_{\text{in}}\left(1 - \frac{T_0}{T_R}\right)$$
$$= -186.21\,\text{kJ/kg} - (100\,\text{kPa})(0.099114\,\text{m}^3/\text{kg})\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) + (298\,\text{K})(0.78316\,\text{kJ/kg}\cdot\text{K})$$
$$+ (206.03\,\text{kJ/kg})\left(1 - \frac{298\,\text{K}}{273\,\text{K}}\right)$$
$$= 18.39\,\text{kJ/kg}$$

$$\eta_{\text{II}} = \frac{w_u}{\Delta\phi} = \frac{9.911\,\text{kJ/kg}}{18.39\,\text{kJ/kg}} = \mathbf{0.539}$$

The process at 200 kPa is more effective from a work production standpoint.

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8-100 An electrical radiator is placed in a room and it is turned on for a period of time. The time period for which the heater was on, the exergy destruction, and the second-law efficiency are to be determined.

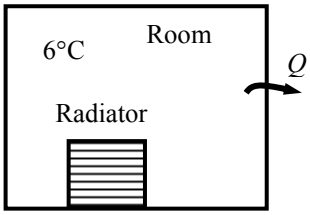
Assumptions **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats. **3** The room is well-sealed. **4** Standard atmospheric pressure of 101.3 kPa is assumed.

Properties The properties of air at room temperature are $R = 0.287$ kPa.m^3/kg.K, $c_p = 1.005$ kJ/kg.K, $c_v = 0.718$ kJ/kg.K (Table A-2). The properties of oil are given to be $\rho = 950$ kg/m^3, $c_{oil} = 2.2$ kJ/kg.K.

Analysis (*a*) The masses of air and oil are

$$m_a = \frac{P_1 V}{RT_1} = \frac{(101.3\,\text{kPa})(75\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(6+273\,\text{K})} = 94.88\,\text{kg}$$

$$m_{oil} = \rho_{oil} V_{oil} = (950\,\text{kg/m}^3)(0.050\,\text{m}^3) = 47.50\,\text{kg}$$



An energy balance on the system can be used to determine time period for which the heater was kept on

$$(\dot{W}_{in} - \dot{Q}_{out})\Delta t = [mc_v(T_2 - T_1)]_a + [mc(T_2 - T_1)]_{oil}$$

$$(2.4 - 0.75\,\text{kW})\Delta t = [(94.88\,\text{kg})(0.718\,\text{kJ/kg.°C})(20-6)°\text{C}] + [(47.50\,\text{kg})(2.2\,\text{kJ/kg.°C})(60-6)°\text{C}]$$

$$\Delta t = 3988\,\text{s} = \mathbf{66.6\ min}$$

(*b*) The pressure of the air at the final state is

$$P_{a2} = \frac{m_a R T_{a2}}{V} = \frac{(94.88\,\text{kg})(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\,\text{K})}{75\,\text{m}^3} = 106.4\,\text{kPa}$$

The amount of heat transfer to the surroundings is

$$Q_{out} = \dot{Q}_{out}\Delta t = (0.75\,\text{kJ/s})(3988\,\text{s}) = 2999\,\text{kJ}$$

The entropy generation is the sum of the entropy changes of air, oil, and the surroundings

$$\Delta S_a = m\left[c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right]$$

$$= (94.88\,\text{kg})\left[(1.005\,\text{kJ/kg.K})\ln\frac{(20+273)\,\text{K}}{(6+273)\,\text{K}} - (0.287\,\text{kJ/kg.K})\ln\frac{106.4\,\text{kPa}}{101.3\,\text{kPa}}\right]$$

$$= 3.335\,\text{kJ/K}$$

$$\Delta S_{oil} = mc\ln\frac{T_2}{T_1} = (47.50\,\text{kg})(2.2\,\text{kJ/kg.K})\ln\frac{(60+273)\,\text{K}}{(6+273)\,\text{K}} = 18.49\,\text{kJ/K}$$

$$\Delta S_{surr} = \frac{Q_{out}}{T_{surr}} = \frac{2999\,\text{kJ}}{(6+273)\,\text{K}} = 10.75\,\text{kJ/K}$$

$$S_{gen} = \Delta S_a + \Delta S_{oil} + \Delta S_{surr} = 3.335 + 18.49 + 10.75 = 32.57\,\text{kJ/K}$$

The exergy destruction is determined from

$$X_{dest} = T_0 S_{gen} = (6+273\,\text{K})(32.57\,\text{kJ/K}) = 9088\,\text{kJ} = \mathbf{9.09\ MJ}$$

(*c*) The second-law efficiency may be defined in this case as the ratio of the exergy recovered to the exergy input. That is,

$$X_{a,2} = m[c_v(T_2 - T_1)] - T_0\Delta S_a$$

$$= (94.88\,\text{kg})[(0.718\,\text{kJ/kg.°C})(20-6)°\text{C}] - (6+273\,\text{K})(3.335\,\text{kJ/K}) = 23.16\,\text{kJ}$$

$$X_{oil,2} = m[C(T_2 - T_1)] - T_0\Delta S_a$$

$$= (47.50\,\text{kg})[(2.2\,\text{kJ/kg.°C})(60-6)°\text{C}] - (6+273\,\text{K})(18.49\,\text{kJ/K}) = 484.5\,\text{kJ}$$

$$\eta_{II} = \frac{X_{recovered}}{X_{supplied}} = \frac{X_{a,2} + X_{oil,2}}{\dot{W}_{in}\Delta t} = \frac{(23.16+484.5)\,\text{kJ}}{(2.4\,\text{kJ/s})(3998\,\text{s})} = 0.0529 = \mathbf{5.3\%}$$

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8-101 Hot exhaust gases leaving an internal combustion engine is to be used to obtain saturated steam in an adiabatic heat exchanger. The rate at which the steam is obtained, the rate of exergy destruction, and the second-law efficiency are to be determined.

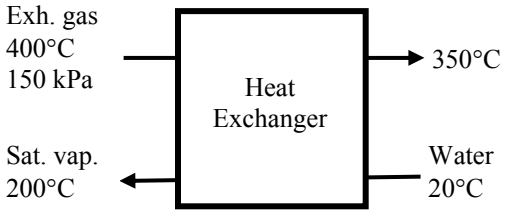
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air properties are used for exhaust gases. **4** Pressure drops in the heat exchanger are negligible.

Properties The gas constant of air is $R = 0.287$ kJkg.K. The specific heat of air at the average temperature of exhaust gases (650 K) is $c_p = 1.063$ kJ/kg.K (Table A-2).

Analysis (*a*) We denote the inlet and exit states of exhaust gases by (1) and (2) and that of the water by (3) and (4). The properties of water are (Table A-4)

$$\left.\begin{matrix} T_3 = 20°\text{C} \\ x_3 = 0 \end{matrix}\right\} \begin{matrix} h_3 = 83.91\ \text{kJ/kg} \\ s_3 = 0.29649\ \text{kJ/kg.K} \end{matrix}$$

$$\left.\begin{matrix} T_4 = 200°\text{C} \\ x_4 = 1 \end{matrix}\right\} \begin{matrix} h_4 = 2792.0\ \text{kJ/kg} \\ s_4 = 6.4302\ \text{kJ/kg.K} \end{matrix}$$



An energy balance on the heat exchanger gives

$$\dot{m}_a h_1 + \dot{m}_w h_3 = \dot{m}_a h_2 + \dot{m}_w h_4$$

$$\dot{m}_a c_p (T_1 - T_2) = \dot{m}_w (h_4 - h_3)$$

$$(0.8\ \text{kg/s})(1.063\ \text{kJ/kg°C})(400 - 350)°\text{C} = \dot{m}_w (2792.0 - 83.91)\text{kJ/kg}$$

$$\dot{m}_w = \mathbf{0.01570\ kg/s}$$

(*b*) The specific exergy changes of each stream as it flows in the heat exchanger is

$$\Delta s_a = c_p \ln \frac{T_2}{T_1} = (0.8\ \text{kg/s})(1.063\ \text{kJ/kg.K}) \ln \frac{(350 + 273)\ \text{K}}{(400 + 273)\ \text{K}} = -0.08206\ \text{kJ/kg.K}$$

$$\begin{aligned} \Delta\psi_a &= c_p (T_2 - T_1) - T_0 \Delta s_a \\ &= (1.063\ \text{kJ/kg.°C})(350 - 400)°\text{C} - (20 + 273\ \text{K})(-0.08206\ \text{kJ/kg.K}) \\ &= -29.106\ \text{kJ/kg} \end{aligned}$$

$$\begin{aligned} \Delta\psi_w &= h_4 - h_3 - T_0 (s_4 - s_3) \\ &= (2792.0 - 83.91)\text{kJ/kg} - (20 + 273\ \text{K})(6.4302 - 0.29649)\text{kJ/kg.K} \\ &= 910.913\ \text{kJ/kg} \end{aligned}$$

The exergy destruction is determined from an exergy balance on the heat exchanger to be

$$-\dot{X}_{\text{dest}} = \dot{m}_a \Delta\psi_a + \dot{m}_w \Delta\psi_w = (0.8\ \text{kg/s})(-29.106\ \text{kJ/kg}) + (0.01570\ \text{kg/s})(910.913)\ \text{kJ/kg} = -8.98\ \text{kW}$$

or

$$\dot{X}_{\text{dest}} = \mathbf{8.98\ kW}$$

(*c*) The second-law efficiency for a heat exchanger may be defined as the exergy increase of the cold fluid divided by the exergy decrease of the hot fluid. That is,

$$\eta_{\text{II}} = \frac{\dot{m}_w \Delta\psi_w}{-\dot{m}_a \Delta\psi_a} = \frac{(0.01570\ \text{kg/s})(910.913\ \text{kJ/kg})}{-(0.8\ \text{kg/s})(-29.106\ \text{kJ/kg})} = \mathbf{0.614}$$

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8-102 The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat loss and the amount of exergy destruction in 5 h are to be determined

Assumptions Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values.

Analysis We take the glass to be the system, which is a closed system. The amount of heat loss is determined from

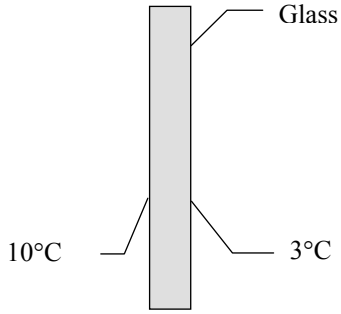
$$Q = \dot{Q}\Delta t = (4.4 \text{ kJ/s})(5\times 3600 \text{ s}) = \mathbf{79{,}200 \text{ kJ}}$$

Under steady conditions, the rate form of the entropy balance for the glass simplifies to

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,glass}} = 0$$

$$\frac{4400 \text{ W}}{283 \text{ K}} - \frac{4400 \text{ W}}{276 \text{ K}} + \dot{S}_{\text{gen,wall}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,glass}} = 0.3943 \text{ W/K}$$



Then the amount of entropy generation over a period of 5 h becomes

$$S_{\text{gen,glass}} = \dot{S}_{\text{gen,glass}}\Delta t = (0.3943 \text{ W/K})(5\times 3600 \text{ s}) = 7098 \text{ J/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (278 \text{ K})(7.098 \text{ kJ/K}) = \mathbf{1973 \text{ kJ}}$$

Discussion The total entropy generated during this process can be determined by applying the entropy balance on an *extended system* that includes the glass and its immediate surroundings on both sides so that the boundary temperature of the extended system is the room temperature on one side and the environment temperature on the other side at all times. Using this value of entropy generation will give the total exergy destroyed during the process, including the temperature gradient zones on both sides of the window.

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8-103 Heat is transferred steadily to boiling water in the pan through its bottom. The inner and outer surface temperatures of the bottom of the pan are given. The rate of exergy destruction within the bottom plate is to be determined.

Assumptions Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values.

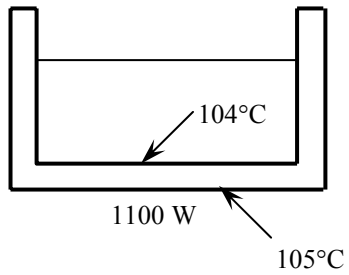
Analysis We take the bottom of the pan to be the system, which is a closed system. Under steady conditions, the rate form of the entropy balance for this system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,system}} = 0$$

$$\frac{1100\ \text{W}}{378\ \text{K}} - \frac{1100\ \text{W}}{377\ \text{K}} + \dot{S}_{\text{gen,system}} = 0$$

$$\dot{S}_{\text{gen,system}} = 0.007719\ \text{W/K}$$



The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298\ \text{K})(0.007719\ \text{W/K}) = \mathbf{2.30\ W}$$

8-104 Elevation, base area, and the depth of a crater lake are given. The maximum amount of electricity that can be generated by a hydroelectric power plant is to be determined.

Assumptions The evaporation of water from the lake is negligible.

Analysis The exergy or work potential of the water is the potential energy it possesses relative to the ground level,

$$\text{Exergy} = \text{PE} = mgh$$

Therefore,

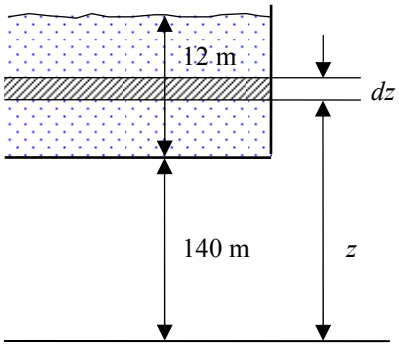
$$\text{Exergy} = PE = \int dPE = \int gz\,dm = \int gz(\rho A\,dz)$$

$$= \rho Ag \int_{z_1}^{z_2} z\,dz = \rho Ag(z_2^2 - z_1^2)/2$$

$$= 0.5(1000\ \text{kg/m}^3)(2\times 10^4\ \text{m}^2)(9.81\ \text{m/s}^2)$$

$$\times \left((152\ \text{m})^2 - (140\ \text{m}^2)\right)\left(\frac{1\ \text{h}}{3600\ \text{s}}\right)\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$= \mathbf{9.55\times 10^4\ kWh}$$



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8-105 An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature, the minimum work input, and the exergy destroyed during this process are to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible. **4** The environment temperature is given to be $T_0 = 20°C$.

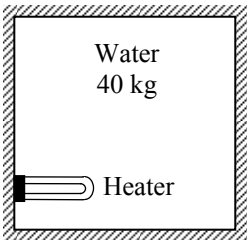
Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis Taking the water in the container as the system, which is a closed system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = (\Delta U)_{\text{water}}$$

$$\dot{W}_{\text{e,in}}\Delta t = mc(T_2 - T_1)_{\text{water}}$$



Substituting,

$$(800 \text{ J/s})\Delta t = (40 \text{ kg})(4180 \text{ J/kg}\cdot°\text{C})(80 - 20)°\text{C}$$

Solving for Δt gives

$$\Delta t = \mathbf{12{,}540 \text{ s} = 209 \text{ min} = 3.48 \text{ h}}$$

Again we take the water in the tank to be the system. Noting that no heat or mass crosses the boundaries of this system and the energy and entropy contents of the heater are negligible, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{water}}$$

Therefore, the entropy generated during this process is

$$S_{\text{gen}} = \Delta S_{\text{water}} = mc\ln\frac{T_2}{T_1} = (40 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\frac{353 \text{ K}}{293 \text{ K}} = 31.15 \text{ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293 \text{ K})(31.15 \text{ kJ/K}) = \mathbf{9127 \text{ kJ}}$$

The actual work input for this process is

$$W_{\text{act,in}} = \dot{W}_{\text{act,in}}\Delta t = (0.8 \text{ kJ/s})(12{,}540 \text{ s}) = 10{,}032 \text{ kJ}$$

Then the reversible (or minimum required)work input becomes

$$W_{\text{rev,in}} = W_{\text{act,in}} - X_{\text{destroyed}} = 10{,}032 - 9127 = \mathbf{906 \text{ kJ}}$$

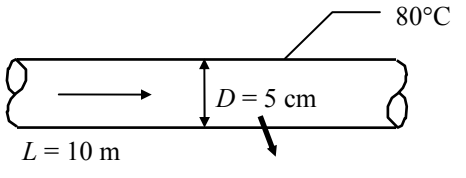
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8-106 A hot water pipe at a specified temperature is losing heat to the surrounding air at a specified rate. The rate at which the work potential is wasted during this process is to be determined.

Assumptions Steady operating conditions exist.

Analysis We take the air in the vicinity of the pipe (excluding the pipe) as our system, which is a closed system.. The system extends from the outer surface of the pipe to a distance at which the temperature drops to the surroundings temperature. In steady operation, the rate form of the entropy balance for this system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}^{\nearrow 0} = 0$$



$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,system}} = 0$$

$$\frac{1175\ \text{W}}{353\ \text{K}} - \frac{1175\ \text{W}}{278\ \text{K}} + \dot{S}_{\text{gen,system}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,system}} = 0.8980\ \text{W/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (278\ \text{K})(0.8980\ \text{W/K}) = \mathbf{250\ W}$$

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8-107 Air expands in an adiabatic turbine from a specified state to another specified state. The second-law efficiency is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The device is adiabatic and thus heat transfer is negligible. **3** Air is an ideal gas with constant specific heats. **4** Kinetic and potential energy changes are negligible.

Properties At the average temperature of (425 + 325)/2 = 375 K, the constant pressure specific heat of air is c_p = 1.011 kJ/kg.K (Table A-2b). The gas constant of air is R = 0.287 kJ/kg.K (Table A-1).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

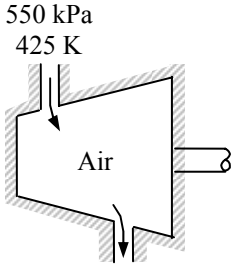
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$

$$w_{\text{out}} = c_p(T_1 - T_2)$$



Substituting,

$$w_{\text{out}} = c_p(T_1 - T_2) = (1.011\,\text{kJ/kg}\cdot\text{K})(425 - 325)\text{K} = 101.1\,\text{kJ/kg}$$

The entropy change of air is

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.011\,\text{kJ/kg}\cdot\text{K})\ln\frac{325\,\text{K}}{425\,\text{K}} - (0.287\,\text{kJ/kg}\cdot\text{K})\ln\frac{110\,\text{kPa}}{550\,\text{kPa}} \\ &= 0.1907\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

The maximum (reversible) work is the exergy difference between the inlet and exit states

$$\begin{aligned} w_{\text{rev,out}} &= c_p(T_1 - T_2) - T_0(s_1 - s_2) \\ &= w_{\text{out}} - T_0(s_1 - s_2) \\ &= 101.1\,\text{kJ/kg} - (298\,\text{K})(-0.1907\,\text{kJ/kg}\cdot\text{K}) \\ &= \mathbf{157.9\ kJ/kg} \end{aligned}$$

The second law efficiency is then

$$\eta_{\text{II}} = \frac{w_{\text{out}}}{w_{\text{rev,out}}} = \frac{101.1\,\text{kJ/kg}}{157.9\,\text{kJ/kg}} = \mathbf{0.640}$$

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8-108 Steam is accelerated in a nozzle. The actual and maximum outlet velocities are to be determined.

Assumptions **1** The nozzle operates steadily. **2** The changes in potential energies are negligible.

Properties The properties of steam at the inlet and the exit of the nozzle are (Tables A-4 through A-6)

$$\left.\begin{aligned} P_1 &= 300\ \text{kPa} \\ T_1 &= 150°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 2761.2\ \text{kJ/kg} \\ s_1 &= 7.0792\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 150\ \text{kPa} \\ x_2 &= 1\ \text{(sat. vapor)} \end{aligned}\right\} \begin{aligned} h_2 &= 2693.1\ \text{kJ/kg} \\ s_2 &= 7.2231\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

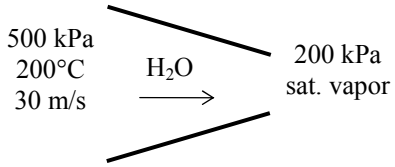
Analysis We take the nozzle to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}^{\not{0}\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$\frac{V_2^2 - V_1^2}{2} = h_1 - h_2 = \Delta \text{ke}_{\text{actual}}$$



Substituting,

$$\Delta \text{ke}_{\text{actual}} = h_1 - h_2 = 2761.2 - 2693.1 = 68.1\ \text{kJ/kg}$$

The actual velocity at the exit is then

$$\frac{V_2^2 - V_1^2}{2} = \Delta \text{ke}_{\text{actual}}$$

$$V_2 = \sqrt{V_1^2 + 2\Delta \text{ke}_{\text{actual}}} = \sqrt{(45\ \text{m/s})^2 + 2(68.1\ \text{kJ/kg})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{371.8\ m/s}$$

The maximum kinetic energy change is determined from

$$\Delta \text{ke}_{\text{max}} = h_1 - h_2 - T_0(s_1 - s_2) = 68.1 - (298)(7.0792 - 7.2231) = 111.0\ \text{kJ/kg}$$

The maximum velocity at the exit is then

$$\frac{V_{2,\text{max}}^2 - V_1^2}{2} = \Delta \text{ke}_{\text{max}}$$

$$V_{2,\text{max}} = \sqrt{V_1^2 + 2\Delta \text{ke}_{\text{max}}} = \sqrt{(45\ \text{m/s})^2 + 2(111.0\ \text{kJ/kg})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)}$$

$$= \mathbf{473.3\ m/s}$$

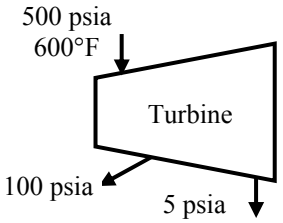
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8-109E Steam is expanded in a two-stage turbine. Six percent of the inlet steam is bled for feedwater heating. The isentropic efficiencies for the two stages of the turbine are given. The second-law efficiency of the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The turbine is well-insulated, and there is no heat transfer from the turbine.

Analysis There is one inlet and two exits. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

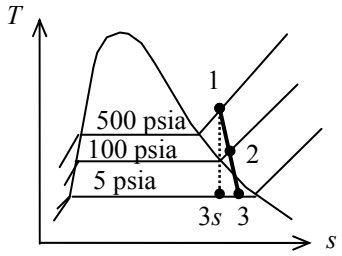
$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$

$$w_{\text{out}} = h_1 - 0.06 h_2 - 0.94 h_3$$

$$w_{\text{out}} = (h_1 - h_2) + 0.94(h_2 - h_3)$$

The isentropic and actual enthalpies at three states are determined using steam tables as follows:



$$\left.\begin{array}{l} P_1 = 500 \text{ psia} \\ T_1 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_1 = 1298.6 \text{ Btu/lbm} \\ s_1 = 1.5590 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ s_{2s} = s_1 = 1.5590 \text{ Btu/lbm} \cdot \text{R} \end{array}\right\} \begin{array}{l} x_{2s} = 0.9609 \\ h_{2s} = 1152.7 \text{ Btu/lbm} \end{array}$$

$$\eta_{T,1} = \frac{h_1 - h_2}{h_1 - h_{2s}} \longrightarrow h_2 = h_1 - \eta_{T,1}(h_1 - h_{2s}) = 1298.6 - (0.97)(1298.6 - 1152.7) = 1157.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ h_2 = 1157.1 \text{ Btu/lbm} \end{array}\right\} \begin{array}{l} x_2 = 0.9658 \\ s_2 = 1.5646 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_3 = 5 \text{ psia} \\ s_3 = s_2 = 1.5646 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} \begin{array}{l} x_{3s} = 0.8265 \\ h_{3s} = 957.09 \text{ Btu/lbm} \end{array}$$

$$\eta_{T,2} = \frac{h_2 - h_3}{h_2 - h_{3s}} \longrightarrow h_3 = h_2 - \eta_{T,2}(h_2 - h_{3s}) = 1157.1 - (0.95)(1157.1 - 957.09) = 967.09 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 5 \text{ psia} \\ h_3 = 967.09 \text{ Btu/lbm} \end{array}\right\} \begin{array}{l} x_3 = 0.8364 \\ s_3 = 1.5807 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

Substituting into the energy balance per unit mass flow at the inlet of the turbine, we obtain

$$\begin{aligned} w_{\text{out}} &= (h_1 - h_2) + 0.94(h_2 - h_3) \\ &= (1298.6 - 1157.1) + 0.94(1157.1 - 967.09) = 320.1 \text{ Btu/lbm} \end{aligned}$$

The reversible work output per unit mass flow at the turbine inlet is

$$\begin{aligned} w_{\text{rev}} &= h_1 - h_2 - T_0(s_1 - s_2) + 0.94\left[h_2 - h_3 - T_0(s_2 - s_3)\right] \\ &= 1298.6 - 1157.1 - (537)(1.5590 - 1.5646) + 0.94\left[(1157.1 - 967.09 - (537)(1.5646 - 1.5807)\right] \\ &= 331.2 \text{ Btu/lbm} \end{aligned}$$

The second law efficiency is then

$$\eta_{\text{II}} = \frac{w_{\text{out}}}{w_{\text{rev}}} = \frac{320.1 \text{ Btu/lbm}}{331.2 \text{ Btu/lbm}} = \mathbf{0.966}$$

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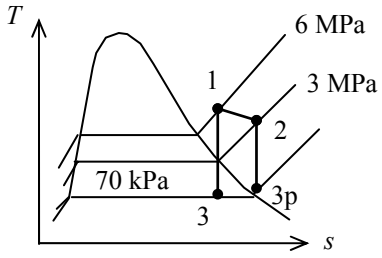
8-110 A throttle valve is placed in the steam line supplying the turbine inlet in order to control an isentropic steam turbine. The second-law efficiency of this system when the valve is partially open to when it is fully open is to be compared.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The turbine is well-insulated, and there is no heat transfer from the turbine.

Analysis

Valve is fully open:

The properties of steam at various states are



$$\left.\begin{array}{l} P_0 = 100\ \text{kPa} \\ T_1 = 25°\text{C} \end{array}\right\} \begin{array}{l} h_0 \cong h_{f\,@25°\text{C}} = 104.8\ \text{kJ/kg} \\ s_0 \cong s_{f\,@25°\text{C}} = 0.3672\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_1 = P_2 = 6\ \text{MPa} \\ T_1 = T_2 = 700°\text{C} \end{array}\right\} \begin{array}{l} h_1 = h_2 = 3894.3\ \text{kJ/kg} \\ s_1 = s_2 = 7.4247\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 70\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} x_3 = 0.9914 \\ h_3 = 2639.7\ \text{kJ/kg} \end{array}$$

The stream exergy at the turbine inlet is

$$\psi_1 = h_1 - h_0 - T_0(s_1 - s_0) = 3894.3 - 104.8 - (298)(7.4247 - 0.3672) = \mathbf{1686\ kJ/kg}$$

The second law efficiency of the entire system is then

$$\eta_{\text{II}} = \frac{w_{\text{out}}}{w_{\text{rev}}} = \frac{h_1 - h_3}{h_1 - h_3 - T_0(s_1 - s_3)} = \frac{h_1 - h_3}{h_1 - h_3} = \mathbf{1.0}$$

since $s_1 = s_3$ for this system.

Valve is partly open:

$$\left.\begin{array}{l} P_2 = 3\ \text{MPa} \\ h_2 = h_1 = 3894.3\ \text{kJ/kg} \end{array}\right\} \quad s_2 = 7.7405\ \text{kJ/kg}\cdot\text{K} \qquad \text{(from EES)}$$

$$\left.\begin{array}{l} P_3 = 70\ \text{kPa} \\ s_3 = s_2 \end{array}\right\} \quad h_3 = 2760.8\ \text{kJ/kg} \quad \text{(from EES)}$$

$$\psi_2 = h_2 - h_0 - T_0(s_2 - s_0) = 3894.3 - 104.8 - (298)(7.7405 - 0.3672) = \mathbf{1592\ kJ/kg}$$

$$\eta_{\text{II}} = \frac{w_{\text{out}}}{w_{\text{rev}}} = \frac{h_2 - h_3}{h_2 - h_3 - T_0(s_2 - s_3)} = \frac{3894.3 - 2760.8}{3894.3 - 2760.8 - (298)(7.7405 - 7.7405)} = \mathbf{1.0}$$

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8-111 Two rigid tanks that contain water at different states are connected by a valve. The valve is opened and steam flows from tank A to tank B until the pressure in tank A drops to a specified value. Tank B loses heat to the surroundings. The final temperature in each tank and the work potential wasted during this process are to be determined.

Assumptions **1** Tank A is insulated and thus heat transfer is negligible. **2** The water that remains in tank A undergoes a reversible adiabatic process. **3** The thermal energy stored in the tanks themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible. **5** There are no work interactions.

Analysis (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

Tank A :

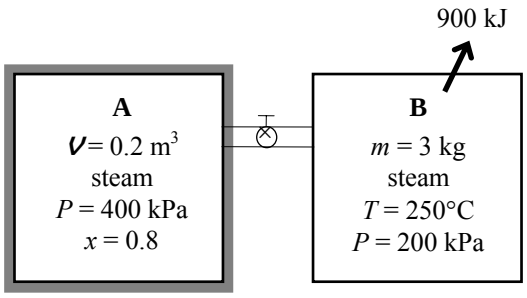
$$\left.\begin{array}{l} P_1 = 400 \text{ kPa} \\ x_1 = 0.8 \end{array}\right\} \begin{array}{l} v_{1,A} = v_f + x_1 v_{fg} = 0.001084 + (0.8)(0.46242 - 0.001084) = 0.37015 \text{ m}^3/\text{kg} \\ u_{1,A} = u_f + x_1 u_{fg} = 604.22 + (0.8)(1948.9) = 2163.3 \text{ kJ/kg} \\ s_{1,A} = s_f + x_1 s_{fg} = 1.7765 + (0.8)(5.1191) = 5.8717 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$T_{2,A} = T_{sat@300kPa} = \mathbf{133.52°C}$$

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ s_2 = s_1 \\ (\text{sat. mixture}) \end{array}\right\} \begin{array}{l} x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{5.8717 - 1.6717}{5.3200} = 0.7895 \\ v_{2,A} = v_f + x_{2,A} v_{fg} = 0.001073 + (0.7895)(0.60582 - 0.001073) = 0.47850 \text{ m}^3/\text{kg} \\ u_{2,A} = u_f + x_{2,A} u_{fg} = 561.11 + (0.7895)(1982.1 \text{ kJ/kg}) = 2125.9 \text{ kJ/kg} \end{array}$$

Tank B :

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 250°\text{C} \end{array}\right\} \begin{array}{l} v_{1,B} = 1.1989 \text{ m}^3/\text{kg} \\ u_{1,B} = 2731.4 \text{ kJ/kg} \\ s_{1,B} = 7.7100 \text{ kJ/kg} \cdot \text{K} \end{array}$$



The initial and the final masses in tank A are

$$m_{1,A} = \frac{V_A}{v_{1,A}} = \frac{0.2 \text{ m}^3}{0.37015 \text{ m}^3/\text{kg}} = 0.5403 \text{ kg}$$

and

$$m_{2,A} = \frac{V_A}{v_{2,A}} = \frac{0.2 \text{m}^3}{0.479 \text{m}^3/\text{kg}} = 0.4180 \text{ kg}$$

Thus, 0.540 - 0.418 = 0.122 kg of mass flows into tank B. Then,

$$m_{2,B} = m_{1,B} - 0.122 = 3 + 0.122 = 3.122 \text{ kg}$$

The final specific volume of steam in tank B is determined from

$$v_{2,B} = \frac{V_B}{m_{2,B}} = \frac{(m_1 v_1)_B}{m_{2,B}} = \frac{(3 \text{ kg})(1.1989 \text{ m}^3/\text{kg})}{3.122 \text{ m}^3} = 1.152 \text{ m}^3/\text{kg}$$

We take the entire contents of both tanks as the system, which is a closed system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = (\Delta U)_A + (\Delta U)_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$-Q_{\text{out}} = (m_2 u_2 - m_1 u_1)_A + (m_2 u_2 - m_1 u_1)_B$$

Substituting,

$$-900 = \{(0.418)(2125.9) - (0.5403)(2163.3)\} + \{(3.122)u_{2,B} - (3)(2731.4)\}$$

$$u_{2,B} = 2425.9 \text{ kJ/kg}$$

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Thus,

$$\left.\begin{array}{l} v_{2,B} = 1.152 \text{ m}^3/\text{kg} \\ u_{2,B} = 2425.9 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_{2,B} = \mathbf{110.1°C} \\ s_{2,B} = 6.9772 \text{ kJ/kg} \cdot \text{K} \end{array}$$

(*b*) The total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes both tanks and their immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,surr}}} + S_{\text{gen}} = \Delta S_{\text{A}} + \Delta S_{\text{B}}$$

Rearranging and substituting, the total entropy generated during this process is determined to be

$$\begin{aligned} S_{\text{gen}} &= \Delta S_A + \Delta S_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}} = \left(m_2 s_2 - m_1 s_1\right)_A + \left(m_2 s_2 - m_1 s_1\right)_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}} \\ &= \{(0.418)(5.8717) - (0.5403)(5.8717)\} + \{(3.122)(6.9772) - (3)(7.7100)\} + \frac{900 \text{ kJ}}{273 \text{ K}} \\ &= 1.234 \text{ kJ/K} \end{aligned}$$

The work potential wasted is equivalent to the exergy destroyed during a process, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (273 \text{ K})(1.234 \text{ kJ/K}) = \mathbf{337 \text{ kJ}}$$

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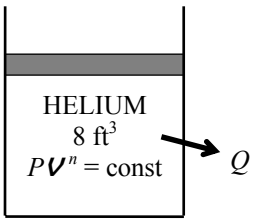
8-112E A cylinder initially filled with helium gas at a specified state is compressed polytropically to a specified temperature and pressure. The actual work consumed and the minimum useful work input needed are to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium. **5** The environment temperature is 70°F.

Properties The gas constant of helium is $R = 2.6805$ psia.ft^3/lbm.R = 0.4961 Btu/lbm.R (Table A-1E). The specific heats of helium are $c_v = 0.753$ and $c_p = 1.25$ Btu/lbm.R (Table A-2E).

Analysis (a) Helium at specified conditions can be treated as an ideal gas. The mass of helium is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(40 \text{ psia})(8 \text{ ft}^3)}{(2.6805 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(530 \text{ R})} = 0.2252 \text{ lbm}$$



The exponent n and the boundary work for this polytropic process are determined to be

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow V_2 = \frac{T_2}{T_1}\frac{P_1}{P_2} V_1 = \frac{(780 \text{ R})(40 \text{ psia})}{(530 \text{ R})(140 \text{ psia})}(8 \text{ ft}^3) = 3.364 \text{ ft}^3$$

$$P_2 V_2^n = P_1 V_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{V_1}{V_2}\right)^n \longrightarrow \left(\frac{140}{40}\right) = \left(\frac{8}{3.364}\right)^n \longrightarrow n = 1.446$$

Then the boundary work for this polytropic process can be determined from

$$W_{b,in} = -\int_1^2 P dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.2252 \text{ lbm})(0.4961 \text{ Btu/lbm} \cdot \text{R})(780 - 530)\text{R}}{1 - 1.446} = 62.62 \text{ Btu}$$

Also,

$$W_{surr,in} = -P_0(V_2 - V_1) = -(14.7 \text{ psia})(3.364 - 8)\text{ft}^3\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia} \cdot \text{ft}^3}\right) = 12.61 \text{ Btu}$$

Thus,

$$W_{u,in} = W_{b,in} - W_{surr,in} = 62.62 - 12.61 = \mathbf{50.0 \text{ Btu}}$$

(*b*) We take the helium in the cylinder as the system, which is a closed system. Taking the direction of heat transfer to be from the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{out} + W_{b,in} = \Delta U = m(u_2 - u_1)$$

$$-Q_{out} = m(u_2 - u_1) - W_{b,in}$$

$$Q_{out} = W_{b,in} - mc_v(T_2 - T_1)$$

Substituting,

$$Q_{out} = 62.62 \text{ Btu} - (0.2252 \text{ lbm})(0.753 \text{ Btu/lbm} \cdot \text{R})(780 - 530)\text{R} = 20.69 \text{ Btu}$$

The total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the cylinder and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. It gives

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{out}}{T_{b,surr}} + S_{gen} = \Delta S_{sys}$$

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where the entropy change of helium is

$$\Delta S_{\text{sys}} = \Delta S_{\text{helium}} = m\left(c_{p,\text{avg}} \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right)$$
$$= (0.2252 \text{ lbm})\left[(1.25 \text{ Btu/lbm} \cdot \text{R}) \ln \frac{780 \text{ R}}{530 \text{ R}} - (0.4961 \text{ Btu/lbm} \cdot \text{R}) \ln \frac{140 \text{ psia}}{40 \text{ psia}} \right]$$
$$= -0.03201 \text{ Btu/R}$$

Rearranging and substituting, the total entropy generated during this process is determined to be

$$S_{\text{gen}} = \Delta S_{\text{helium}} + \frac{Q_{\text{out}}}{T_0} = (-0.03201 \text{ Btu/R}) + \frac{20.69 \text{ Btu}}{530 \text{ R}} = 0.007022 \text{ Btu/R}$$

The work potential wasted is equivalent to the exergy destroyed during a process, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (530 \text{ R})(0.007022 \text{ Btu/R}) = 3.722 \text{ Btu}$$

The minimum work with which this process could be accomplished is the reversible work input, $W_{\text{rev, in}}$ which can be determined directly from

$$W_{\text{rev,in}} = W_{\text{u,in}} - X_{\text{destroyed}} = 50.0 - 3.722 = \mathbf{46.3 \text{ Btu}}$$

Discussion The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction term equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}^{\nearrow 0 \text{ (reversible)}}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}} \rightarrow W_{\text{rev,in}} = X_2 - X_1$$

Substituting the closed system exergy relation, the reversible work input during this process is determined to be

$$W_{\text{rev}} = (U_2 - U_1) - T_0 (S_2 - S_1) + P_0 (V_2 - V_1)$$
$$= (0.2252 \text{ lbm})(0.753 \text{ Btu/lbm} \cdot \text{R})(320 - 70)^\circ\text{F} - (530 \text{ R})(-0.03201 \text{ Btu/R})$$
$$+ (14.7 \text{ psia})(3.364 - 8)\text{ft}^3 [\text{Btu}/5.4039 \text{ psia} \cdot \text{ft}^3]$$
$$= \mathbf{46.7 \text{ Btu}}$$

The slight difference is due to round-off error.

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8-113 Steam expands in a two-stage adiabatic turbine from a specified state to specified pressure. Some steam is extracted at the end of the first stage. The wasted power potential is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible. **4** The environment temperature is given to be T_0 = 25°C.

Analysis The wasted power potential is equivalent to the rate of exergy destruction during a process, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$.

The total rate of entropy generation during this process is determined by taking the entire turbine, which is a control volume, as the system and applying the entropy balance. Noting that this is a steady-flow process and there is no heat transfer,

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}^{\nearrow 0}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} = 0$$

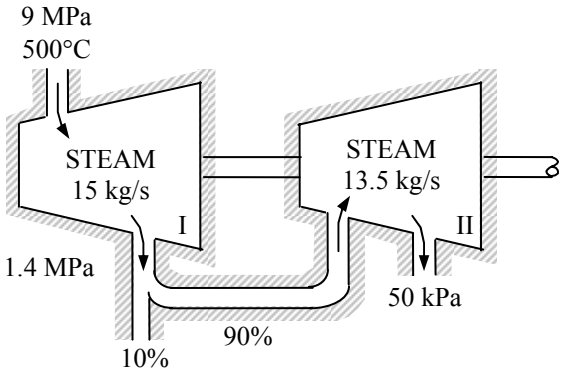
$$\dot{m}_1 s_1 - 0.1\dot{m}_1 s_2 - 0.9\dot{m}_1 s_3 + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}_1[0.9s_3 + 0.1s_2 - s_1]$$

and $$X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0 \dot{m}_1[0.9s_3 + 0.1s_2 - s_1]$$

From the steam tables (Tables A-4 through 6)

$$\left.\begin{matrix} P_1 = 9\text{ MPa} \\ T_1 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3387.4\text{ kJ/kg} \\ s_1 = 6.6603\text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 1.4\text{ MPa} \\ s_{2s} = s_1 \end{matrix}\right\} h_{2s} = 2882.4\text{ kJ/kg}$$



and,

$$\eta_T = \frac{h_1 - h_2}{h_1 - h_{2s}} \longrightarrow h_2 = h_1 - \eta_T(h_1 - h_{2s})$$
$$= 3387.4 - 0.88(3387.4 - 2882.4)$$
$$= 2943.0\text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 1.4\text{ MPa} \\ h_2 = 2943.0\text{ kJ/kg} \end{matrix}\right\} s_2 = 6.7776\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{matrix} P_3 = 50\text{ kPa} \\ s_{3s} = s_1 \end{matrix}\right\} \begin{matrix} x_{3s} = \dfrac{s_{3s} - s_f}{s_{fg}} = \dfrac{6.6603 - 1.0912}{6.5019} = 0.8565 \\ h_{3s} = h_f + x_{3s} h_{fg} = 340.54 + 0.8565 \times 2304.7 = 2314.6\text{ kJ/kg} \end{matrix}$$

and $$\eta_T = \frac{h_1 - h_3}{h_1 - h_{3s}} \longrightarrow h_3 = h_1 - \eta_T(h_1 - h_{3s})$$
$$= 3387.4 - 0.88(3387.4 - 2314.6)$$
$$= 2443.3\text{ kJ/kg}$$

$$\left.\begin{matrix} P_3 = 50\text{ kPa} \\ h_3 = 2443.3\text{ kJ/kg} \end{matrix}\right\} \begin{matrix} x_3 = \dfrac{h_3 - h_f}{h_{fg}} = \dfrac{2443.3 - 340.54}{2304.7} = 0.9124 \\ s_3 = s_f + x_3 s_{fg} = 1.0912 + 0.9124 \times 6.5019 = 7.0235\text{ kJ/kg}\cdot\text{K} \end{matrix}$$

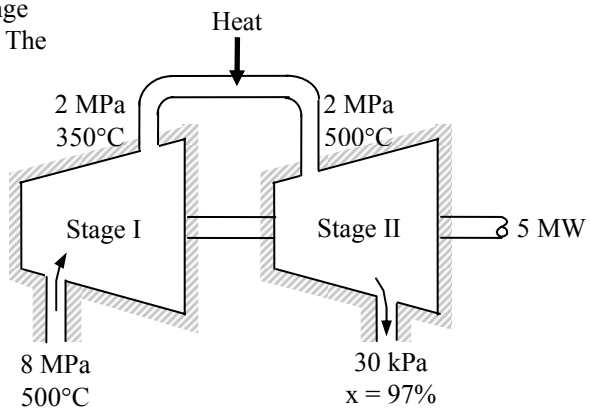
Substituting, the wasted work potential is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298\text{ K})(15\text{ kg/s})(0.9 \times 7.0235 + 0.1 \times 6.7776 - 6.6603)\text{kJ/kg} = \mathbf{1514\ kW}$$

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8-114 Steam expands in a two-stage adiabatic turbine from a specified state to another specified state. Steam is reheated between the stages. For a given power output, the reversible power output and the rate of exergy destruction are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible. **4** The environment temperature is given to be $T_0 = 25°C$.



Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{matrix} P_1 = 8\ \text{MPa} \\ T_1 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 3399.5\ \text{kJ/kg} \\ s_1 = 6.7266\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 2\ \text{MPa} \\ T_2 = 350°\text{C} \end{matrix}\right\} \begin{matrix} h_2 = 3137.7\ \text{kJ/kg} \\ s_2 = 6.9583\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_3 = 2\ \text{MPa} \\ T_3 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3468.3\ \text{kJ/kg} \\ s_3 = 7.4337\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 30\ \text{kPa} \\ x_4 = 0.97 \end{matrix}\right\} \begin{matrix} h_4 = h_f + x_4 h_{fg} = 289.27 + 0.97 \times 2335.3 = 2554.5\ \text{kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 0.9441 + 0.97 \times 6.8234 = 7.5628\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

Analysis We take the entire turbine, excluding the reheat section, as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{m}h_3 = \dot{m}h_2 + \dot{m}h_4 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}[(h_1 - h_2) + (h_3 - h_4)]$$

Substituting, the mass flow rate of the steam is determined from the steady-flow energy equation applied to the actual process,

$$\dot{m} = \frac{\dot{W}_{\text{out}}}{h_1 - h_2 + h_3 - h_4} = \frac{5000\ \text{kJ/s}}{(3399.5 - 3137.7 + 3468.3 - 2554.5)\text{kJ/kg}} = 4.253\ \text{kg/s}$$

The reversible (or maximum) power output is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 + \dot{m}\psi_3 = \dot{m}\psi_2 + \dot{m}\psi_4 + \dot{W}_{\text{rev,out}}$$

$$\dot{W}_{\text{rev,out}} = \dot{m}(\psi_1 - \psi_2) + \dot{m}(\psi_3 - \psi_4)$$

$$= \dot{m}[(h_1 - h_2) + T_0(s_2 - s_1) - \Delta \text{ke}^{\nearrow 0} - \Delta \text{pe}^{\nearrow 0}]$$

$$+ \dot{m}[(h_3 - h_4) + T_0(s_4 - s_3) - \Delta \text{ke}^{\nearrow 0} - \Delta \text{pe}^{\nearrow 0}]$$

Then the reversible power becomes

$$\dot{W}_{\text{rev,out}} = \dot{m}\left[h_1 - h_2 + h_3 - h_4 + T_0(s_2 - s_1 + s_4 - s_3)\right]$$

$$= (4.253\ \text{kg/s})[(3399.5 - 3137.7 + 3468.3 - 2554.5)\text{kJ/kg}$$

$$+ (298\ \text{K})(6.9583 - 6.7266 + 7.5628 - 7.4337)\text{kJ/kg}\cdot\text{K}]$$

$$= \mathbf{5457\ kW}$$

Then the rate of exergy destruction is determined from its definition,

$$\dot{X}_{\text{destroyed}} = \dot{W}_{\text{rev,out}} - \dot{W}_{\text{out}} = 5457 - 5000 = \mathbf{457\ kW}$$

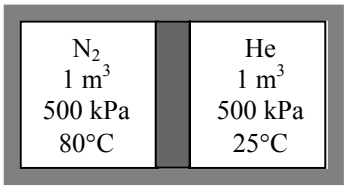
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8-115 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the wasted work potential are to be determined for the cases of piston being fixed and moving freely.

Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

Properties The gas constants and the specific heats are $R = 0.2968$ kPa.m^3/kg.K, $c_p = 1.039$ kJ/kg·°C, and $c_v = 0.743$ kJ/kg·°C for N_2, and $R = 2.0769$ kPa.m^3/kg.K, $c_p = 5.1926$ kJ/kg·°C, and $c_v = 3.1156$ kJ/kg·°C for He (Tables A-1 and A-2).

Analysis The mass of each gas in the cylinder is



$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\ \text{K})} = 4.772\ \text{kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\ \text{K})} = 0.8079\ \text{kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He}$$

Substituting,

$$(4.772\ \text{kg})(0.743\ \text{kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.8079\ \text{kg})(3.1156\ \text{kJ/kg}\cdot^\circ\text{C})(T_f - 25)^\circ\text{C} = 0$$

It gives

$$T_f = \mathbf{57.2°C}$$

where T_f is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

(*b*) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{N_2} + \Delta S_{He}$$

But first we determine the final pressure in the cylinder:

$$N_{total} = N_{N_2} + N_{He} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{He} = \frac{4.772\ \text{kg}}{28\ \text{kg/kmol}} + \frac{0.8079\ \text{kg}}{4\ \text{kg/kmol}} = 0.3724\ \text{kmol}$$

$$P_2 = \frac{N_{total} R_u T}{V_{total}} = \frac{(0.3724\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(330.2\ \text{K})}{2\ \text{m}^3} = 511.1\ \text{kPa}$$

Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$

$$= (4.772\ \text{kg})\left[(1.039\ \text{kJ/kg}\cdot\text{K})\ln\frac{330.2\ \text{K}}{353\ \text{K}} - (0.2968\ \text{kJ/kg}\cdot\text{K})\ln\frac{511.1\ \text{kPa}}{500\ \text{kPa}}\right] = -0.3628\ \text{kJ/K}$$

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$$\Delta S_{\text{He}} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \right)_{\text{He}}$$

$$= (0.8079\ \text{kg})\left[(5.1926\ \text{kJ/kg}\cdot\text{K})\ln\frac{330.2\ \text{K}}{298\ \text{K}} - (2.0769\ \text{kJ/kg}\cdot\text{K})\ln\frac{511.1\ \text{kPa}}{500\ \text{kPa}} \right] = 0.3931\ \text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{N}_2} + \Delta S_{\text{He}} = -0.3628 + 0.3931 = 0.0303\ \text{kJ/K}$$

The wasted work potential is equivalent to the exergy destroyed during a process, and it can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$,

$$X_{\text{destroyed}} = T_0 S_{gen} = (298\ \text{K})(0.0303\ \text{kJ/K}) = \mathbf{9.03\ kJ}$$

If the piston were not free to move, we would still have T_2 = 330.2 K but the volume of each gas would remain constant in this case:

$$\Delta S_{\text{N}_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{v_2}{v_1}^{\nearrow 0} \right)_{\text{N}_2} = (4.772\ \text{kg})(0.743\ \text{kJ/kg}\cdot\text{K})\ln\frac{330.2\ \text{K}}{353\ \text{K}} = -0.2371\ \text{kJ/K}$$

$$\Delta S_{\text{He}} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{v_2}{v_1}^{\nearrow 0} \right)_{\text{He}} = (0.8079\ \text{kg})(3.1156\ \text{kJ/kg}\cdot\text{K})\ln\frac{330.2\ \text{K}}{298\ \text{K}} = 0.258\ \text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{N}_2} + \Delta S_{\text{He}} = -0.2371 + 0.258 = 0.02089\ \text{kJ/K}$$

and

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\ \text{K})(0.02089\ \text{kJ/K}) = \mathbf{6.23\ kJ}$$

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8-116 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the wasted work potential are to be determined for the cases of piston being fixed and moving freely. √

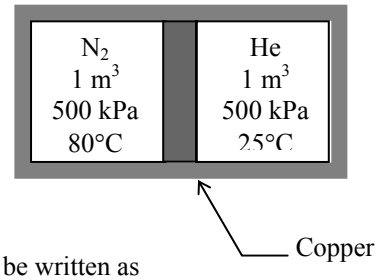
Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

Properties The gas constants and the specific heats are $R = 0.2968$ kPa.m^3/kg.K, $c_p = 1.039$ kJ/kg·°C, and $c_v = 0.743$ kJ/kg·°C for N_2, and $R = 2.0769$ kPa.m^3/kg.K, $c_p = 5.1926$ kJ/kg·°C, and $c_v = 3.1156$ kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of copper piston is $c = 0.386$ kJ/kg·°C (Table A-3).

Analysis The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\ \text{K})} = 4.772\ \text{kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\ \text{kPa})(1\ \text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\ \text{K})} = 0.8079\ \text{kg}$$



Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He} + [mc(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\,Cu} = (80 + 25)/2 = 52.5°\text{C}$$

Substituting,

$$(4.772\ \text{kg})(0.743\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.8079\ \text{kg})(3.1156\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_f - 25)^\circ\text{C}$$
$$+ (5.0\ \text{kg})(0.386\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_f - 52.5)^\circ\text{C} = 0$$

It gives

$$T_f = \mathbf{56.0°C}$$

where T_f is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

(*b*) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston}$$

But first we determine the final pressure in the cylinder:

$$N_{total} = N_{N_2} + N_{He} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{He} = \frac{4.772\ \text{kg}}{28\ \text{kg/kmol}} + \frac{0.8079\ \text{kg}}{4\ \text{kg/kmol}} = 0.3724\ \text{kmol}$$

$$P_2 = \frac{N_{total} R_u T}{V_{total}} = \frac{(0.3724\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(329\ \text{K})}{2\ \text{m}^3} = 509.4\ \text{kPa}$$

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Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$
$$= (4.772\ \text{kg})\left[(1.039\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} - (0.2968\ \text{kJ/kg}\cdot\text{K})\ln\frac{509.4\ \text{kPa}}{500\ \text{kPa}}\right] = -0.3749\ \text{kJ/K}$$

$$\Delta S_{He} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{He}$$
$$= (0.8079\ \text{kg})\left[(5.1926\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} - (2.0769\ \text{kJ/kg}\cdot\text{K})\ln\frac{509.4\ \text{kPa}}{500\ \text{kPa}}\right] = 0.3845\ \text{kJ/K}$$

$$\Delta S_{\text{piston}} = \left(mc\ln\frac{T_2}{T_1}\right)_{\text{piston}} = (5\ \text{kg})(0.386\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{325.5\ \text{K}} = 0.021\ \text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{\text{piston}} = -0.3749 + 0.3845 + 0.021 = 0.03047\ \text{kJ/K}$$

The wasted work potential is equivalent to the exergy destroyed during a process, and it can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\ \text{K})(0.03047\ \text{kJ/K}) = \mathbf{9.08\ kJ}$$

If the piston were not free to move, we would still have T_2 = 330.2 K but the volume of each gas would remain constant in this case:

$$\Delta S_{N_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{\cancel{V_2}^{\,=0}}{V_1}\right)_{N_2} = (4.772\ \text{kg})(0.743\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} = -0.2492\ \text{kJ/K}$$

$$\Delta S_{He} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{\cancel{V_2}^{\,=0}}{V_1}\right)_{He} = (0.8079\ \text{kg})(3.1156\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} = 0.2494\ \text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{\text{piston}} = -0.2492 + 0.2494 + 0.021 = 0.02104\ \text{kJ/K}$$

and

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\ \text{K})(0.02104\ \text{kJ/K}) = \mathbf{6.27\ kJ}$$

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8-117E Argon enters an adiabatic turbine at a specified state with a specified mass flow rate, and leaves at a specified pressure. The isentropic and second-law efficiencies of the turbine are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

Properties The specific heat ratio of argon is $k = 1.667$. The constant pressure specific heat of argon is $c_p = 0.1253$ Btu/lbm.R. The gas constant is $R = 0.04971$ Btu/lbm.R (Table A-2E).

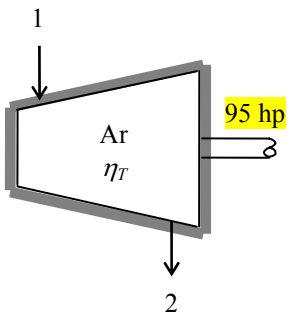
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the isentropic turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{s,\text{out}} + \dot{m}h_{2s} \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{s,\text{out}} = \dot{m}(h_1 - h_{2s})$$



From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (1960\text{ R})\left(\frac{30\text{ psia}}{200\text{ psia}}\right)^{0.667/1.667} = 917.5\text{ R}$$

Then the power output of the isentropic turbine becomes

$$\dot{W}_{s,\text{out}} = \dot{m}c_p(T_1 - T_{2s}) = (40\text{ lbm/min})(0.1253\text{ Btu/lbm}\cdot\text{R})(1960 - 917.5)\text{R}\left(\frac{1\text{ hp}}{42.41\text{ Btu/min}}\right) = 123.2\text{ hp}$$

Then the isentropic efficiency of the turbine is determined from

$$\eta_T = \frac{\dot{W}_{a,\text{out}}}{\dot{W}_{s,\text{out}}} = \frac{95\text{ hp}}{123.2\text{ hp}} = 0.771 = \mathbf{77.1\%}$$

(*b*) Using the steady-flow energy balance relation $\dot{W}_{a,\text{out}} = \dot{m}c_p(T_1 - T_2)$ above, the actual turbine exit temperature is determined to be

$$T_2 = T_1 - \frac{\dot{W}_{a,\text{out}}}{\dot{m}c_p} = 1500 - \frac{95\text{ hp}}{(40\text{ lbm/min})(0.1253\text{ Btu/lbm}\cdot\text{R})}\left(\frac{42.41\text{ Btu/min}}{1\text{ hp}}\right) = 696.1°\text{F} = 1156.1\text{ R}$$

The entropy generation during this process can be determined from an entropy balance on the turbine,

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}^{\not\searrow 0} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

where

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (0.1253\text{ Btu/lbm}\cdot\text{R})\ln\frac{1156.1\text{ R}}{1960\text{ R}} - (0.04971\text{ Btu/lbm}\cdot\text{R})\ln\frac{30\text{ psia}}{200\text{ psia}}$$

$$= 0.02816\text{ Btu/lbm.R}$$

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The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\begin{aligned}\dot{X}_{\text{destroyed}} &= T_0 \dot{S}_{\text{gen}} = \dot{m} T_0 (s_2 - s_1) \\ &= (40\ \text{lbm/min})(537\ \text{R})(0.02816\ \text{Btu/lbm} \cdot \text{R})\left(\frac{1\ \text{hp}}{42.41\ \text{Btu/min}}\right) \\ &= 14.3\ \text{hp}\end{aligned}$$

Then the reversible power and second-law efficiency become

$$\dot{W}_{\text{rev,out}} = \dot{W}_{a,\text{out}} + \dot{X}_{\text{destroyed}} = 95 + 14.3 = 109.3\ \text{hp}$$

and

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{a,out}}}{\dot{W}_{\text{rev,out}}} = \frac{95\ \text{hp}}{109.3\ \text{hp}} = \mathbf{86.9\%}$$

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8-118 The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted steam and the feedwater are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

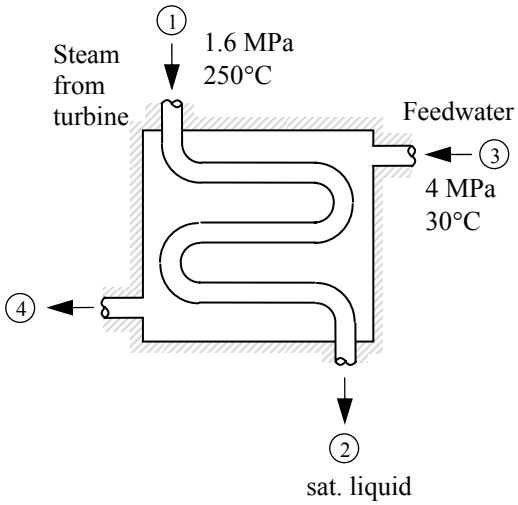
Properties The properties of steam and feedwater are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 1.6\ \text{MPa} \\ T_1 = 250°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2919.9\ \text{kJ/kg} \\ s_1 = 6.6753\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.6\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_2 = h_{f@1.6\ \text{MPa}} = 858.44\ \text{kJ/kg} \\ s_2 = s_{f@1.6\ \text{MPa}} = 2.3435\ \text{kJ/kg}\cdot\text{K} \\ T_2 = 201.4°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 4\ \text{MPa} \\ T_3 = 30°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@30°\text{C}} = 129.37\ \text{kJ/kg} \\ s_3 \cong s_{f@30°\text{C}} = 0.4355\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 4\ \text{MPa} \\ T_4 = T_2 - 10°\text{C} \cong 191.4°\text{C} \end{array}\right\} \begin{array}{l} h_4 \cong h_{f@191.4°\text{C}} = 814.78\ \text{kJ/kg} \\ s_4 \cong s_{f@191.4°\text{C}} = 2.2446\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis (*a*) We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_s \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_{fw}$$

Energy balance (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_s(h_2 - h_1) = \dot{m}_{fw}(h_3 - h_4)$

Dividing by $\dot{m}_{fw}$ and substituting,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_3 - h_4}{h_2 - h_1} = \frac{(129.37 - 814.78)\text{kJ/kg}}{(858.44 - 2919.9)\text{kJ/kg}} = \mathbf{0.3325}$$

(*b*) The entropy generation during this process per unit mass of feedwater can be determined from an entropy balance on the feedwater heater expressed in the rate form as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\swarrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{m}_s(s_1 - s_2) + \dot{m}_{fw}(s_3 - s_4) + \dot{S}_{\text{gen}} = 0$$

$$\frac{\dot{S}_{\text{gen}}}{\dot{m}_{fw}} = \frac{\dot{m}_s}{\dot{m}_{fw}}(s_2 - s_1) + (s_4 - s_3) = (0.3325)(2.3435 - 6.6753) + (2.2446 - 0.4355) = 0.3688\ \text{kJ/K}\cdot\text{kg fw}$$

Noting that this process involves no actual work, the reversible work and exergy destruction become equivalent since

$$X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act,out}} \rightarrow W_{\text{rev,out}} = X_{\text{destroyed}}.$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\ \text{K})(0.3688\ \text{kJ/K}\cdot\text{kgfw}) = \mathbf{109.9\ kJ/kg\ feedwater}$$

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8-119 Problem 8-118 is reconsidered. The effect of the state of the steam at the inlet of the feedwater heater on the ratio of mass flow rates and the reversible power is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
"Input Data"
"Steam (let st=steam data):"
Fluid$='Steam_IAPWS'
T_st[1]=250 [C]
{P_st[1]=1600 [kPa]}
P_st[2] = P_st[1]
x_st[2]=0 "saturated liquid, quality = 0%"
T_st[2]=temperature(steam, P=P_st[2], x=x_st[2])

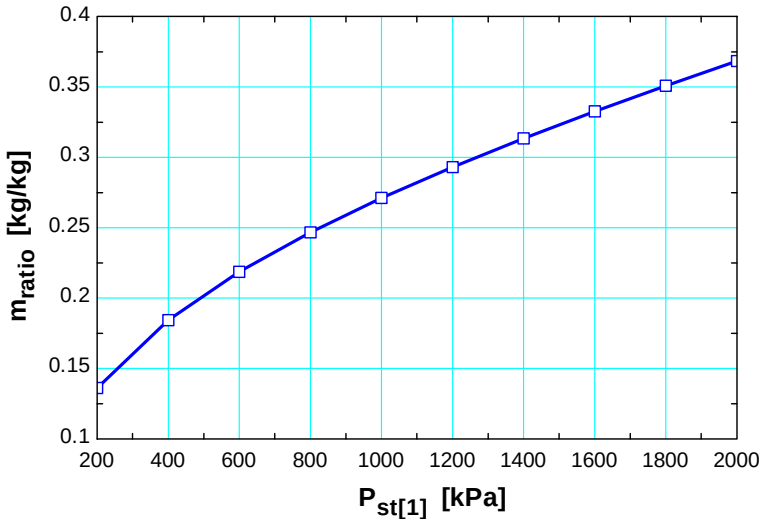
"Feedwater (let fw=feedwater data):"
T_fw[1]=30 [C]
P_fw[1]=4000 [kPa]
P_fw[2]=P_fw[1]  "assume no pressure drop for the feedwater"
T_fw[2]=T_st[2]-10

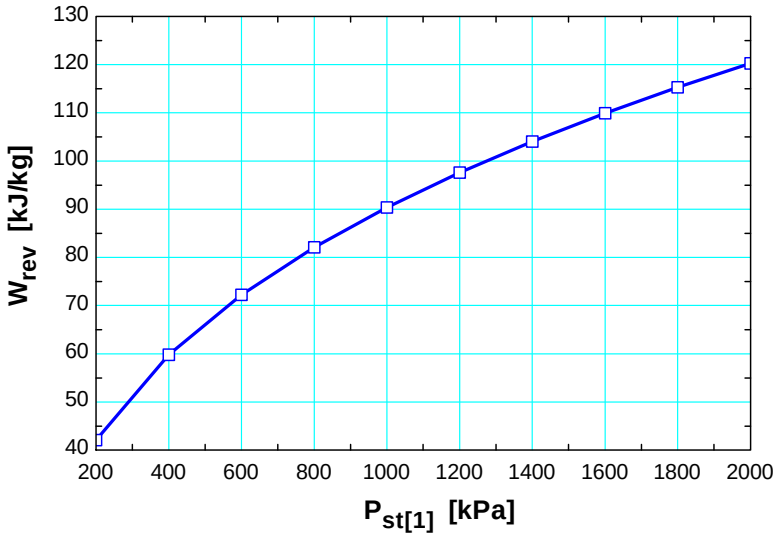
"Surroundings:"
T_o = 25 [C]
P_o = 100 [kPa] "Assumed value for the surrroundings pressure"
"Conservation of mass:"
"There is one entrance, one exit for both the steam and feedwater."
"Steam: m_dot_st[1] = m_dot_st[2]"
"Feedwater: m_dot_fw[1] = m_dot_fw[2]"
"Let m_ratio = m_dot_st/m_dot_fw"
"Conservation of Energy:"
"We write the conservation of energy for steady-flow control volume
having two entrances and two exits with the above assumptions.  Since
neither of the flow rates is know or can be found, write the conservation
of energy per unit mass of  the feedwater."
E_in - E_out =DELTAE_cv
DELTAE_cv=0  "Steady-flow requirement"
 E_in = m_ratio*h_st[1] + h_fw[1]
h_st[1]=enthalpy(Fluid$, T=T_st[1], P=P_st[1])
h_fw[1]=enthalpy(Fluid$,T=T_fw[1], P=P_fw[1])
E_out = m_ratio*h_st[2] + h_fw[2]
h_fw[2]=enthalpy(Fluid$, T=T_fw[2], P=P_fw[2])
h_st[2]=enthalpy(Fluid$, x=x_st[2], P=P_st[2])

"The reversible work is given by Eq. 7-47, where the heat transfer is zero
(the feedwater heater is adiabatic) and the Exergy destroyed is set equal
to zero"
W_rev = m_ratio*(Psi_st[1]-Psi_st[2]) +(Psi_fw[1]-Psi_fw[2])
Psi_st[1]=h_st[1]-h_st_o -(T_o + 273)*(s_st[1]-s_st_o)
s_st[1]=entropy(Fluid$,T=T_st[1], P=P_st[1])
h_st_o=enthalpy(Fluid$, T=T_o, P=P_o)
s_st_o=entropy(Fluid$, T=T_o, P=P_o)
Psi_st[2]=h_st[2]-h_st_o -(T_o + 273)*(s_st[2]-s_st_o)
s_st[2]=entropy(Fluid$,x=x_st[2], P=P_st[2])
Psi_fw[1]=h_fw[1]-h_fw_o -(T_o + 273)*(s_fw[1]-s_fw_o)
h_fw_o=enthalpy(Fluid$, T=T_o, P=P_o)
s_fw[1]=entropy(Fluid$,T=T_fw[1], P=P_fw[1])
s_fw_o=entropy(Fluid$, T=T_o, P=P_o)
Psi_fw[2]=h_fw[2]-h_fw_o -(T_o + 273)*(s_fw[2]-s_fw_o)
s_fw[2]=entropy(Fluid$,T=T_fw[2], P=P_fw[2])
```

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$P_{st,1}$ [kPa]	m_{ratio} [kg/kg]	W_{rev} [kJ/kg]
200	0.1361	42.07
400	0.1843	59.8
600	0.2186	72.21
800	0.2466	82.06
1000	0.271	90.35
1200	0.293	97.58
1400	0.3134	104
1600	0.3325	109.9
1800	0.3508	115.3
2000	0.3683	120.3





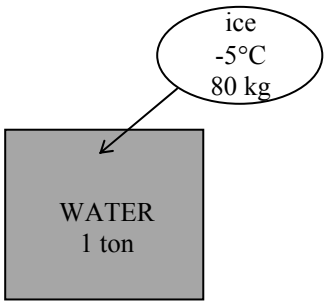
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8-120 A 1-ton (1000 kg) of water is to be cooled in a tank by pouring ice into it. The final equilibrium temperature in the tank and the exergy destruction are to be determined.

Assumptions **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water tank is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C, and the specific heat of ice at about 0°C is $c = 2.11$ kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg..

Analysis (*a*) We take the ice and the water as the system, and disregard any heat transfer between the system and the surroundings. Then the energy balance for this process can be written as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mc(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(80\text{ kg})\{(2.11\text{ kJ/kg}\cdot^\circ\text{C})[0-(-5)]^\circ\text{C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 0)^\circ\text{C}\}$$
$$+(1000\text{ kg})(4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 20)^\circ\text{C} = 0$$

It gives $T_2 =$ **12.42°C**

which is the final equilibrium temperature in the tank.

(*b*) We take the ice and the water as our system, which is a closed system .Considering that the tank is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{ice}} + \Delta S_{water}$$

where

$$\Delta S_{\text{water}} = \left(mc\ln\frac{T_2}{T_1}\right)_{\text{water}} = (1000\text{ kg})(4.18\text{ kJ/kg}\cdot\text{K})\ln\frac{285.42\text{ K}}{293\text{ K}} = -109.590\text{ kJ/K}$$

$$\Delta S_{\text{ice}} = (\Delta S_{\text{solid}} + \Delta S_{\text{melting}} + \Delta S_{\text{liquid}})_{\text{ice}}$$

$$= \left(\left(mc\ln\frac{T_{\text{melting}}}{T_1}\right)_{\text{solid}} + \frac{mh_{ig}}{T_{\text{melting}}} + \left(mc\ln\frac{T_2}{T_1}\right)_{\text{liquid}}\right)_{\text{ice}}$$

$$= (80\text{ kg})\left((2.11\text{ kJ/kg}\cdot\text{K})\ln\frac{273\text{ K}}{268\text{ K}} + \frac{333.7\text{ kJ/kg}}{273\text{ K}} + (4.18\text{ kJ/kg}\cdot\text{K})\ln\frac{285.42\text{ K}}{273\text{ K}}\right)$$

$$= 115.783\text{ kJ/K}$$

Then, $S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{ice}} = -109.590 + 115.783 = 6.193$ kJ/K

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293\text{ K})(6.193\text{ kJ/K}) = \mathbf{1815\ kJ}$$

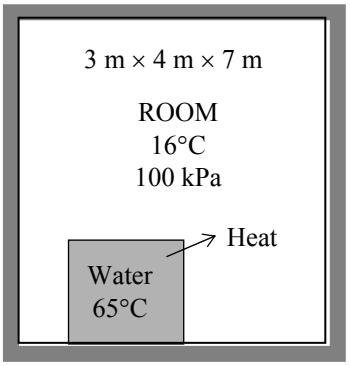
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8-121 One ton of liquid water at 65°C is brought into a room. The final equilibrium temperature in the room and the entropy generated are to be determined.

Assumptions **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant at room temperature. **3** The system is stationary and thus the kinetic and potential energy changes are zero. **4** There are no work interactions involved.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). The constant volume specific heat of water at room temperature is $c_v = 0.718$ kJ/kg·°C (Table A-2). The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis The volume and the mass of the air in the room are



$$V = 3 \times 4 \times 7 = 84 \text{ m}^3$$

$$m_{\text{air}} = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(84 \text{ m}^3)}{(0.2870 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(289 \text{ K})} = 101.3 \text{ kg}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

or $$[mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$$

Substituting, $$(1000 \text{ kg})(4.18 \text{ kJ/kg} \cdot °\text{C})(T_f - 65)°\text{C} + (101.3 \text{ kg})(0.718 \text{ kJ/kg} \cdot °\text{C})(T_f - 16)°\text{C} = 0$$

It gives the final equilibrium temperature in the room to be

$$T_f = \mathbf{64.2°C}$$

(*b*) We again take the room and the water in it as the system, which is a closed system. Considering that the system is well-insulated and no mass is entering and leaving, the entropy balance for this system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{air}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{air}} = mc_v \ln \frac{T_2}{T_1} + mR \ln \frac{V_2}{V_1}^{\nearrow 0} = (101.3 \text{ kg})(0.718 \text{ kJ/kg} \cdot \text{K}) \ln \frac{337.2 \text{ K}}{289 \text{ K}} = 11.21 \text{ kJ/K}$$

$$\Delta S_{\text{water}} = mc \ln \frac{T_2}{T_1} = (1000 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K}) \ln \frac{337.2 \text{ K}}{338 \text{ K}} = -10.37 \text{ kJ/K}$$

Substituting, the entropy generation is determined to be

$$S_{\text{gen}} = 11.21 - 10.37 = 0.834 \text{ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (283 \text{ K})(0.834 \text{ kJ/K}) = \mathbf{236 \text{ kJ}}$$

(c) The work potential (the maximum amount of work that can be produced) during a process is simply the reversible work output. Noting that the actual work for this process is zero, it becomes

$$X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act,out}} \rightarrow W_{\text{rev,out}} = X_{\text{destroyed}} = \mathbf{236 \text{ kJ}}$$

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8-122 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established and the amount of exergy destroyed are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1).

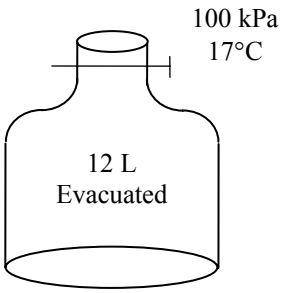
Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2$ (since $m_{\text{out}} = m_{\text{initial}} = 0$)

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = \text{ke} \cong \text{pe} \cong 0)$$



Combining the two balances:

$$Q_{\text{in}} = m_2 (u_2 - h_i)$$

where

$$m_2 = \frac{P_2 \mathcal{V}}{R T_2} = \frac{(100 \text{ kPa})(0.012 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 0.0144 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{matrix} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{matrix}$$

Substituting,

$$Q_{\text{in}} = (0.0144 \text{ kg})(206.91 - 290.16) \text{ kJ/kg} = -1.2 \text{ kJ} \quad \rightarrow \quad Q_{\text{out}} = \mathbf{1.2 \text{ kJ}}$$

Note that the negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reversed the direction.

The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the bottle and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$m_i s_i - \frac{Q_{\text{out}}}{T_{\text{b,in}}} + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1 \cancelto{0}{} = m_2 s_2$$

Therefore, the total entropy generated during this process is

$$S_{\text{gen}} = -m_i s_i + m_2 s_2 + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = m_2 (s_2 - s_i) \cancelto{0}{} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{1.2 \text{ kJ}}{290 \text{ K}} = 0.00415 \text{ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (290 \text{ K})(0.00415 \text{ kJ/K}) = \mathbf{1.2 \text{ kJ}}$$

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8-123 Argon gas in a piston–cylinder device expands isothermally as a result of heat transfer from a furnace. The useful work output, the exergy destroyed, and the reversible work are to be determined.

Assumptions **1** Argon at specified conditions can be treated as an ideal gas since it is well above its critical temperature of 151 K. **2** The kinetic and potential energies are negligible.

Analysis We take the *argon gas* contained within the piston–cylinder device as the system. This is a *closed system* since no mass crosses the system boundary during the process. We note that heat is transferred to the system

from a source at 1200 K, but there is no heat exchange with the environment at 300 K. Also, the temperature of the system remains constant during the expansion process, and its volume doubles, that is, $T_2 = T_1$ and $V_2 = 2V_1$.

(*a*) The only work interaction involved during this isothermal process is the quasi-equilibrium boundary work, which is determined from

$$W = W_{\rm b} = \int_1^2 P dV = P_1 V_1 \ln \frac{V_2}{V_1} = (350\ {\rm kPa})(0.01\ {\rm m}^3)\ln \frac{0.02\ {\rm m}^3}{0.01\ {\rm m}^3} = 2.43\ {\rm kPa \cdot m}^3 = 2.43\ {\rm kJ}$$

This is the total boundary work done by the argon gas. Part of this work is done against the atmospheric pressure P_0 to push the air out of the way, and it cannot be used for any useful purpose. It is determined from

$$W_{\rm surr} = P_0(V_2 - V_1) = (100\ {\rm kPa})(0.02 - 0.01){\rm m}^3 = 1\ {\rm kPa \cdot m}^3 = 1\ {\rm kJ}$$

The useful work is the difference between these two:

$$W_{\rm u} = W - W_{\rm surr} = 2.43 - 1 = \mathbf{1.43\ kJ}$$

That is, 1.43 kJ of the work done is available for creating a useful effect such as rotating a shaft. Also, the heat transfer from the furnace to the system is determined from an energy balance on the system to be

$$\underbrace{E_{\rm in} - E_{\rm out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\rm system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\rm in} - Q_{\rm b,out} = \Delta U = mc_v \Delta T = 0$$

$$Q_{\rm in} = Q_{\rm b,out} = 2.43\ {\rm kJ}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance, or directly from $X_{\rm destroyed} = T_0 S_{\rm gen}$. We will use the second approach since it is usually easier. But first we determine the entropy generation by

applying an entropy balance on an *extended system* (system + immediate surroundings), which includes the temperature gradient zone between the cylinder and the furnace so that the temperature at the boundary where heat

transfer occurs is T_R = 1200 K. This way, the entropy generation associated with the heat transfer is included. Also, the entropy change of the argon gas can be determined from $Q/T_{\rm sys}$ since its temperature remains constant.

$$\underbrace{S_{\rm in} - S_{\rm out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\rm gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\rm system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q}{T_R} + S_{\rm gen} = \Delta S_{\rm sys} = \frac{Q}{T_{sys}}$$

Therefore,

$$S_{\rm gen} = \frac{Q}{T_{sys}} - \frac{Q}{T_R} = \frac{2.43\ {\rm kJ}}{400\ {\rm K}} - \frac{2.43\ {\rm kJ}}{1200\ {\rm K}} = 0.00405\ {\rm kJ/K}$$

and

$$X_{\rm dest} = T_0 S_{\rm gen} = (300\ {\rm K})(0.00405\ {\rm kJ/K}) = \mathbf{1.22\ kJ}$$

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(*c*) The reversible work, which represents the maximum useful work that could be produced $W_{\text{rev,out}}$, can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} \quad \underbrace{-X_{\text{destroyed}}^{\nearrow 0 \text{ (reversible)}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}}$$

$$\left(1-\frac{T_0}{T_b}\right)Q - W_{\text{rev,out}} = X_2 - X_1$$

$$= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)$$

$$= 0 + W_{\text{surr}} - T_0\frac{Q}{T_{sys}}$$

since $\Delta\text{KE} = \Delta\text{PE} = 0$ and $\Delta U = 0$ (the change in internal energy of an ideal gas is zero during an isothermal process), and $\Delta S_{\text{sys}} = Q/T_{\text{sys}}$ for isothermal processes in the absence of any irreversibilities. Then,

$$W_{\text{rev,out}} = T_0\frac{Q}{T_{sys}} - W_{\text{surr}} + \left(1-\frac{T_0}{T_R}\right)Q$$

$$= (300\,\text{K})\frac{2.43\,\text{kJ}}{400\,\text{K}} - (1\,\text{kJ}) + \left(1-\frac{300\,\text{K}}{1200\,\text{K}}\right)(2.43\,\text{kJ})$$

$$= \mathbf{2.65\,kJ}$$

Therefore, the useful work output would be 2.65 kJ instead of 1.43 kJ if the process were executed in a totally reversible manner.

Alternative Approach The reversible work could also be determined by applying the basics only, without resorting to exergy balance. This is done by replacing the irreversible portions of the process by reversible ones that create

the same effect on the system. The useful work output of this idealized process (between the actual end states) is the reversible work. The only irreversibility the actual process involves is the heat transfer between the system and the furnace through a finite temperature difference. This irreversibility can be eliminated by operating a reversible heat engine between the furnace at 1200 K and the surroundings at 300 K. When 2.43 kJ of heat is supplied to this heat engine, it produces a work output of

$$W_{\text{HE}} = \eta_{rev}Q_H = \left(1-\frac{T_L}{T_H}\right)Q_H = \left(1-\frac{300\,\text{K}}{1200\,\text{K}}\right)(2.43\,\text{kJ}) = 1.82\,\text{kJ}$$

The 2.43 kJ of heat that was transferred to the system from the source is now extracted from the surrounding air at 300 K by a reversible heat pump that requires a work input of

$$W_{\text{HP,in}} = \frac{Q_H}{COP_{\text{HP}}} = \frac{Q_H}{T_H/(T_H - T_L)} = \frac{2.43\,\text{kJ}}{(400\,\text{K})/(400-300)\text{K}} = 0.61\,\text{kJ}$$

Then the net work output of this reversible process (i.e., the reversible work) becomes

$$W_{\text{rev}} = W_u + W_{\text{HE}} - W_{\text{HP,in}} = 1.43 + 1.82 - 0.61 = 2.64\,\text{kJ}$$

which is practically identical to the result obtained before. Also, the exergy destroyed is the difference between the reversible work and the useful work, and is determined to be

$$X_{\text{dest}} = W_{\text{rev,out}} - W_{\text{u,out}} = 2.65 - 1.43 = 1.22\,\text{kJ}$$

which is identical to the result obtained before.

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8-124 A heat engine operates between two constant-pressure cylinders filled with air at different temperatures. The maximum work that can be produced and the final temperatures of the cylinders are to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg.K (Table A-2).

Analysis For maximum power production, the entropy generation must be zero. We take the two cylinders (the heat source and heat sink) and the heat engine as the system. Noting that the system involves no heat and mass transfer and that the entropy change for cyclic devices is zero, the entropy balance can be expressed as

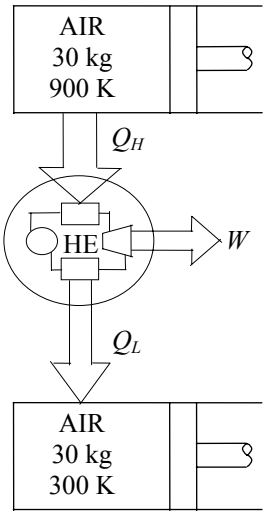
$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\text{Net entropy transfer by heat and mass}} + \underbrace{S_{\text{gen}}^{\nearrow 0}}_{\text{Entropy generation}} = \underbrace{\Delta S_{\text{system}}}_{\text{Change in entropy}}$$

$$0 + S_{\text{gen}}^{\nearrow 0} = \Delta S_{\text{cylinder,source}} + \Delta S_{\text{cylinder,sink}} + \Delta S_{\text{heat engine}}^{\nearrow 0}$$

$$\Delta S_{\text{cylinder,source}} + \Delta S_{\text{cylinder,sink}} = 0$$

$$\left(mc_p \ln \frac{T_2}{T_1} - mR \ln \frac{P_2}{P_1}^{\nearrow 0} \right)_{\text{source}} + 0 + \left(mc_p \ln \frac{T_2}{T_1} - mR \ln \frac{P_2}{P_1}^{\nearrow 0} \right)_{\text{sink}} = 0$$

$$\ln \frac{T_2}{T_{1A}} \frac{T_2}{T_{1B}} = 0 \longrightarrow T_2^2 = T_{1A} T_{1B}$$



where T_{1A} and T_{1B} are the initial temperatures of the source and the sink, respectively, and T_2 is the common final temperature. Therefore, the final temperature of the tanks for maximum power production is

$$T_2 = \sqrt{T_{1A} T_{1B}} = \sqrt{(900\text{ K})(300\text{ K})} = \mathbf{519.6\ K}$$

The energy balance $E_{in} - E_{out} = \Delta E_{\text{system}}$ for the source and sink can be expressed as follows:

Source:

$$-Q_{\text{source,out}} + W_{b,in} = \Delta U \rightarrow Q_{\text{source,out}} = \Delta H = mc_p (T_{1A} - T_2)$$

$$Q_{\text{source,out}} = mc_p (T_{1A} - T_2) = (30\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(900 - 519.6)\text{K} = 11{,}469\text{ kJ}$$

Sink:

$$Q_{\text{sink,in}} - W_{b,out} = \Delta U \rightarrow Q_{\text{sink,in}} = \Delta H = mc_p (T_2 - T_{1A})$$

$$Q_{\text{sink,in}} = mc_p (T_2 - T_{1B}) = (30\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(519.6 - 300)\text{K} = 6621\text{ kJ}$$

Then the work produced becomes

$$W_{\text{max,out}} = Q_H - Q_L = Q_{\text{source,out}} - Q_{\text{sink,in}} = 11{,}469 - 6621 = \mathbf{4847\ kJ}$$

Therefore, a maximum of 4847 kJ of work can be produced during this process

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8-125 A heat engine operates between a nitrogen tank and an argon cylinder at different temperatures. The maximum work that can be produced and the final temperatures are to be determined.

Assumptions Nitrogen and argon are ideal gases with constant specific heats at room temperature.

Properties The constant volume specific heat of nitrogen at room temperature is $c_v = 0.743$ kJ/kg.K. The constant pressure specific heat of argon at room temperature is $c_p = 0.5203$ kJ/kg.K (Table A-2).

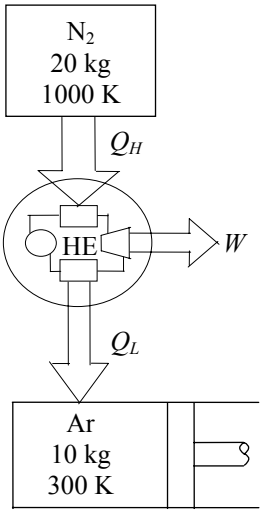
Analysis For maximum power production, the entropy generation must be zero. We take the tank, the cylinder (the heat source and the heat sink) and the heat engine as the system. Noting that the system involves no heat and mass transfer and that the entropy change for cyclic devices is zero, the entropy balance can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}^{\nearrow 0}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}}^{\nearrow 0} = \Delta S_{\text{tank,source}} + \Delta S_{\text{cylinder,sink}} + \Delta S_{\text{heat engine}}^{\nearrow 0}$$

$$(\Delta S)_{\text{source}} + (\Delta S)_{\text{sink}} = 0$$

$$\left(mc_v \ln \frac{T_2}{T_1} - mR \ln \frac{\cancel{V_2}^{\nearrow 0}}{V_1} \right)_{\text{source}} + 0 + \left(mc_p \ln \frac{T_2}{T_1} - mR \ln \frac{\cancel{P_2}^{\nearrow 0}}{P_1} \right)_{\text{sink}} = 0$$



Substituting,

$$(20\text{ kg})(0.743\text{ kJ/kg}\cdot\text{K}) \ln \frac{T_2}{1000\text{ K}} + (10\text{ kg})(0.5203\text{ kJ/kg}\cdot\text{K}) \ln \frac{T_2}{300\text{ K}} = 0$$

Solving for T_2 yields

$T_2 =$ **731.8 K**

where T_2 is the common final temperature of the tanks for maximum power production.

The energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ for the source and sink can be expressed as follows:

Source:

$$-Q_{\text{source,out}} = \Delta U = mc_v (T_2 - T_{1A}) \rightarrow Q_{\text{source,out}} = mc_v (T_{1A} - T_2)$$

$$Q_{\text{source,out}} = mc_v (T_{1A} - T_2) = (20\text{ kg})(0.743\text{ kJ/kg}\cdot\text{K})(1000 - 731.8)\text{K} = 3985\text{ kJ}$$

Sink:

$$Q_{\text{sink,in}} - W_{\text{b,out}} = \Delta U \rightarrow Q_{\text{sink,in}} = \Delta H = mc_p (T_2 - T_{1A})$$

$$Q_{\text{sink,in}} = mc_v (T_2 - T_{1A}) = (10\text{ kg})(0.5203\text{ kJ/kg}\cdot\text{K})(731.8 - 300)\text{K} = 2247\text{ kJ}$$

Then the work produced becomes

$$W_{\text{max,out}} = Q_H - Q_L = Q_{\text{source,out}} - Q_{\text{sink,in}} = 3985 - 2247 = \mathbf{1739\ kJ}$$

Therefore, a maximum of 1739 kJ of work can be produced during this process

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8-126 A rigid tank containing nitrogen is considered. Heat is now transferred to the nitrogen from a reservoir and nitrogen is allowed to escape until the mass of nitrogen becomes one-half of its initial mass. The change in the nitrogen's work potential is to be determined.

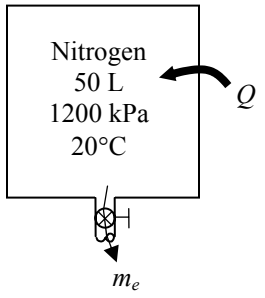
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** Nitrogen is an ideal gas with constant specific heats.

Properties The properties of nitrogen at room temperature are c_p = 1.039 kJ/kg·K, c_v = 0.743 kJ/kg·K, and R = 0.2968 kJ/kg·K (Table A-2a).

Analysis The initial and final masses in the tank are

$$m_1 = \frac{P\mathcal{V}}{RT_1} = \frac{(1200\ \text{kPa})(0.050\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 0.690\ \text{kg}$$

$$m_2 = m_e = \frac{m_1}{2} = \frac{0.690\ \text{kg}}{2} = 0.345\ \text{kg}$$



The final temperature in the tank is

$$T_2 = \frac{P\mathcal{V}}{m_2 R} = \frac{(1200\ \text{kPa})(0.050\ \text{m}^3)}{(0.345\ \text{kg})(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 586\ \text{K}$$

We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$Q_{\text{out}} = m_e h_e + m_2 u_2 - m_1 u_1$$

Using the average of the initial and final temperatures for the exiting nitrogen, $T_e = 0.5(T_1 + T_2) = 0.5((293 + 586) = 439.5\ \text{K}$ this energy balance equation becomes

$$\begin{aligned} Q_{\text{out}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= m_e c_p T_e + m_2 c_v T_2 - m_1 c_v T_1 \\ &= (0.345)(1.039)(439.5) + (0.345)(0.743)(586) - (0.690)(0.743)(293) \\ &= 157.5\ \text{kJ} \end{aligned}$$

The work potential associated with this process is equal to the exergy destroyed during the process. The exergy destruction during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on the system:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_R} - m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_R}$$

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Noting that pressures are same, rearranging and substituting gives

$$\begin{aligned} S_{\text{gen}} &= m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_R} \\ &= m_2 c_p \ln T_2 - m_1 c_p \ln T_1 + m_e c_p \ln T_e - \frac{Q_{\text{in}}}{T_R} \\ &= (0.345)(1.039)\ln(586) - (0.690)(1.039)\ln(293) + (0.345)(1.039)\ln(439.5) - \frac{157.5}{773} \\ &= 0.190 \text{ kJ/K} \end{aligned}$$

Then,

$$W_{\text{rev}} = X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293 \text{ K})(0.190 \text{ kJ/K}) = \mathbf{55.7 \text{ kJ}}$$

Alternative More Accurate Solution

This problem may also be solved by considering the variation of gas temperature at the outlet of the tank. The mass and energy balances are

$$\dot{m}_e = -\frac{dm}{dt}$$

$$\dot{Q} = \frac{d(mu)}{dt} - h\frac{dm}{dt} = \frac{c_v d(mT)}{dt} - c_p T \frac{dm}{dt}$$

Combining these expressions and replacing T in the last term gives

$$\dot{Q} = c_v \frac{d(mT)}{dt} - \frac{c_p P \mathbf{V}}{Rm}\frac{dm}{dt}$$

Integrating this over the time required to release one-half the mass produces

$$Q = c_v (m_2 T_2 - m_1 T_1) - \frac{c_p P \mathbf{V}}{R} \ln \frac{m_2}{m_1}$$

The reduced combined first and second law becomes

$$\dot{W}_{\text{rev}} = \dot{Q}\left(1 - \frac{T_0}{T_R}\right) - \frac{d(U - T_0 S)}{dt} + (h - T_0 s)\frac{dm}{dt}$$

when the mass balance is substituted and the entropy generation is set to zero (for maximum work production). Expanding the system time derivative gives

$$\begin{aligned} \dot{W}_{\text{rev}} &= \dot{Q}\left(1 - \frac{T_0}{T_R}\right) - \frac{d(mu - T_0 ms)}{dt} + (h - T_0 s)\frac{dm}{dt} \\ &= \dot{Q}\left(1 - \frac{T_0}{T_R}\right) - \frac{d(mu)}{dt} + T_0 m\frac{ds}{dt} + T_0 s\frac{dm}{dt} + (h - T_0 s)\frac{dm}{dt} \\ &= \dot{Q}\left(1 - \frac{T_0}{T_R}\right) - \frac{d(mu)}{dt} + h\frac{dm}{dt} + m\frac{T_0}{T}\frac{dh}{dt} \end{aligned}$$

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Substituting $\dot{Q}$ from the first law,

$$\begin{aligned}\dot{W}_{\text{rev}} &= \left[\frac{d(mu)}{dt} - h\frac{dm}{dt}\right]\left(1 - \frac{T_0}{T_R}\right) - \left[\frac{d(mu)}{dt} - h\frac{dm}{dt}\right] + m\frac{T_0}{T}\frac{dh}{dt} \\ &= -\frac{T_0}{T_R}\left[\frac{d(mu)}{dt} - h\frac{dm}{dt} - m\frac{dh}{dt}\right] \\ &= -\frac{T_0}{T_R}\left[c_v\frac{d(mT)}{dt} - c_pT\frac{dm}{dt} - mc_p\frac{dT}{dt}\right]\end{aligned}$$

At any time,

$$T = \frac{P\mathcal{V}}{mR}$$

which further reduces this result to

$$\dot{W}_{\text{rev}} = \frac{T_0}{T_R}c_p\frac{P\mathcal{V}}{mR}\frac{dm}{dt} + T_0m\left(\frac{c_p}{T}\frac{dT}{dt} - \frac{R}{P}\frac{dP}{dt}\right)$$

When this integrated over the time to complete the process, the result is

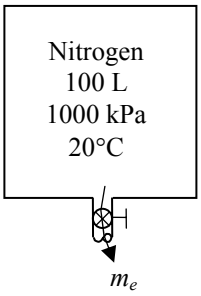
$$\begin{aligned}W_{\text{rev}} &= \frac{T_0}{T_R}\frac{c_pP\mathcal{V}}{R}\ln\frac{m_2}{m_1} + T_0\frac{c_pP\mathcal{V}}{R}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \\ &= \frac{293}{773}\frac{(1.039)(1200)(0.050)}{0.2968}\ln\frac{1}{2} + (293)\frac{(1.039)(1200)(0.050)}{0.2968}\left(\frac{1}{293} - \frac{1}{586}\right) \\ &= \mathbf{49.8\ kJ}\end{aligned}$$

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8-127 A rigid tank containing nitrogen is considered. Nitrogen is allowed to escape until the mass of nitrogen becomes one-half of its initial mass. The change in the nitrogen's work potential is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** Nitrogen is an ideal gas with constant specific heats.

Properties The properties of nitrogen at room temperature are $c_p = 1.039$ kJ/kg·K, $c_v = 0.743$ kJ/kg·K, $k = 1.4$, and $R = 0.2968$ kJ/kg·K (Table A-2a).



Analysis The initial and final masses in the tank are

$$m_1 = \frac{P\boldsymbol{V}}{RT_1} = \frac{(1000\,\text{kPa})(0.100\,\text{m}^3)}{(0.2968\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})} = 1.150\,\text{kg}$$

$$m_2 = m_e = \frac{m_1}{2} = \frac{1.150\,\text{kg}}{2} = 0.575\,\text{kg}$$

We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-m_e h_e = m_2 u_2 - m_1 u_1$$

Using the average of the initial and final temperatures for the exiting nitrogen, this energy balance equation becomes

$$-m_e h_e = m_2 u_2 - m_1 u_1$$

$$-m_e c_p T_e = m_2 c_v T_2 - m_1 c_v T_1$$

$$-(0.575)(1.039)(0.5)(293 + T_2) = (0.575)(0.743)T_2 - (1.150)(0.743)(293)$$

Solving for the final temperature, we get

$$T_2 = 224.3\,\text{K}$$

The final pressure in the tank is

$$P_2 = \frac{m_2 R T_2}{\boldsymbol{V}} = \frac{(0.575\,\text{kg})(0.2968\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(224.3\,\text{K})}{0.100\,\text{m}^3} = 382.8\,\text{kPa}$$

The average temperature and pressure for the exiting nitrogen is

$$T_e = 0.5(T_1 + T_2) = 0.5(293 + 224.3) = 258.7\,\text{K}$$

$$P_e = 0.5(P_1 + P_2) = 0.5(1000 + 382.8) = 691.4\,\text{kPa}$$

The work potential associated with this process is equal to the exergy destroyed during the process. The exergy destruction during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on the system:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 + m_e s_e$$

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Rearranging and substituting gives

$$\begin{aligned}
S_{\text{gen}} &= m_2 s_2 - m_1 s_1 + m_e s_e \\
&= m_2 (c_p \ln T_2 - R \ln P_2) - m_1 (c_p \ln T_1 - R \ln P_1) + m_e (c_p \ln T_e - R \ln P_e) \\
&= (0.575)\left[1.039 \ln(224.3) - (0.2968) \ln(382.8)\right] - (1.15)\left[1.039 \ln(293) - (0.2968) \ln(1000)\right] \\
&\quad + (0.575)\left[1.039 \ln(258.7) - (0.2968) \ln(691.4)\right] \\
&= 2.2188 - 4.4292 + 2.2032 = -0.007152 \text{ kJ/K}
\end{aligned}$$

Then,

$$W_{\text{rev}} = X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293 \text{ K})(-0.007152 \text{ kJ/K}) = \mathbf{-2.10 \text{ kJ}}$$

The entropy generation cannot be negative for a thermodynamically possible process. This result is probably due to using average temperature and pressure values for the exiting gas and using constant specific heats for nitrogen. This sensitivity occurs because the entropy generation is very small in this process.

Alternative More Accurate Solution

This problem may also be solved by considering the variation of gas temperature and pressure at the outlet of the tank. The mass balance in this case is

$$\dot{m}_e = -\frac{dm}{dt}$$

which when combined with the reduced first law gives

$$\frac{d(mu)}{dt} = h \frac{dm}{dt}$$

Using the specific heats and the ideal gas equation of state reduces this to

$$c_v \frac{\boldsymbol{V}}{R} \frac{dP}{dt} = c_p T \frac{dm}{dt}$$

which upon rearrangement and an additional use of ideal gas equation of state becomes

$$\frac{1}{P} \frac{dP}{dt} = \frac{c_p}{c_v} \frac{1}{m} \frac{dm}{dt}$$

When this is integrated, the result is

$$P_2 = P_1 \left(\frac{m_2}{m_1} \right)^k = 1000 \left(\frac{1}{2} \right)^{1.4} = 378.9 \text{ kPa}$$

The final temperature is then

$$T_2 = \frac{P_2 \boldsymbol{V}}{m_2 R} = \frac{(378.9 \text{ kPa})(0.100 \text{ m}^3)}{(0.575 \text{ kg})(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})} = 222.0 \text{ K}$$

The process is then one of

$$\frac{m^k}{P} = \text{const} \quad \text{or} \quad \frac{m^{k-1}}{T} = \text{const}$$

The reduced combined first and second law becomes

$$\dot{W}_{\text{rev}} = -\frac{d(U - T_0 S)}{dt} + (h - T_0 s) \frac{dm}{dt}$$

when the mass balance is substituted and the entropy generation is set to zero (for maximum work production). Replacing the enthalpy term with the first law result and canceling the common *dU*/*dt* term reduces this to

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$$\dot{W}_{\text{rev}} = T_0 \frac{d(ms)}{dt} - T_0 s \frac{dm}{dt}$$

Expanding the first derivative and canceling the common terms further reduces this to

$$\dot{W}_{\text{rev}} = T_0 m \frac{ds}{dt}$$

Letting $a = P_1 / m_1^k$ and $b = T_1 / m_1^{k-1}$, the pressure and temperature of the nitrogen in the system are related to the mass by

$$P = am^k \quad \text{and} \quad T = bm^{k-1}$$

according to the first law. Then,

$$dP = akm^{k-1} dm \quad \text{and} \quad dT = b(k-1)m^{k-2} dm$$

The entropy change relation then becomes

$$ds = c_p \frac{dT}{T} - R \frac{dP}{P} = \left[(k-1)c_p - Rk\right] \frac{dm}{m}$$

Now, multiplying the combined first and second laws by *dt* and integrating the result gives

$$\begin{aligned} W_{\text{rev}} &= T_0 \int_1^2 mds = T_0 \int_1^2 mds \left[(k-1)c_p - Rk\right] dm \\ &= T_0 \left[(k-1)c_p - Rk\right](m_2 - m_1) \\ &= (293)\left[(1.4-1)(1.039) - (0.2968)(1.4)\right](0.575 - 1.15) \\ &= \mathbf{-0.0135\ kJ} \end{aligned}$$

Once again the entropy generation is negative, which cannot be the case for a thermodynamically possible process. This is probably due to using constant specific heats for nitrogen. This sensitivity occurs because the entropy generation is very small in this process.

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8-128 Steam is condensed by cooling water in the condenser of a power plant. The rate of condensation of steam and the rate of exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The enthalpy and entropy of vaporization of water at 45°C are $h_{fg} = 2394.0$ kJ/kg and $s_{fg} = 7.5247$ kJ/kg.K (Table A-4). The specific heat of water at room temperature is $c_p = 4.18$ kJ/kg.°C (Table A-3).

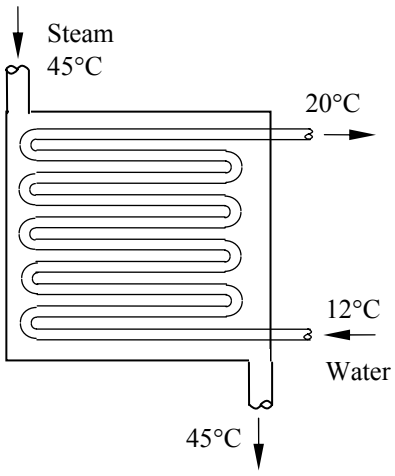
Analysis (*a*) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the heat transfer rate to the cooling water in the condenser becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cooling water}}$$
$$= (330 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(20°\text{C} - 12°\text{C})$$
$$= 11{,}035 \text{ kJ/s}$$

The rate of condensation of steam is determined to be

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{11{,}035 \text{ kJ/s}}{2394.0 \text{ kJ/kg}} = \mathbf{4.61 \text{ kg/s}}$$

(*b*) The rate of entropy generation within the condenser during this process can be determined by applying the rate form of the entropy balance on the entire condenser. Noting that the condenser is well-insulated and thus heat transfer is negligible, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{steam}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{steam}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{steam}}(s_f - s_g) = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} - \dot{m}_{\text{steam}} s_{fg}$$
$$= (330 \text{ kg/s})(4.18 \text{ kJ/kg.K})\ln\frac{20 + 273}{12 + 273} - (4.61 \text{ kg/s})(7.5247 \text{ kJ/kg.K}) = 3.501 \text{ kW/K}$$

Then the exergy destroyed can be determined directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (285 \text{ K})(3.501 \text{ kW/K}) = \mathbf{998 \text{ kW}}$$

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8-129 A system consisting of a compressor, a storage tank, and a turbine as shown in the figure is considered. The change in the exergy of the air in the tank and the work required to compress the air as the tank was being filled are to be determined.

Assumptions **1** Changes in the kinetic and potential energies are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $R = 0.287$ kPa·m^3/kg·K, $c_p = 1.005$ kJ/kg·K, $c_v = 0.718$ kJ/kg·K, $k = 1.4$ (Table A-2a).

Analysis The initial mass of air in the tank is

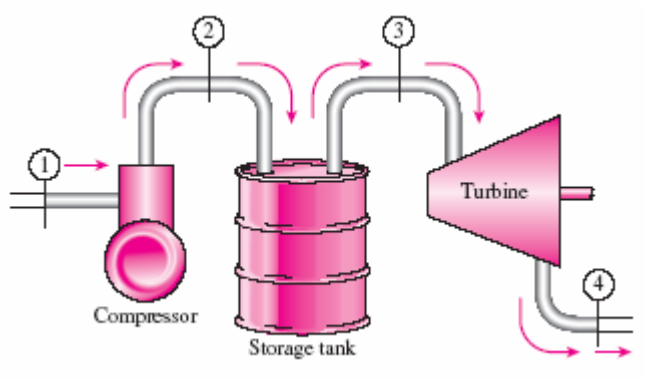
$$m_{\text{initial}} = \frac{P_{\text{initial}}\mathcal{V}}{RT_{\text{initial}}} = \frac{(100\,\text{kPa})(5\times10^5\ \text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})} = 0.5946\times10^6\ \text{kg}$$

and the final mass in the tank is

$$m_{\text{final}} = \frac{P_{\text{final}}\mathcal{V}}{RT_{\text{final}}} = \frac{(600\,\text{kPa})(5\times10^5\ \text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})} = 3.568\times10^6\ \text{kg}$$

Since the compressor operates as an isentropic device,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$



The conservation of mass applied to the tank gives

$$\frac{dm}{dt} = \dot{m}_{\text{in}}$$

while the first law gives

$$\dot{Q} = \frac{d(mu)}{dt} - h\frac{dm}{dt}$$

Employing the ideal gas equation of state and using constant specific heats, expands this result to

$$\dot{Q} = \frac{\mathcal{V}c_v}{R}\frac{dP}{dt} - c_pT_2\frac{\mathcal{V}}{RT}\frac{dP}{dt}$$

Using the temperature relation across the compressor and multiplying by dt puts this result in the form

$$\dot{Q}dt = \frac{\mathcal{V}c_v}{R}dP - c_pT_1\left(\frac{P}{P_1}\right)^{(k-1)/k}\frac{\mathcal{V}}{RT}dP$$

When this integrated, it yields (i and f stand for initial and final states)

$$Q = \frac{\mathcal{V}c_v}{R}(P_f - P_i) - \frac{k}{2k-1}\frac{c_p\mathcal{V}}{R}\left[P_f\left(\frac{P_f}{P_i}\right)^{(k-1)/k} - P_i\right]$$

$$= \frac{(5\times10^5)(0.718)}{0.287}(600-100) - \frac{1.4}{2(1.4)-1}\frac{(1.005)(5\times10^5)}{0.287}\left[600\left(\frac{600}{100}\right)^{0.4/1.4} - 100\right]$$

$$= -6.017\times10^8\ \text{kJ}$$

The negative result show that heat is transferred from the tank. Applying the first law to the tank and compressor gives

$$(\dot{Q} - \dot{W}_{\text{out}})dt = d(mu) - h_1dm$$

which integrates to

$$Q - W_{\text{out}} = (m_fu_f - m_iu_i) - h_1(m_f - m_i)$$

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Upon rearrangement,

$$\begin{aligned} W_{\text{out}} &= Q + (c_p - c_v)T(m_f - m_i) \\ &= -6.017\times10^8 + (1.005 - 0.718)(293)[(3.568 - 0.5946)\times10^6] \\ &= \mathbf{-3.516\times10^8\ kJ} \end{aligned}$$

The negative sign shows that work is done on the compressor. When the combined first and second laws is reduced to fit the compressor and tank system and the mass balance incorporated, the result is

$$\dot{W}_{\text{rev}} = \dot{Q}\left(1 - \frac{T_0}{T_R}\right) - \frac{d(U - T_0 S)}{dt} + (h - T_0 s)\frac{dm}{dt}$$

which when integrated over the process becomes

$$\begin{aligned} W_{\text{rev}} &= Q\left(1 - \frac{T_0}{T_R}\right) + m_i\left[(u_i - h_1) - T_0(s_i - s_1)\right] - m_f\left[(u_f - h_1) - T_0(s_f - s_1)\right] \\ &= Q\left(1 - \frac{T_0}{T_R}\right) + m_i\left[T_i(c_v - c_p)\right] - m_f\left[T_f(c_v - c_p) - T_0 R\ln\left(\frac{P_f}{P_i}\right)\right] \\ &= -6.017\times10^8\left(1 - \frac{293}{293}\right) + 0.5946\times10^6\left[(0.718 - 1.005)293\right] \\ &\quad - 3.568\times10^6\left[(0.718 - 1.005)293 + 293(0.287)\ln\frac{600}{100}\right] \\ &= \mathbf{7.875\times10^8\ kJ} \end{aligned}$$

This is the exergy change of the air stored in the tank.

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8-130 The air stored in the tank of the system shown in the figure is released through the isentropic turbine. The work produced and the change in the exergy of the air in the tank are to be determined.

Assumptions **1** Changes in the kinetic and potential energies are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $R = 0.287$ kPa·m^3/kg·K, $c_p = 1.005$ kJ/kg·K, $c_v = 0.718$ kJ/kg·K, $k = 1.4$ (Table A-2a).

Analysis The initial mass of air in the tank is

$$m_{\text{initial}} = \frac{P_{\text{initial}}\boldsymbol{V}}{RT_{\text{initial}}} = \frac{(600\ \text{kPa})(5\times10^5\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 3.568\times10^6\ \text{kg}$$

and the final mass in the tank is

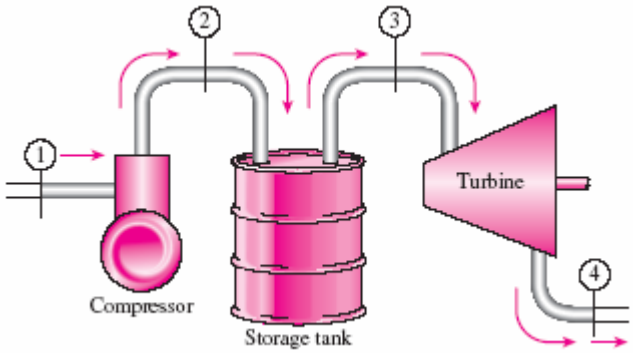
$$m_{\text{final}} = \frac{P_{\text{final}}\boldsymbol{V}}{RT_{\text{final}}} = \frac{(100\ \text{kPa})(5\times10^5\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 0.5946\times10^6\ \text{kg}$$

The conservation of mass is

$$\frac{dm}{dt} = \dot{m}_{\text{in}}$$

while the first law gives

$$\dot{Q} = \frac{d(mu)}{dt} - h\frac{dm}{dt}$$



Employing the ideal gas equation of state and using constant specific heats, expands this result to

$$\begin{aligned}\dot{Q} &= \frac{\boldsymbol{V}c_v}{R}\frac{dP}{dt} - c_pT\frac{\boldsymbol{V}}{RT}\frac{dP}{dt}\\ &= \frac{c_v - c_p}{R}\boldsymbol{V}\frac{dP}{dt}\\ &= -\boldsymbol{V}\frac{dP}{dt}\end{aligned}$$

When this is integrated over the process, the result is (i and f stand for initial and final states)

$$Q = -\boldsymbol{V}(P_f - P_i) = -5\times10^5(100-600) = 2.5\times10^8\ \text{kJ}$$

Applying the first law to the tank and compressor gives

$$(\dot{Q} - \dot{W}_{\text{out}})dt = d(mu) - hdm$$

which integrates to

$$\begin{aligned}Q - W_{\text{out}} &= (m_fu_f - m_iu_i) + h(m_i - m_f)\\ -W_{\text{out}} &= -Q + m_fu_f - m_iu_i + h(m_i - m_f)\\ W_{\text{out}} &= Q - m_fu_f + m_iu_i - h(m_i - m_f)\\ &= Q - m_fc_vT + m_ic_pT - c_pT(m_i - m_f)\\ &= 2.5\times10^8 - (0.5946\times10^6)(0.718)(293) + (3.568\times10^6)(1.005)(293)\\ &\quad -(1.005)(293)(3.568\times10^6 - 0.5946\times10^6)\\ &= \mathbf{3.00\times10^8\ kJ}\end{aligned}$$

This is the work output from the turbine. When the combined first and second laws is reduced to fit the turbine and tank system and the mass balance incorporated, the result is

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$$\dot{W}_{\text{rev}} = \dot{Q}\left(1-\frac{T_0}{T_R}\right) - \frac{d(U-T_0S)}{dt} + (h-T_0s)\frac{dm}{dt}$$

$$= \dot{Q}\left(1-\frac{T_0}{T_R}\right) - (u-T_0s)\frac{dm}{dt} - m\frac{d(u-T_0s)}{dt} + (h-T_0s)\frac{dm}{dt}$$

$$= \dot{Q}\left(1-\frac{T_0}{T_R}\right) + (c_p-c_v)T\frac{dm}{dt} + mT_0\frac{ds}{dt}$$

$$= \dot{Q}\left(1-\frac{T_0}{T_R}\right) + (c_p-c_v)T\frac{dm}{dt} + \boldsymbol{\mathcal{V}}\frac{T_0}{T}(P_f-P_i)$$

where the last step uses entropy change equation. When this is integrated over the process it becomes

$$W_{\text{rev}} = Q\left(1-\frac{T_0}{T_R}\right) + (c_p-c_v)T(m_f-m_i) + \boldsymbol{\mathcal{V}}\frac{T_0}{T}(P_f-P_i)$$

$$= 3.00\times10^8\left(1-\frac{293}{293}\right) + (1.005-0.718)(293)(0.5946-3.568)\times10^6 + 5\times10^5\,\frac{293}{293}(100-600)$$

$$= 0 - 2.500\times10^8 - 2.5\times10^8$$

$$= \mathbf{-5.00\times10^8\ kJ}$$

This is the exergy change of the air in the storage tank.

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8-131 A heat engine operates between a tank and a cylinder filled with air at different temperatures. The maximum work that can be produced and the final temperatures are to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The specific heats of air are $c_v = 0.718$ kJ/kg.K and $c_p = 1.005$ kJ/kg.K (Table A-2).

Analysis For maximum power production, the entropy generation must be zero. We take the tank, the cylinder (the heat source and the heat sink) and the heat engine as the system. Noting that the system involves no heat and mass transfer and that the entropy change for cyclic devices is zero, the entropy balance can be expressed as

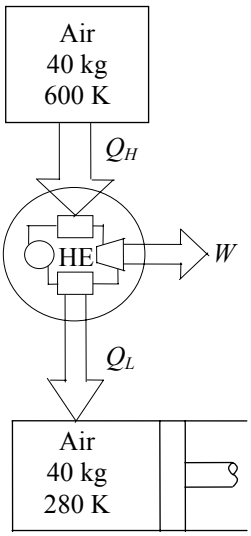
$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\text{Net entropy transfer by heat and mass}} + \underbrace{S_{\text{gen}}^{\nearrow 0}}_{\text{Entropy generation}} = \underbrace{\Delta S_{\text{system}}}_{\text{Change in entropy}}$$

$$0 + S_{\text{gen}}^{\nearrow 0} = \Delta S_{\text{tank,source}} + \Delta S_{\text{cylinder,sink}} + \Delta S_{\text{heat engine}}^{\nearrow 0}$$

$$(\Delta S)_{\text{source}} + (\Delta S)_{\text{sink}} = 0$$

$$\left(mc_v \ln \frac{T_2}{T_1} - mR \ln \frac{v_2}{v_1}^{\nearrow 0} \right)_{\text{source}} + 0 + \left(mc_p \ln \frac{T_2}{T_1} - mR \ln \frac{P_2}{P_1}^{\nearrow 0} \right)_{\text{sink}} = 0$$

$$\ln \frac{T_2}{T_{1A}} + \frac{c_p}{c_v} \ln \frac{T_2}{T_{1B}} = 0 \longrightarrow \frac{T_2}{T_{1A}} \left(\frac{T_2}{T_{1B}} \right)^k = 1 \longrightarrow T_2 = \left(T_{1A} T_{1B}^k \right)^{1/(k+1)}$$



where T_{1A} and T_{1B} are the initial temperatures of the source and the sink, respectively, and T_2 is the common final temperature. Therefore, the final temperature of the tanks for maximum power production is

$$T_2 = \left((600 \text{ K})(280 \text{ K})^{1.4} \right)^{\frac{1}{2.4}} = \mathbf{384.7\ K}$$

Source:

$$-Q_{\text{source,out}} = \Delta U = mc_v (T_2 - T_{1A}) \rightarrow Q_{\text{source,out}} = mc_v (T_{1A} - T_2)$$

$$Q_{\text{source,out}} = mc_v (T_{1A} - T_2) = (40 \text{ kg})(0.718 \text{ kJ/kg} \cdot \text{K})(600 - 384.7)\text{K} = 6184 \text{ kJ}$$

Sink:

$$Q_{\text{sink,in}} - W_{\text{b,out}} = \Delta U \rightarrow Q_{\text{sink,in}} = \Delta H = mc_p (T_2 - T_{1A})$$

$$Q_{\text{sink,in}} = mc_v (T_2 - T_{1A}) = (40 \text{ kg})(1.005 \text{ kJ/kg} \cdot \text{K})(384.7 - 280)\text{K} = 4208 \text{ kJ}$$

Then the work produced becomes

$$W_{\text{max,out}} = Q_H - Q_L = Q_{\text{source,out}} - Q_{\text{sink,in}} = 6184 - 4208 = \mathbf{1977\ kJ}$$

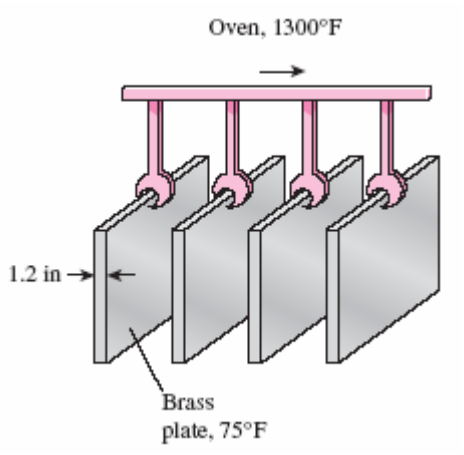
Therefore, a maximum of 1977 kJ of work can be produced during this process.

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8-132E Large brass plates are heated in an oven at a rate of 300/min. The rate of heat transfer to the plates in the oven and the rate of exergy destruction associated with this heat transfer process are to be determined.

Assumptions **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible. **3** The environment temperature is 75°F.

Properties The density and specific heat of the brass are given to be $\rho = 532.5$ lbm/ft^3 and $c_p = 0.091$ Btu/lbm.°F.



Analysis We take the plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

The mass of each plate and the amount of heat transfer to each plate is

$$m = \rho V = \rho LA = (532.5 \text{ lbm/ft}^3)[(1.2/12 \text{ ft})(2 \text{ ft})(2 \text{ ft})] = 213 \text{ lbm}$$

$$Q_{\text{in}} = mc(T_2 - T_1) = (213 \text{ lbm/plate})(0.091 \text{ Btu/lbm.°F})(1000 - 75)\text{°F} = 17{,}930 \text{ Btu/plate}$$

Then the total rate of heat transfer to the plates becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{plate}} Q_{\text{in, per plate}} = (300 \text{ plates/min}) \times (17{,}930 \text{ Btu/plate}) = \mathbf{5{,}379{,}000 \text{ Btu/min} = 89{,}650 \text{ Btu/s}}$$

We again take a single plate as the system. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the plate and its immediate surroundings so that the boundary temperature of the extended system is at 1300°F at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln \frac{T_2}{T_1} = (213 \text{ lbm})(0.091 \text{ Btu/lbm.R}) \ln \frac{(1000 + 460) \text{ R}}{(75 + 460) \text{ R}} = 19.46 \text{ Btu/R}$$ Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{17{,}930 \text{ Btu}}{1300 + 460 \text{ R}} + 19.46 \text{ Btu/R} = 9.272 \text{ Btu/R} \quad \text{(per plate)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = S_{\text{gen}} \dot{n}_{\text{ball}} = (9.272 \text{ Btu/R} \cdot \text{plate})(300 \text{ plates/min}) = 2781 \text{ Btu/min.R} = 46.35 \text{ Btu/s.R}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (535 \text{ R})(46.35 \text{ Btu/s.R}) = \mathbf{24{,}797 \text{ Btu/s}}$$

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8-133 Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven and the rate of exergy destruction associated with this heat transfer process are to be determined.

Assumptions **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible. **3** The environment temperature is 30°C.

Properties The density and specific heat of the steel rods are given to be $\rho = 7833$ kg/m^3 and $c_p = 0.465$ kJ/kg.°C.

Analysis Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho V = \rho LA = \rho L(\pi D^2/4) = (7833\ \text{kg/m}^3)(3\ \text{m})[\pi(0.1\ \text{m})^2/4] = 184.6\ \text{kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (184.6\ \text{kg})(0.465\ \text{kJ/kg.°C})(700 - 30)°\text{C} = 57{,}512\ \text{kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}}/\Delta t = (57{,}512\ \text{kJ})/(1\ \text{min}) = 57{,}512\ \text{kJ/min} = \mathbf{958.5\ kW}$$

We again take the 3-m long section of the rod as the system The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the rod and its immediate surroundings so that the boundary temperature of the extended system is at 900°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (184.6\ \text{kg})(0.465\ \text{kJ/kg.K}) \ln\frac{700 + 273}{30 + 273} = 100.1\ \text{kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{57{,}512\ \text{kJ}}{(900 + 273)\ \text{R}} + 100.1\ \text{kJ/K} = 51.1\ \text{kJ/K}$$

Noting that this much entropy is generated in 1 min, the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = \frac{S_{\text{gen}}}{\Delta t} = \frac{51.1\ \text{kJ/K}}{1\ \text{min}} = 51.1\ \text{kJ/min.K} = 0.852\ \text{kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298\ \text{K})(0.852\ \text{kW/K}) = \mathbf{254\ kW}$$

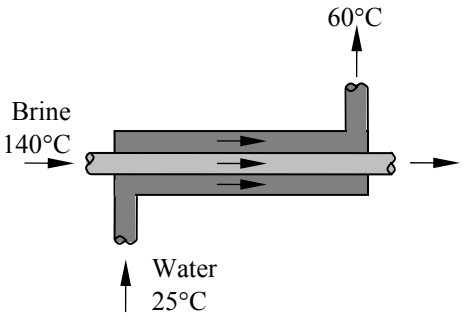
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8-134 Water is heated in a heat exchanger by geothermal water. The rate of heat transfer to the water and the rate of exergy destruction within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** The environment temperature is 25°C.

Properties The specific heats of water and geothermal fluid are given to be 4.18 and 4.31 kJ/kg.°C, respectively.

Analysis (*a*) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in the heat exchanger becomes

$$\dot{Q}_{\text{in,water}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (0.4\text{ kg/s})(4.18\text{ kJ/kg.°C})(60°\text{C} - 25°\text{C}) = \mathbf{58.52\text{ kW}}$$

Noting that heat transfer to the cold water is equal to the heat loss from the geothermal water, the outlet temperature of the geothermal water is determined from

$$\dot{Q}_{\text{out}} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{geo}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}_{\text{out}}}{\dot{m}c_p} = 140°\text{C} - \frac{58.52\text{ kW}}{(0.3\text{ kg/s})(4.31\text{ kJ/kg.°C})} = 94.7°\mathbf{C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{geo}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{geo}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{geo}}(s_4 - s_3)$$

Noting that both fresh and geothermal water are incompressible substances, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{geo}} c_p \ln\frac{T_4}{T_3}$$

$$= (0.4\text{ kg/s})(4.18\text{ kJ/kg.K})\ln\frac{60+273}{25+273} + (0.3\text{ kg/s})(4.31\text{ kJ/kg.K})\ln\frac{94.7+273}{140+273} = 0.0356\text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298\text{ K})(0.0356\text{ kW/K}) = \mathbf{10.61\text{ kW}}$$

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8-135 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year and the rate of exergy destruction within the regenerator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The properties of the milk are constant. **5** The environment temperature is 18°C.

Properties The average density and specific heat of milk can be taken to be $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1$ kg/L and $c_{p,\text{milk}} = 3.79$ kJ/kg.°C (Table A-3).

Analysis The mass flow rate of the milk is

$$\dot{m}_{\text{milk}} = \rho \dot{V}_{\text{milk}} = (1\,\text{kg/L})(12\,\text{L/s}) = 12\,\text{kg/s} = 43{,}200\,\text{kg/h}$$

Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{milk}} c_p (T_2 - T_1)$$

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot{Q}_{\text{current}} = [\dot{m} c_p (T_{\text{pasturization}} - T_{\text{refrigeration}})]_{\text{milk}} = (12\ \text{kg/s})(3.79\ \text{kJ/kg.°C})(72 - 4)°\text{C} = 3093\ \text{kJ/s}$$

The proposed regenerator has an effectiveness of ε = 0.82, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{\text{saved}} = \varepsilon \dot{Q}_{\text{current}} = (0.82)(3093\ \text{kJ/s}) = 2536\ \text{kJ/s}$$

Noting that the boiler has an efficiency of $\eta_{\text{boiler}} = 0.82$, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot{Q}_{\text{saved}}}{\eta_{\text{boiler}}} = \frac{(2536\ \text{kJ/s})}{(0.82)} \frac{(1\text{therm})}{(105{,}500\ \text{kJ})} = 0.02931\,\text{therm/s}$$

Noting that 1 year = 365×24=8760 h and unit cost of natural gas is $1.04/therm, the annual fuel and money savings will be

Fuel Saved = (0.02931 therms/s)(8760×3600 s) = **924,450 therms/yr**

Money saved = (Fuel saved)(Unit cost of fuel) = (924,450 therm/yr)($1.04/therm) = **$961,430/yr**

The rate of entropy generation during this process is determined by applying the rate form of the entropy balance on an *extended system* that includes the regenerator and the immediate surroundings so that the boundary temperature is the surroundings temperature, which we take to be the cold water temperature of 18°C.:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} \;\rightarrow\; \dot{S}_{\text{gen}} = \dot{S}_{\text{out}} - \dot{S}_{\text{in}}$$

Disregarding entropy transfer associated with fuel flow, the only significant difference between the two cases is the reduction in the entropy transfer to water due to the reduction in heat transfer to water, and is determined to be

$$\dot{S}_{\text{gen, reduction}} = \dot{S}_{\text{out, reduction}} = \frac{\dot{Q}_{\text{out,reduction}}}{T_{\text{surr}}} = \frac{\dot{Q}_{\text{saved}}}{T_{\text{surr}}} = \frac{2536\,\text{kJ/s}}{18 + 273} = 8.715\,\text{kW/K}$$

$$S_{\text{gen, reduction}} = \dot{S}_{\text{gen, reduction}} \Delta t = (8.715\ \text{kJ/s.K})(8760 \times 3600\ \text{s/year}) = 2.75 \times 10^8\ \text{kJ/K (per year)}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed, reduction}} = T_0 S_{\text{gen, reduction}} = (291\,\text{K})(2.75 \times 10^8\ \text{kJ/K}) = \mathbf{8.00 \times 10^{10}\ kJ} \quad \text{(per year)}$$

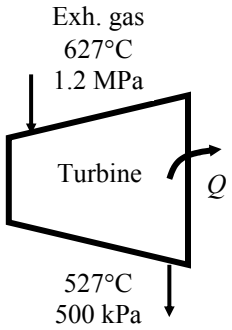
8-136 Exhaust gases are expanded in a turbine, which is not well-insulated. Tha actual and reversible power outputs, the exergy destroyed, and the second-law efficiency are to be determined.

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Assumptions **1** Steady operating conditions exist. **2** Potential energy change is negligible. **3** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is R = 0.287 kJ/kg.K and the specific heat of air at the average temperature of (627+527)/2 = 577ºC = 850 K is c_p = 1.11 kJ/kg.ºC (Table A-2).

Analysis (*a*) The enthalpy and entropy changes of air across the turbine are



$$\Delta h = c_p (T_2 - T_1) = (1.11\,\text{kJ/kg.°C})(527-627)\text{°C} = -111\,\text{kJ/kg}$$

$$\Delta s = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (1.11\,\text{kJ/kg.K})\ln\frac{(527+273)\,\text{K}}{(627+273)\,\text{K}} - (0.287\,\text{kJ/kg.K})\ln\frac{500\,\text{kPa}}{1200\,\text{kPa}}$$

$$= 0.1205\,\text{kJ/kg.K}$$

The actual and reversible power outputs from the turbine are

$$-\dot{W}_{\text{a,out}} = \dot{m}\Delta h + \dot{Q}_{\text{out}} = (2.5\,\text{kg/s})(\text{-}111\,\text{kJ/kg}) + 20\,\text{kW} = \mathbf{-257.5\,kW}$$

$$-\dot{W}_{\text{rev,out}} = \dot{m}(\Delta h - T_0\Delta s) = (2.5\,\text{kg/s})(111\,\text{kJ/kg}) - (25+273\,\text{K})(0.1205\,\text{kJ/kg.K}) = \mathbf{-367.3\,kW}$$

or

$$\dot{W}_{\text{a,out}} = \mathbf{257.5\,kW}$$

$$\dot{W}_{\text{rev,out}} = \mathbf{367.3\,kW}$$

(b) The exergy destroyed in the turbine is

$$\dot{X}_{\text{dest}} = \dot{W}_{\text{rev}} - \dot{W}_{\text{a}} = 367.3 - 257.5 = \mathbf{109.8\,kW}$$

(*c*) The second-law efficiency is

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{a}}}{\dot{W}_{\text{rev}}} = \frac{257.5\,\text{kW}}{367.3\,\text{kW}} = 0.701 = \mathbf{70.1\%}$$

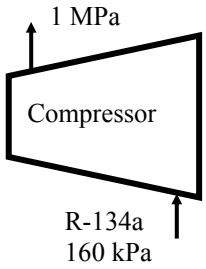
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8-137 Refrigerant-134a is compressed in an adiabatic compressor, whose second-law efficiency is given. The actual work input, the isentropic efficiency, and the exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The properties of the refrigerant at the inlet of the compressor are (Tables A-11 through A-13)

$$T_{\text{sat@160 kPa}} = -15.60°\text{C}$$
$$\left.\begin{aligned} P_1 &= 160\ \text{kPa} \\ T_1 &= (-15.60+3)°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 243.60\ \text{kJ/kg} \\ s_1 &= 0.95153\ \text{kJ/kg.K} \end{aligned}$$



The enthalpy at the exit for if the process was isentropic is

$$\left.\begin{aligned} P_2 &= 1\ \text{MPa} \\ s_2 &= s_1 = 0.95153\ \text{kJ/kg.K} \end{aligned}\right\} h_{2s} = 282.41\ \text{kJ/kg}$$

The expressions for actual and reversible works are

$$w_a = h_2 - h_1 = (h_2 - 243.60)\text{kJ/kg}$$

$$w_{rev} = h_2 - h_1 - T_0(s_2 - s_1) = (h_2 - 243.60)\text{kJ/kg} - (25+273\ \text{K})(s_2 - 0.95153)\text{kJ/kg.K}$$

Substituting these into the expression for the second-law efficiency

$$\eta_{II} = \frac{w_{rev}}{w_a} \longrightarrow 0.80 = \frac{h_2 - 243.60 - (298)(s_2 - 0.95153)}{h_2 - 243.60}$$

The exit pressure is given (1 MPa). We need one more property to fix the exit state. By a trial-error approach or using EES, we obtain the exit temperature to be 60ºC. The corresponding enthalpy and entropy values satisfying this equation are

$$h_2 = 293.36\ \text{kJ/kg}$$
$$s_2 = 0.98492\ \text{kJ/kg.K}$$

Then,

$$w_a = h_2 - h_1 = 293.36 - 243.60 = \mathbf{49.76 kJ/kg}$$

$$w_{rev} = h_2 - h_1 - T_0(s_2 - s_1) = (293.36 - 243.60)\text{kJ/kg} - (25+273\ \text{K})(0.98492 - 0.9515)\text{kJ/kg}\cdot\text{K} = 39.81\ \text{kJ/kg}$$

(b) The isentropic efficiency is determined from its definition

$$\eta_s = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{(282.41 - 243.60)\text{kJ/kg}}{(293.36 - 243.60)\text{kJ/kg}} = \mathbf{0.780}$$

(b) The exergy destroyed in the compressor is

$$x_{dest} = w_a - w_{rev} = 49.76 - 39.81 = \mathbf{9.95\ kJ/kg}$$

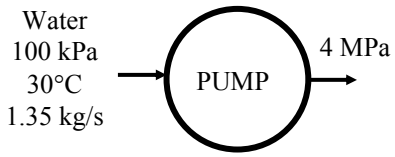
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8-138 The isentropic efficiency of a water pump is specified. The actual power output, the rate of frictional heating, the exergy destruction, and the second-law efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) Using saturated liquid properties at the given temperature for the inlet state (Table A-4)

$$\left.\begin{array}{l} T_1 = 30°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 125.82\ \text{kJ/kg} \\ s_1 = 0.4367\ \text{kJ/kg.K} \\ \nu_1 = 0.001004\ \text{m}^3/\text{kg} \end{array}$$



The power input if the process was isentropic is

$$\dot{W}_s = \dot{m}\nu_1(P_2 - P_1) = (1.35\ \text{kg/s})(0.001004\ \text{m}^3/\text{kg})(4000 - 100)\text{kPa} = 5.288\ \text{kW}$$

Given the isentropic efficiency, the actual power may be determined to be

$$\dot{W}_a = \frac{\dot{W}_s}{\eta_s} = \frac{5.288\ \text{kW}}{0.70} = \mathbf{7.554\ kW}$$

(*b*) The difference between the actual and isentropic works is the frictional heating in the pump

$$\dot{Q}_{\text{frictional}} = \dot{W}_a - \dot{W}_s = 7.554 - 5.288 = \mathbf{2.266\ kW}$$

(*c*) The enthalpy at the exit of the pump for the actual process can be determined from

$$\dot{W}_a = \dot{m}(h_2 - h_1) \longrightarrow 7.554\ \text{kW} = (1.35\ \text{kg/s})(h_2 - 125.82)\text{kJ/kg} \longrightarrow h_2 = 131.42\ \text{kJ/kg}$$

The entropy at the exit is

$$\left.\begin{array}{l} P_2 = 4\ \text{MPa} \\ h_2 = 131.42\ \text{kJ/kg} \end{array}\right\} s_2 = 0.4423\ \text{kJ/kg.K}$$

The reversible power and the exergy destruction are

$$\dot{W}_{\text{rev}} = \dot{m}\left[h_2 - h_1 - T_0(s_2 - s_1)\right]$$
$$= (1.35\ \text{kg/s})\left[(131.42 - 125.82)\text{kJ/kg} - (20 + 273\ \text{K})(0.4423 - 0.4367)\text{kJ/kg.K}\right] = 5.362\ \text{kW}$$

$$\dot{X}_{\text{dest}} = \dot{W}_a - \dot{W}_{\text{rev}} = 7.554 - 5.362 = \mathbf{2.193\ kW}$$

(*d*) The second-law efficiency is

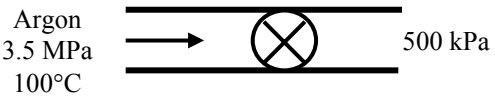
$$\eta_{\text{II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_a} = \frac{5.362\ \text{kW}}{7.554\ \text{kW}} = \mathbf{0.710}$$

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8-139 Argon gas is expanded adiabatically in an expansion valve. The exergy of argon at the inlet, the exergy destruction, and the second-law efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are zero. **3** Argon is an ideal gas with constant specific heats.

Properties The properties of argon gas are R = 0.2081 kJ/kg.K, c_p = 0.5203 kJ/kg.ºC (Table A-2).



Analysis (*a*) The exergy of the argon at the inlet is

$$\begin{aligned} x_1 &= h_1 - h_0 - T_0(s_1 - s_0) \\ &= c_p(T_1 - T_0) - T_0\left[c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0}\right] \\ &= (0.5203\,\text{kJ/kg.K})(100-25)°\text{C} - (298\,\text{K})\left[(0.5203\,\text{kJ/kg.K})\ln\frac{373\,\text{K}}{298\,\text{K}} - (0.2081\,\text{kJ/kg.K})\ln\frac{3500\,\text{kPa}}{100\,\text{kPa}}\right] \\ &= \mathbf{224.7\ kJ/kg} \end{aligned}$$

(*b*) Noting that the temperature remains constant in a throttling process of an ideal gas, the exergy destruction is determined from

$$\begin{aligned} x_{\text{dest}} &= T_0 s_{\text{gen}} \\ &= T_0(s_2 - s_1) \\ &= T_0\left(-R\ln\frac{P_1}{P_0}\right) = (298\,\text{K})\left[-(0.2081\,\text{kJ/kg.K})\ln\left(\frac{500\,\text{kPa}}{3500\,\text{kPa}}\right)\right] \\ &= \mathbf{120.7\ kJ/kg} \end{aligned}$$

(*c*) The second-law efficiency is

$$\eta_{\text{II}} = \frac{x_1 - x_{\text{dest}}}{x_1} = \frac{(224.7 - 120.7)\text{kJ/kg}}{224.7\,\text{kJ/kg}} = \mathbf{0.463}$$

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8-140 Heat is lost from the air flowing in a diffuser. The exit temperature, the rate of exergy destruction, and the second law efficiency are to be determined.

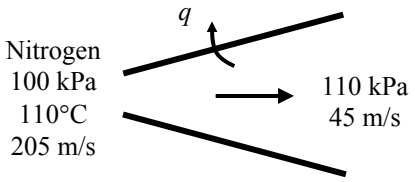
Assumptions **1** Steady operating conditions exist. **2** Potential energy change is negligible. **3** Nitrogen is an ideal gas with variable specific heats.

Properties The gas constant of nitrogen is R = 0.2968 kJ/kg.K.

Analysis (*a*) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure. At the inlet of the diffuser and at the dead state, we have

$$\left.\begin{array}{l} T_1 = 110°\text{C} = 383\ \text{K} \\ P_1 = 100\ \text{kPa} \end{array}\right\} \begin{array}{l} h_1 = 88.39\ \text{kJ/kg} \\ s_1 = 7.101\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_1 = 300\ \text{K} \\ P_1 = 100\ \text{kPa} \end{array}\right\} \begin{array}{l} h_0 = 1.93\ \text{kJ/kg} \\ s_0 = 6.846\ \text{kJ/kg}\cdot\text{K} \end{array}$$



An energy balance on the diffuser gives

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

$$88.39\ \text{kJ/kg} + \frac{(205\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = h_2 + \frac{(45\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) + 2.5\ \text{kJ/kg}$$

$$\longrightarrow h_2 = 105.9\ \text{kJ/kg}$$

The corresponding properties at the exit of the diffuser are

$$\left.\begin{array}{l} h_2 = 105.9\ \text{kJ/kg} \\ P_1 = 110\ \text{kPa} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{127°C} = 400\ \text{K} \\ s_2 = 7.117\ \text{kJ/kg}\cdot\text{K} \end{array}$$

(*b*) The mass flow rate of the nitrogen is determined to be

$$\dot{m} = \rho_2 A_2 V_2 = \frac{P_2}{RT_2} A_2 V_2 = \frac{110\ \text{kPa}}{(0.2968\ \text{kJ/kg.K})(400\ \text{K})}(0.04\ \text{m}^2)(45\ \text{m/s}) = 1.669\ \text{kg/s}$$

The exergy destruction in the nozzle is the exergy difference between the inlet and exit of the diffuser

$$\dot{X}_{\text{dest}} = \dot{m}\left[h_1 - h_2 + \frac{V_1^2 - V_2^2}{2} - T_0(s_1 - s_2)\right]$$

$$= (1.669\ \text{kg/s})\left[\begin{array}{l}(88.39 - 105.9)\text{kJ/kg} + \dfrac{(205\ \text{m/s})^2 - (45\ \text{m/s})^2}{2}\left(\dfrac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) \\ -(300\ \text{K})(7.101 - 7.117)\text{kJ/kg.K}\end{array}\right] = \mathbf{12.4\ kW}$$

(*c*) The second-law efficiency for this device may be defined as the exergy output divided by the exergy input:

$$\dot{X}_1 = \dot{m}\left[h_1 - h_0 + \frac{V_1^2}{2} - T_0(s_1 - s_0)\right]$$

$$= (1.669\ \text{kg/s})\left[(88.39 - 1.93)\ \text{kJ/kg} + \frac{(205\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) - (300\ \text{K})(7.101 - 6.846)\text{kJ/kg.K}\right]$$

$$= 51.96\ \text{kW}$$

$$\eta_{\text{II}} = \frac{\dot{X}_2}{\dot{X}_1} = 1 - \frac{\dot{X}_{\text{dest}}}{\dot{X}_1} = 1 - \frac{12.4\ \text{kW}}{51.96\ \text{kW}} = 0.761 = \mathbf{76.1\%}$$

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8-141 Using an incompressible substance as an example, it is to be demonstrated if closed system and flow exergies can be negative.

Analysis The availability of a closed system cannot be negative. However, the flow availability can be negative at low pressures. A closed system has zero availability at dead state, and positive availability at any other state since we can always produce work when there is a pressure or temperature differential.

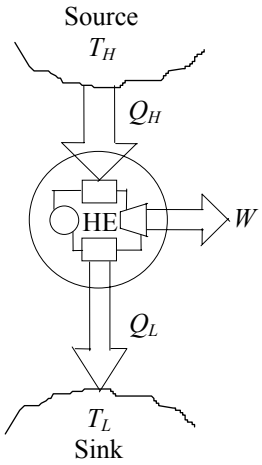
To see that the flow availability can be negative, consider an incompressible substance. The flow availability can be written as

$$\begin{aligned}\psi &= h - h_0 + T_0(s - s_0) \\ &= (u - u_0) + v(P - P_0) + T_0(s - s_0) \\ &= \xi + v(P - P_0)\end{aligned}$$

The closed system availability ξ is always positive or zero, and the flow availability can be negative when $P << P_0$.

8-142 A relation for the second-law efficiency of a heat engine operating between a heat source and a heat sink at specified temperatures is to be obtained.

Analysis The second-law efficiency is defined as the ratio of the availability recovered to availability supplied during a process. The work W produced is the availability recovered. The decrease in the availability of the heat supplied Q_H is the availability supplied or invested.



Therefore,

$$\eta_{II} = \frac{W}{\left(1 - \frac{T_0}{T_H}\right)Q_H - \left(1 - \frac{T_0}{T_L}\right)(Q_H - W)}$$

Note that the first term in the denominator is the availability of heat supplied to the heat engine whereas the second term is the availability of the heat rejected by the heat engine. The difference between the two is the availability consumed during the process.

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8-143 Writing energy and entropy balances, a relation for the reversible work is to be obtained for a closed system that exchanges heat with surroundings at T_0 in the amount of Q_0 as well as a heat reservoir at temperature T_R in the amount Q_R.

Assumptions Kinetic and potential changes are negligible.

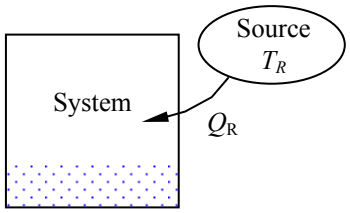
Analysis We take the direction of heat transfers to be to the system (heat input) and the direction of work transfer to be from the system (work output). The result obtained is still general since quantities wit opposite directions can be handled the same way by using negative signs. The energy and entropy balances for this stationary closed system can be expressed as

Energy balance: $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \rightarrow Q_0 + Q_R - W = U_2 - U_1 \longrightarrow W = U_1 - U_2 + Q_0 + Q_R$ (1)

Entropy balance: $S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}} \rightarrow S_{\text{gen}} = (S_2 - S_1) + \dfrac{-Q_R}{T_R} + \dfrac{-Q_0}{T_0}$ (2)

Solving for Q_0 from (2) and substituting in (1) yields

$$W = (U_1 - U_2) - T_0(S_1 - S_2) - Q_R\left(1 - \frac{T_0}{T_R}\right) - T_0 S_{\text{gen}}$$



The useful work relation for a closed system is obtained from

$$W_u = W - W_{\text{surr}}$$
$$= (U_1 - U_2) - T_0(S_1 - S_2) - Q_R\left(1 - \frac{T_0}{T_R}\right) - T_0 S_{\text{gen}} - P_0(V_2 - V_1)$$

Then the reversible work relation is obtained by substituting $S_{\text{gen}} = 0$,

$$W_{\text{rev}} = (U_1 - U_2) - T_0(S_1 - S_2) + P_0(V_1 - V_2) - Q_R\left(1 - \frac{T_0}{T_R}\right)$$

A positive result for W_{rev} indicates work output, and a negative result work input. Also, the Q_R is a positive quantity for heat transfer to the system, and a negative quantity for heat transfer from the system.

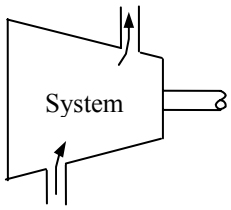
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8-144 Writing energy and entropy balances, a relation for the reversible work is to be obtained for a steady-flow system that exchanges heat with surroundings at T_0 at a rate of $\dot{Q}_0$ as well as a heat reservoir at temperature T_R in the amount $\dot{Q}_R$.

Analysis We take the direction of heat transfers to be to the system (heat input) and the direction of work transfer to be from the system (work output). The result obtained is still general since quantities wit opposite directions can be handled the same way by using negative signs. The energy and entropy balances for this stationary closed system can be expressed as

Energy balance: $\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$

$$\dot{Q}_0 + \dot{Q}_R - \dot{W} = \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) - \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right)$$



or $$\dot{W} = \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) - \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) + \dot{Q}_0 + \dot{Q}_R \quad (1)$$

Entropy balance:

$$\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = \Delta \dot{S}_{\text{system}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{S}_{\text{out}} - \dot{S}_{\text{in}}$$

$$\dot{S}_{\text{gen}} = \sum \dot{m}_e s_e - \sum \dot{m}_i s_i + \frac{-\dot{Q}_R}{T_R} + \frac{-Q_0}{T_0} \qquad (2)$$

Solving for $\dot{Q}_0$ from (2) and substituting in (1) yields

$$\dot{W} = \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right) - T_0 \dot{S}_{\text{gen}} - \dot{Q}_R \left(1 - \frac{T_0}{T_R}\right)$$

Then the reversible work relation is obtained by substituting $S_{gen} = 0$,

$$\dot{W}_{\text{rev}} = \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right) - \dot{Q}_R \left(1 - \frac{T_0}{T_R}\right)$$

A positive result for W_{rev} indicates work output, and a negative result work input. Also, the Q_R is a positive quantity for heat transfer to the system, and a negative quantity for heat transfer from the system.

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8-145 Writing energy and entropy balances, a relation for the reversible work is to be obtained for a uniform-flow system that exchanges heat with surroundings at T_0 in the amount of Q_0 as well as a heat reservoir at temperature T_R in the amount Q_R.

Assumptions Kinetic and potential changes are negligible.

Analysis We take the direction of heat transfers to be to the system (heat input) and the direction of work transfer to be from the system (work output). The result obtained is still general since quantities wit opposite directions can be handled the same way by using negative signs. The energy and entropy balances for this stationary closed system can be expressed as

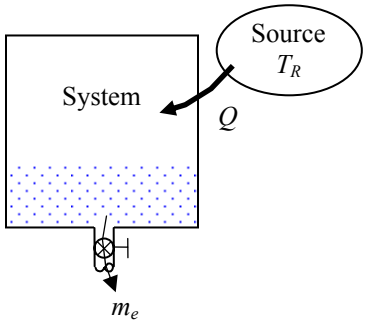
Energy balance: $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$

$$Q_0 + Q_R - W = \sum m_e\left(h_e + \frac{V_e^2}{2} + gz_e\right) - \sum m_i\left(h_i + \frac{V_i^2}{2} + gz_i\right) + (U_2 - U_1)_{cv}$$

or, $$W = \sum m_i\left(h_i + \frac{V_i^2}{2} + gz_i\right) - \sum m_e\left(h_e + \frac{V_e^2}{2} + gz_e\right) - (U_2 - U_1)_{cv} + Q_0 + Q_R \quad (1)$$

Entropy balance: $S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$

$$S_{\text{gen}} = (S_2 - S_1)_{cv} + \sum m_e s_e - \sum m_i s_i + \frac{-Q_R}{T_R} + \frac{-Q_0}{T_0} \quad (2)$$



Solving for Q_0 from (2) and substituting in (1) yields

$$W = \sum m_i\left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum m_e\left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right)$$
$$+\left[(U_1 - U_2) - T_0(S_1 - S_2)\right]_{cv} - T_0 S_{gen} - Q_R\left(1 - \frac{T_0}{T_R}\right)$$

The useful work relation for a closed system is obtained from

$$W_u = W - W_{\text{surr}} = \sum m_i\left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum m_e\left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right)$$
$$+\left[(U_1 - U_2) - T_0(S_1 - S_2)\right]_{cv} - T_0 S_{\text{gen}} - Q_R\left(1 - \frac{T_0}{T_R}\right) - P_0(\boldsymbol{V}_2 - \boldsymbol{V}_1)$$

Then the reversible work relation is obtained by substituting $S_{\text{gen}} = 0$,

$$W_{\text{rev}} = \sum m_i\left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum m_e\left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right)$$
$$+\left[(U_1 - U_2) - T_0(S_1 - S_2) + P_0(\boldsymbol{V}_1 - \boldsymbol{V}_2)\right]_{cv} - Q_R\left(1 - \frac{T_0}{T_R}\right)$$

A positive result for W_{rev} indicates work output, and a negative result work input. Also, the Q_R is a positive quantity for heat transfer to the system, and a negative quantity for heat transfer from the system.

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Fundamentals of Engineering (FE) Exam Problems

8-146 Heat is lost through a plane wall steadily at a rate of 800 W. If the inner and outer surface temperatures of the wall are 20°C and 5°C, respectively, and the environment temperature is 0°C, the rate of exergy destruction within the wall is

(a) 40 W (b) 17,500 W (c) 765 W (d) 32,800 W (e) 0 W

Answer (a) 40 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q=800 "W"
T1=20 "C"
T2=5 "C"
To=0 "C"
"Entropy balance S_in - S_out + S_gen= DS_system for the wall for steady operation gives"
Q/(T1+273)-Q/(T2+273)+S_gen=0   "W/K"
X_dest=(To+273)*S_gen "W"

"Some Wrong Solutions with Common Mistakes:"
Q/T1-Q/T2+Sgen1=0; W1_Xdest=(To+273)*Sgen1   "Using C instead of K in Sgen"
Sgen2=Q/((T1+T2)/2); W2_Xdest=(To+273)*Sgen2    "Using avegage temperature in C for Sgen"
Sgen3=Q/((T1+T2)/2+273); W3_Xdest=(To+273)*Sgen3    "Using avegage temperature in K"
W4_Xdest=To*S_gen   "Using C for To"
```

8-147 Liquid water enters an adiabatic piping system at 15°C at a rate of 3 kg/s. It is observed that the water temperature rises by 0.3°C in the pipe due to friction. If the environment temperature is also 15°C, the rate of exergy destruction in the pipe is

(a) 3.8 kW (b) 24 kW (c) 72 kW (d) 98 kW (e) 124 kW

Answer (a) 3.8 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=4.18 "kJ/kg.K"
m=3 "kg/s"
T1=15 "C"
T2=15.3 "C"
To=15 "C"
S_gen=m*Cp*ln((T2+273)/(T1+273)) "kW/K"
X_dest=(To+273)*S_gen "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Xdest=(To+273)*m*Cp*ln(T2/T1) "Using deg. C in Sgen"
W2_Xdest=To*m*Cp*ln(T2/T1) "Using deg. C in Sgen and To"
W3_Xdest=(To+273)*Cp*ln(T2/T1) "Not using mass flow rate with deg. C"
W4_Xdest=(To+273)*Cp*ln((T2+273)/(T1+273)) "Not using mass flow rate with K"
```

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8-148 A heat engine receives heat from a source at 1500 K at a rate of 600 kJ/s and rejects the waste heat to a sink at 300 K. If the power output of the engine is 400 kW, the second-law efficiency of this heat engine is

(a) 42% (b) 53% (c) 83% (d) 67% (e) 80%

Answer (c) 83%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Qin=600 "kJ/s"
W=400 "kW"
TL=300 "K"
TH=1500 "K"
Eta_rev=1-TL/TH
Eta_th=W/Qin
Eta_II=Eta_th/Eta_rev

"Some Wrong Solutions with Common Mistakes:"
W1_Eta_II=Eta_th1/Eta_rev; Eta_th1=1-W/Qin "Using wrong relation for thermal efficiency"
W2_Eta_II=Eta_th "Taking second-law efficiency to be thermal efficiency"
W3_Eta_II=Eta_rev "Taking second-law efficiency to be reversible efficiency"
W4_Eta_II=Eta_th*Eta_rev "Multiplying thermal and reversible efficiencies instead of dividing"
```

8-149 A water reservoir contains 100 tons of water at an average elevation of 60 m. The maximum amount of electric power that can be generated from this water is

(a) 8 kWh (b) 16 kWh (c) 1630 kWh (d) 16,300 kWh (e) 58,800 kWh

Answer (b) 16 kWh

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=100000 "kg"
h=60 "m"
g=9.81 "m/s^2"
"Maximum power is simply the potential energy change,"
W_max=m*g*h/1000 "kJ"
W_max_kWh=W_max/3600 "kWh"

"Some Wrong Solutions with Common Mistakes:"
W1_Wmax =m*g*h/3600 "Not using the conversion factor 1000"
W2_Wmax =m*g*h/1000 "Obtaining the result in kJ instead of kWh"
W3_Wmax =m*g*h*3.6/1000 "Using worng conversion factor"
W4_Wmax =m*h/3600"Not using g and the factor 1000 in calculations"
```

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8-150 A house is maintained at 21°C in winter by electric resistance heaters. If the outdoor temperature is 9°C, the second-law efficiency of the resistance heaters is

(a) 0% (b) 4.1% (c) 5.7% (d) 25% (e) 100%

Answer (b) 4.1%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=9+273 "K"
TH=21+273 "K"
To=TL
COP_rev=TH/(TH-TL)
COP=1
Eta_II=COP/COP_rev

"Some Wrong Solutions with Common Mistakes:"
W1_Eta_II=COP/COP_rev1; COP_rev1=TL/(TH-TL) "Using wrong relation for COP_rev"
W2_Eta_II=1-(TL-273)/(TH-273) "Taking second-law efficiency to be reversible thermal efficiency with C for temp"
W3_Eta_II=COP_rev "Taking second-law efficiency to be reversible COP"
W4_Eta_II=COP_rev2/COP; COP_rev2=(TL-273)/(TH-TL) "Using C in COP_rev relation instead of K, and reversing"
```

8-151 A 10-kg solid whose specific heat is 2.8 kJ/kg.°C is at a uniform temperature of -10°C. For an environment temperature of 25°C, the exergy content of this solid is

(a) Less than zero (b) 0 kJ (c) 22.3 kJ (d) 62.5 kJ (e) 980 kJ

Answer (d) 62.5 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=10 "kg"
Cp=2.8 "kJ/kg.K"
T1=-10+273 "K"
To=25+273 "K"
 "Exergy content of a fixed mass is x1=u1-uo-To*(s1-so)+Po*(v1-vo)"
 ex=m*(Cp*(T1-To)-To*Cp*ln(T1/To))

"Some Wrong Solutions with Common Mistakes:"
W1_ex=m*Cp*(To-T1) "Taking the energy content as the exergy content"
W2_ex=m*(Cp*(T1-To)+To*Cp*ln(T1/To)) "Using + for the second term instead of -"
W3_ex=Cp*(T1-To)-To*Cp*ln(T1/To) "Using exergy content per unit mass"
W4_ex=0 "Taking the exergy content to be zero"
```

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8-152 Keeping the limitations imposed by the second-law of thermodynamics in mind, choose the wrong statement below:

(a) A heat engine cannot have a thermal efficiency of 100%.
(b) For all reversible processes, the second-law efficiency is 100%.
(c) The second-law efficiency of a heat engine cannot be greater than its thermal efficiency.
(d) The second-law efficiency of a process is 100% if no entropy is generated during that process.
(e) The coefficient of performance of a refrigerator can be greater than 1.

Answer (c) The second-law efficiency of a heat engine cannot be greater than its thermal efficiency.

8-153 A furnace can supply heat steadily at a 1300 K at a rate of 500 kJ/s. The maximum amount of power that can be produced by using the heat supplied by this furnace in an environment at 300 K is

(a) 115 kW (b) 192 kW (c) 385 kW (d) 500 kW (e) 650 kW

Answer (c) 385 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q_in=500 "kJ/s"
TL=300 "K"
TH=1300 "K"
W_max=Q_in*(1-TL/TH) "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Wmax=W_max/2 "Taking half of Wmax"
W2_Wmax=Q_in/(1-TL/TH) "Dividing by efficiency instead of multiplying by it"
W3_Wmax =Q_in*TL/TH "Using wrong relation"
W4_Wmax=Q_in "Assuming entire heat input is converted to work"
```

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8-154 Air is throttled from 50°C and 800 kPa to a pressure of 200 kPa at a rate of 0.5 kg/s in an environment at 25°C. The change in kinetic energy is negligible, and no heat transfer occurs during the process. The power potential wasted during this process is

(a) 0 (b) 0.20 kW (c) 47 kW (d) 59 kW (e) 119 kW

Answer (d) 59 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cp=1.005 "kJ/kg.K"
m=0.5 "kg/s"
T1=50+273 "K"
P1=800 "kPa"
To=25 "C"
P2=200 "kPa"
"Temperature of an ideal gas remains constant during throttling since h=const and h=h(T)"
T2=T1
ds=Cp*ln(T2/T1)-R*ln(P2/P1)
X_dest=(To+273)*m*ds "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_dest=0 "Assuming no loss"
W2_dest=(To+273)*ds  "Not using mass flow rate"
W3_dest=To*m*ds  "Using C for To instead of K"
W4_dest=m*(P1-P2) "Using wrong relations"
```

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8-155 Steam enters a turbine steadily at 4 MPa and 400°C and exits at 0.2 MPa and 150°C in an environment at 25°C. The decrease in the exergy of the steam as it flows through the turbine is

(a) 58 kJ/kg (b) 445 kJ/kg (c) 458 kJ/kg (d) 518 kJ/kg (e) 597 kJ/kg

Answer (e) 597 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=4000 "kPa"
T1=400 "C"
P2=200 "kPa"
T2=150 "C"
To=25 "C"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
s2=ENTROPY(Steam_IAPWS,T=T2,P=P2)
"Exergy change of s fluid stream is Dx=h2-h1-To(s2-s1)"
-Dx=h2-h1-(To+273)*(s2-s1)

"Some Wrong Solutions with Common Mistakes:"
-W1_Dx=0 "Assuming no exergy destruction"
-W2_Dx=h2-h1 "Using enthalpy change"
-W3_Dx=h2-h1-To*(s2-s1) "Using C for To instead of K"
-W4_Dx=(h2+(T2+273)*s2)-(h1+(T1+273)*s1) "Using wrong relations for exergy"
```

8- 156 … 8- 160 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 9
GAS POWER CYCLES

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Actual and Ideal Cycles, Carnot cycle, Air-Standard Assumptions, Reciprocating Engines

9-1C It is less than the thermal efficiency of a Carnot cycle.

9-2C It represents the net work on both diagrams.

9-3C The air standard assumptions are: (1) the working fluid is air which behaves as an ideal gas, (2) all the processes are internally reversible, (3) the combustion process is replaced by the heat addition process, and (4) the exhaust process is replaced by the heat rejection process which returns the working fluid to its original state.

9-4C The cold air standard assumptions involves the additional assumption that air can be treated as an ideal gas with constant specific heats at room temperature.

9-5C The clearance volume is the minimum volume formed in the cylinder whereas the displacement volume is the volume displaced by the piston as the piston moves between the top dead center and the bottom dead center.

9-6C It is the ratio of the maximum to minimum volumes in the cylinder.

9-7C The MEP is the fictitious pressure which, if acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle.

9-8C Yes.

9-9C Assuming no accumulation of carbon deposits on the piston face, the compression ratio will remain the same (otherwise it will increase). The mean effective pressure, on the other hand, will decrease as a car gets older as a result of wear and tear.

9-10C The SI and CI engines differ from each other in the way combustion is initiated; by a spark in SI engines, and by compressing the air above the self-ignition temperature of the fuel in CI engines.

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9-11C Stroke is the distance between the TDC and the BDC, bore is the diameter of the cylinder, TDC is the position of the piston when it forms the smallest volume in the cylinder, and clearance volume is the minimum volume formed in the cylinder.

9-12E The maximum possible thermal efficiency of a gas power cycle with specified reservoirs is to be determined.

Analysis The maximum efficiency this cycle can have is

$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(40+460)\text{ R}}{(940+460)\text{ R}} = \mathbf{0.643}$$

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9-13 An air-standard cycle executed in a piston-cylinder system is composed of three specified processes. The cycle is to be sketcehed on the *P*-v and *T*-*s* diagrams and the back work ratio are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

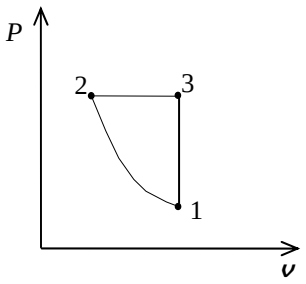
Properties The properties of air are given as R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4.

Analysis (a) The *P*-v and *T*-*s* diagrams of the cycle are shown in the figures.

(b) Process 1-2: Isentropic compression

$$w_{1-2,in} = mc_v(T_2 - T_1)$$

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1}$$

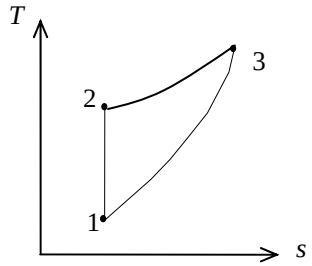


Process 2-3: Constant pressure heat addition

$$w_{2-3,out} = \int_2^3 P\,dv = P_2(V_3 - V_2) = mR(T_3 - T_2)$$

The back work ratio is

$$r_{bw} = \frac{w_{1-2,in}}{w_{2-3,out}} = \frac{mc_v(T_2 - T_1)}{mR(T_3 - T_2)}$$



Noting that

$$R = c_p - c_v \text{ and } k = \frac{c_p}{c_v} \text{ and thus, } c_v = \frac{R}{k-1}$$

From ideal gas relation,

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = \frac{v_1}{v_2} = r$$

Substituting these into back work relation,

$$r_{bw} = \frac{R}{k-1}\frac{1}{R}\frac{T_2}{T_2}\frac{(1 - T_1/T_2)}{(T_3/T_2 - 1)}$$

$$= \frac{1}{k-1}\frac{\left(1 - \frac{1}{r^{k-1}}\right)}{r-1} = \frac{1}{k-1}\frac{\left(1 - r^{1-k}\right)}{r-1}$$

$$= \frac{1}{1.4-1}\frac{\left(1 - 6^{-0.4}\right)}{6-1}$$

$$= \mathbf{0.256}$$

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9-14 The three processes of an air-standard cycle are described. The cycle is to be shown on the *P*-v and *T*-*s* diagrams, and the back work ratio and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air are given as R = 0.287 kJ/kg.K, c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4.

Analysis (a) The *P*-v and *T*-*s* diagrams of the cycle are shown in the figures.

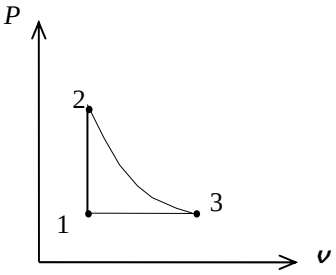
(b) The temperature at state 2 is

$$T_2 = T_1 \frac{P_2}{P_1} = (300\ \text{K})\frac{700\ \text{kPa}}{100\ \text{kPa}} = 2100\ \text{K}$$

$$T_3 = T_2 = 2100\ \text{K}$$

During process 1-3, we have

$$w_{3-1,in} = -\int_3^1 P\,dv = -P_1(v_1 - v_3) = -R(T_1 - T_3)$$
$$= -(0.287\ \text{kJ/kg}\cdot\text{K})(300 - 2100)\text{K} = 516.6\ \text{kJ/kg}$$

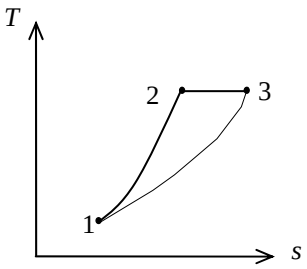


During process 2-3, we have

$$w_{2-3,out} = \int_2^3 P\,dv = \int_2^3 \frac{RT}{v}\,dv = RT\ln\frac{v_3}{v_2} = RT\ln\frac{7v_2}{v_2} = RT\ln 7$$
$$= (0.287\ \text{kJ/kg}\cdot\text{K})(2100)\text{K}\ln 7 = 1172.8\ \text{kJ/kg}$$

The back work ratio is then

$$r_{bw} = \frac{w_{3-1,in}}{w_{2-3,out}} = \frac{516.6\ \text{kJ/kg}}{1172.8\ \text{kJ/kg}} = \mathbf{0.440}$$



Heat input is determined from an energy balance on the cycle during process 1-3,

$$q_{1-3,in} - w_{2-3,out} = \Delta u_{1-3}$$
$$q_{1-3,in} - = \Delta u_{1-3} + w_{2-3,out}$$
$$= c_v(T_3 - T_1) + w_{2-3,out}$$
$$= (0.718\ \text{kJ/kg}\cdot\text{K})(2100 - 300) + 1172.8\ \text{kJ/kg}$$
$$= 2465\ \text{kJ/kg}$$

The net work output is

$$w_{net} = w_{2-3,out} - w_{3-1,in} = 1172.8 - 516.6 = 656.2\ \text{kJ/kg}$$

(c) The thermal efficiency is then

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{656.2\ \text{kJ}}{2465\ \text{kJ}} = 0.266 = \mathbf{26.6\%}$$

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9-15 The three processes of an ideal gas power cycle are described. The cycle is to be shown on the *P*-v and *T*-*s* diagrams, and the maximum temperature, expansion and compression works, and thermal efficiency are to be determined.

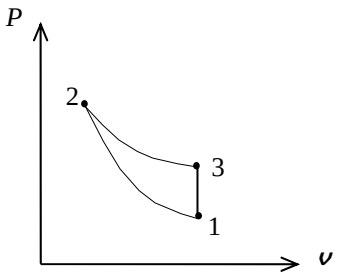
Assumptions **1** Kinetic and potential energy changes are negligible. **2** The ideal gas has constant specific heats.

Properties The properties of ideal gas are given as R = 0.3 kJ/kg.K, c_p = 0.9 kJ/kg.K, c_v = 0.6 kJ/kg·K, and k = 1.5.

Analysis (a) The *P*-v and *T*-*s* diagrams of the cycle are shown in the figures.

(b) The maximum temperature is determined from

$$T_{\max} = T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (27+273\,\text{K})(6)^{1.5-1} = \mathbf{734.8\,K}$$

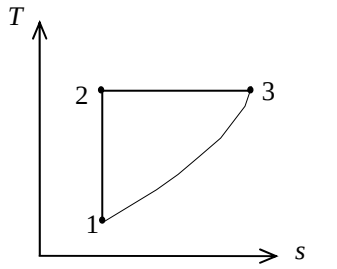


(c) An energy balance during process 2-3 gives

$$q_{2-3,in} - w_{2-3,out} = \Delta u_{2-3} = c_v(T_3 - T_2) = 0 \quad \text{since} \quad T_3 = T_2$$
$$q_{2-3,in} = w_{2-3,out}$$

Then, the work of compression is

$$q_{2-3,in} = w_{2-3,out} = \int_2^3 P\,dv = \int_2^3 \frac{RT}{v}\,dv = RT_2 \ln\frac{v_3}{v_2} = RT_2 \ln r$$
$$= (0.3\,\text{kJ/kg}\cdot\text{K})(734.8\,\text{K})\ln 6 = \mathbf{395.0\ kJ/kg}$$



(d) The work during isentropic compression is determined from an energy balance during process 1-2:

$$w_{1-2,in} = \Delta u_{1-2} = c_v(T_2 - T_1)$$
$$= (0.6\,\text{kJ/kg}\cdot\text{K})(734.8 - 300)$$
$$= \mathbf{260.9\ kJ/kg}$$

(e) Net work output is

$$w_{net} = w_{2-3,out} - w_{1-2,in} = 395.0 - 260.9 = 134.1\,\text{kJ/kg}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{134.1\,\text{kJ}}{395.0\,\text{kJ}} = 0.339 = \mathbf{33.9\%}$$

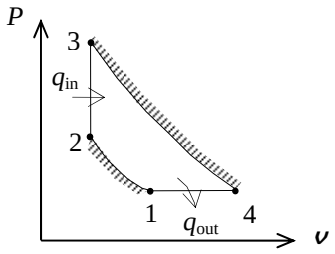
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9-16 The four processes of an air-standard cycle are described. The cycle is to be shown on *P*-ν and *T*-*s* diagrams, and the net work output and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-17.

Analysis (*b*) The properties of air at various states are

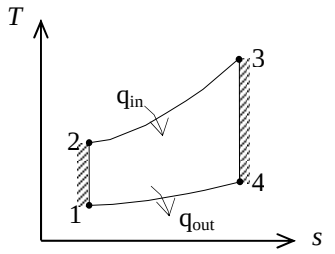


$$T_1 = 295\,\text{K} \longrightarrow \begin{matrix} h_1 = 295.17\ \text{kJ/kg} \\ P_{r_1} = 1.3068 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{600\ \text{kPa}}{100\ \text{kPa}}(1.3068) = 7.841 \longrightarrow \begin{matrix} u_2 = 352.29\ \text{kJ/kg} \\ T_2 = 490.3\ \text{K} \end{matrix}$$

$$T_3 = 1500\ \text{K} \longrightarrow \begin{matrix} u_3 = 1205.41\,\text{kJ/kg} \\ P_{r_3} = 601.9 \end{matrix}$$

$$\frac{P_3 \nu_3}{T_3} = \frac{P_2 \nu_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2} P_2 = \frac{1500\ \text{K}}{490.3\ \text{K}}(600\ \text{kPa}) = 1835.6\ \text{kPa}$$



$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \frac{100\ \text{kPa}}{1835.6\ \text{kPa}}(601.9) = 32.79 \longrightarrow h_4 = 739.71\,\text{kJ/kg}$$

From energy balances,

$$q_{in} = u_3 - u_2 = 1205.41 - 352.29 = 853.1\ \text{kJ/kg}$$

$$q_{out} = h_4 - h_1 = 739.71 - 295.17 = 444.5\ \text{kJ/kg}$$

$$w_{net,out} = q_{in} - q_{out} = 853.1 - 444.5 = \mathbf{408.6\ kJ/kg}$$

(*c*) Then the thermal efficiency becomes

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{408.6\ \text{kJ/kg}}{853.1\,\text{kJ/kg}} = 0.479 = \mathbf{47.9\%}$$

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9-17 Problem 9-16 is reconsidered. The effect of the maximum temperature of the cycle on the net work output and thermal efficiency is to be investigated. Also, *T-s* and *P-v* diagrams for the cycle are to be plotted.

Analysis Using EES, the problem is solved as follows:

```
"Input Data"
T[1]=295 [K]
P[1]=100 [kPa]
P[2] = 600 [kPa]
T[3]=1500 [K]
P[4] = 100 [kPa]

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
"Conservation of energy for process 1 to 2"
q_12 -w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

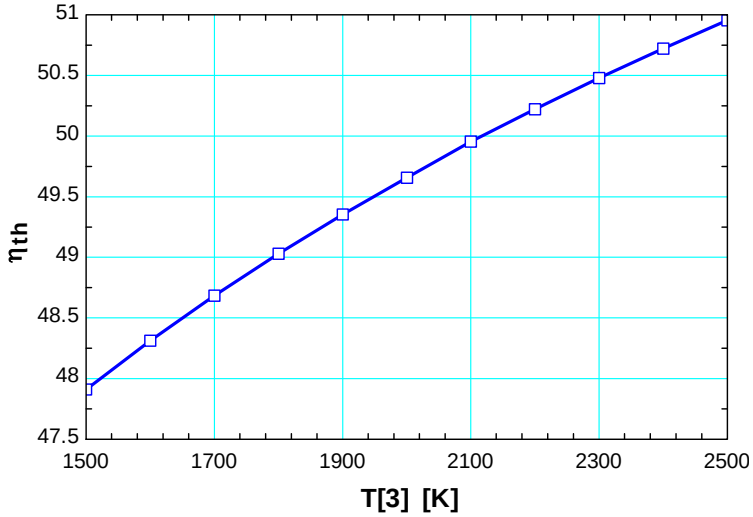
"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=R*T[3]
v[3]=v[2]
"Conservation of energy for process 2 to 3"
q_23 -w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=entropy(air,T=T[4],P=P[4])
s[4]=s[3]
P[4]*v[4]/T[4]=P[3]*v[3]/T[3]
{P[4]*v[4]=0.287*T[4]}
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])

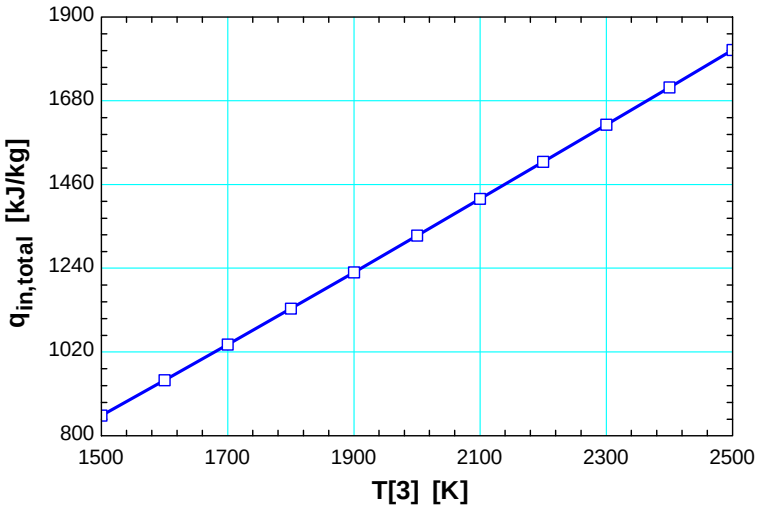
"Process 4-1 is constant pressure heat rejection"
{P[4]*v[4]/T[4]=P[1]*v[1]/T[1]}
"Conservation of energy for process 4 to 1"
q_41 -w_41 = DELTAu_41
w_41 =P[1]*(v[1]-v[4]) "constant pressure process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23

w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
```

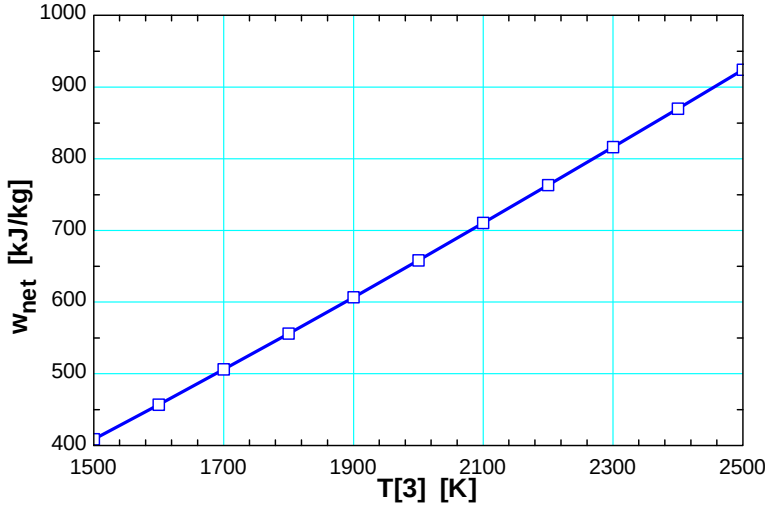
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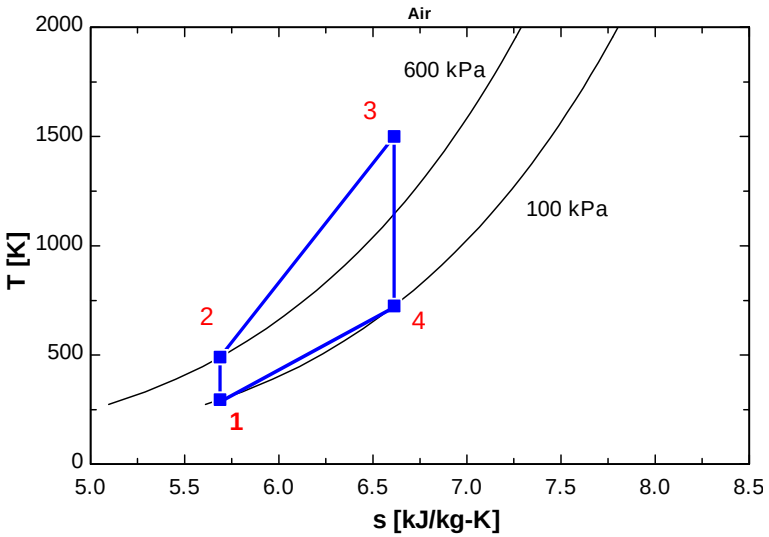
T_3 [K]	η_{th}	$q_{in,total}$ [kJ/kg]	W_{net} [kJ/kg]
1500	47.91	852.9	408.6
1600	48.31	945.7	456.9
1700	48.68	1040	506.1
1800	49.03	1134	556
1900	49.35	1229	606.7
2000	49.66	1325	658.1
2100	49.95	1422	710.5
2200	50.22	1519	763
2300	50.48	1617	816.1
2400	50.72	1715	869.8
2500	50.95	1813	924

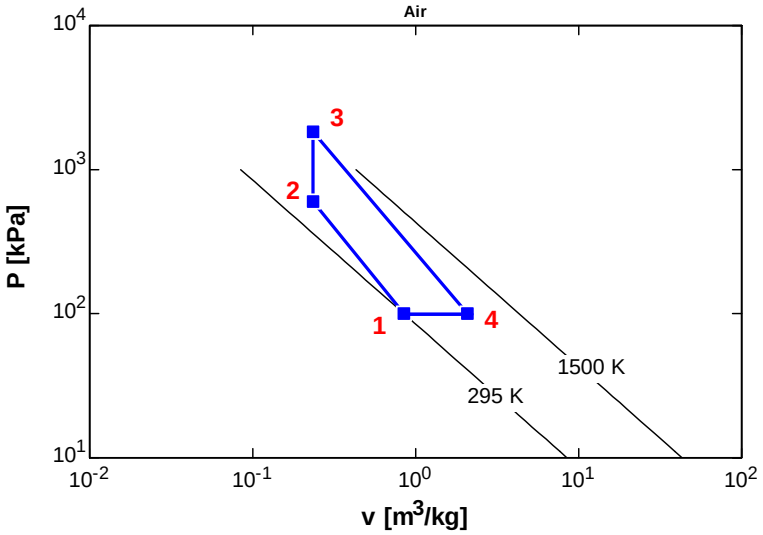




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9-18 The three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the heat rejected and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

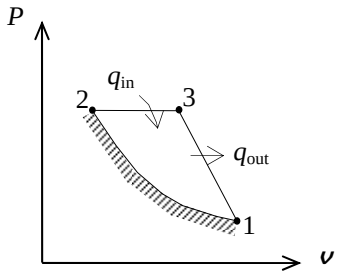
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*b*) The temperature at state 2 and the heat input are

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{1000\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = 579.2\text{ K}$$

$$Q_{\text{in}} = m(h_3 - h_2) = mc_p(T_3 - T_2)$$

$$2.76\text{ kJ} = (0.004\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(T_3 - 579.2) \longrightarrow T_3 = 1266\text{ K}$$

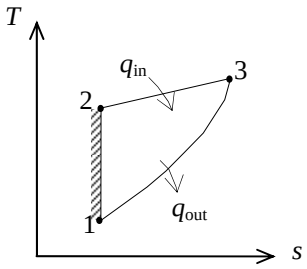


Process 3-1 is a straight line on the *P-v* diagram, thus the w_{31} is simply the area under the process curve,

$$w_{31} = \text{area} = \frac{P_3 + P_1}{2}(v_1 - v_3) = \frac{P_3 + P_1}{2}\left(\frac{RT_1}{P_1} - \frac{RT_3}{P_3}\right)$$

$$= \left(\frac{1000 + 100\text{ kPa}}{2}\right)\left(\frac{300\text{ K}}{100\text{ kPa}} - \frac{1266\text{ K}}{1000\text{ kPa}}\right)(0.287\text{ kJ/kg}\cdot\text{K})$$

$$= 273.7\text{ kJ/kg}$$



Energy balance for process 3-1 gives

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \longrightarrow -Q_{31,\text{out}} - W_{31,\text{out}} = m(u_1 - u_3)$$

$$Q_{31,\text{out}} = -mw_{31,\text{out}} - mc_v(T_1 - T_3) = -m\left[w_{31,\text{out}} + c_v(T_1 - T_3)\right]$$

$$= -(0.004\text{ kg})\left[273.7 + (0.718\text{ kJ/kg}\cdot\text{K})(300 - 1266)\text{K}\right] = \mathbf{1.679\text{ kJ}}$$

(*c*) The thermal efficiency is then

$$\eta_{\text{th}} = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} = 1 - \frac{1.679\text{ kJ}}{2.76\text{ kJ}} = \mathbf{39.2\%}$$

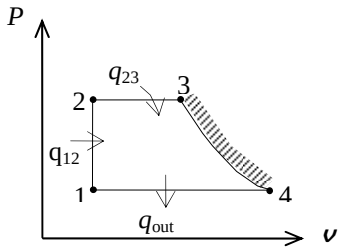
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9-19E The four processes of an air-standard cycle are described. The cycle is to be shown on *P*-v and *T*-*s* diagrams, and the total heat input and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-17E.

Analysis (*b*) The properties of air at various states are



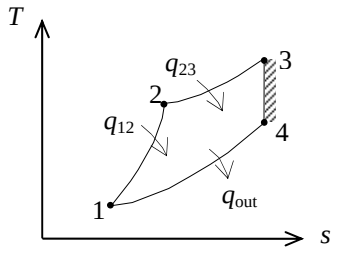
$$T_1 = 540\ \text{R} \longrightarrow u_1 = 92.04\ \text{Btu/lbm}, \quad h_1 = 129.06\ \text{Btu/lbm}$$

$$q_{\text{in},12} = u_2 - u_1 \longrightarrow \begin{matrix} u_2 = u_1 + q_{\text{in},12} = 92.04 + 300 = 392.04\ \text{Btu/lbm} \\ T_2 = 2116\text{R},\ h_2 = 537.1\ \text{Btu/lbm} \end{matrix}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{2116\ \text{R}}{540\ \text{R}}(14.7\ \text{psia}) = 57.6\ \text{psia}$$

$$T_3 = 3200\ \text{R} \longrightarrow \begin{matrix} h_3 = 849.48\ \text{Btu/lbm} \\ P_{r_3} = 1242 \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \frac{14.7\ \text{psia}}{57.6\ \text{psia}}(1242) = 317.0 \longrightarrow h_4 = 593.22\ \text{Btu/lbm}$$



From energy balance,

$$q_{23,\text{in}} = h_3 - h_2 = 849.48 - 537.1 = 312.38\ \text{Btu/lbm}$$

$$q_{\text{in}} = q_{12,\text{in}} + q_{23,\text{in}} = 300 + 312.38 = \mathbf{612.38\ Btu/lbm}$$

$$q_{\text{out}} = h_4 - h_1 = 593.22 - 129.06 = 464.16\ \text{Btu/lbm}$$

(*c*) Then the thermal efficiency becomes

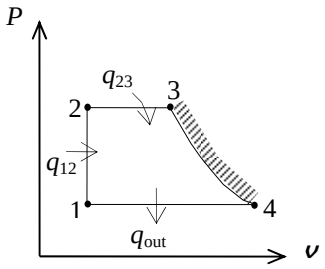
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{464.16\text{Btu/lbm}}{612.38\text{Btu/lbm}} = \mathbf{24.2\%}$$

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9-20E The four processes of an air-standard cycle are described. The cycle is to be shown on *P*-v and *T*-*s* diagrams, and the total heat input and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

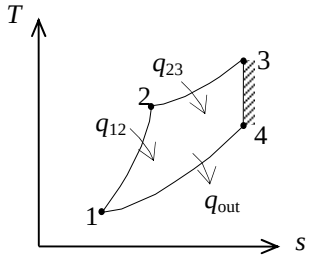
Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm.R, c_v = 0.171 Btu/lbm.R, and k = 1.4 (Table A-2E).



Analysis (*b*) The temperature at state 2 and the heat input are

$$q_{\text{in},12} = u_2 - u_1 = c_v(T_2 - T_1)$$
$$300 \text{ Btu/lbm} = (0.171 \text{ Btu/lbm.R})(T_2 - 540)\text{R}$$
$$T_2 = 2294 \text{ R}$$
$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{2294 \text{ R}}{540 \text{ R}}(14.7 \text{ psia}) = 62.46 \text{ psia}$$
$$q_{\text{in},23} = h_3 - h_2 = c_P(T_3 - T_2) = (0.24 \text{ Btu/lbm} \cdot \text{R})(3200 - 2294)\text{R} = 217.4 \text{ Btu/lbm}$$

Process 3-4 is isentropic:



$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (3200 \text{ R})\left(\frac{14.7 \text{ psia}}{62.46 \text{ psia}}\right)^{0.4/1.4} = 2117 \text{ R}$$
$$q_{\text{in}} = q_{\text{in},12} + q_{\text{in},23} = 300 + 217.4 = \mathbf{517.4 \text{ Btu/lbm}}$$
$$q_{\text{out}} = h_4 - h_1 = c_p(T_4 - T_1) = (0.240 \text{ Btu/lbm.R})(2117 - 540) = 378.5 \text{ Btu/lbm}$$

(c) The thermal efficiency is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{378.5 \text{ Btu/lbm}}{517.4 \text{ Btu/lbm}} = \mathbf{26.8\%}$$

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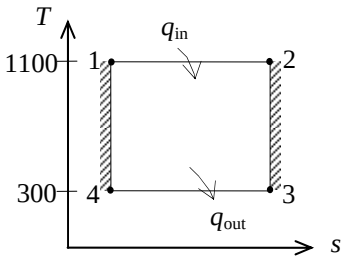
9-21 A Carnot cycle with the specified temperature limits is considered. The net work output per cycle is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg.K, and k = 1.4 (Table A-2).

Analysis The minimum pressure in the cycle is P_3 and the maximum pressure is P_1. Then,

$$\frac{T_2}{T_3}=\left(\frac{P_2}{P_3}\right)^{(k-1)/k}$$



or

$$P_2 = P_3\left(\frac{T_2}{T_3}\right)^{k/(k-1)} = (20\text{ kPa})\left(\frac{1100\text{ K}}{300\text{ K}}\right)^{1.4/0.4} = 1888\text{ kPa}$$

The heat input is determined from

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -(0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{1888\text{ kPa}}{3000\text{ kPa}} = 0.1329\text{ kJ/kg}\cdot\text{K}$$

$$Q_{\text{in}} = mT_H(s_2 - s_1) = (0.6\text{ kg})(1100\text{ K})(0.1329\text{ kJ/kg}\cdot\text{K}) = 87.73\text{ kJ}$$

Then,

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{1100\text{ K}} = 0.7273 = 72.7\%$$

$$W_{\text{net,out}} = \eta_{\text{th}} Q_{\text{in}} = (0.7273)(87.73\text{ kJ}) = \mathbf{63.8\ kJ}$$

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9-22 A Carnot cycle executed in a closed system with air as the working fluid is considered. The minimum pressure in the cycle, the heat rejection from the cycle, the thermal efficiency of the cycle, and the second-law efficiency of an actual cycle operating between the same temperature limits are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperatures are R = 0.287 kJ/kg.K and k = 1.4 (Table A-2).

Analysis (*a*) The minimum temperature is determined from

$$w_{\text{net}} = (s_2 - s_1)(T_H - T_L) \longrightarrow 100 \text{ kJ/kg} = (0.25 \text{ kJ/kg} \cdot \text{K})(750 - T_L)\text{K} \longrightarrow T_L = 350 \text{ K}$$

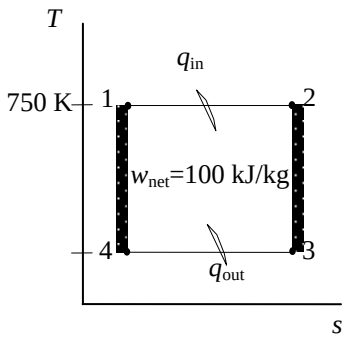
The pressure at state 4 is determined from

$$\frac{T_1}{T_4} = \left(\frac{P_1}{P_4}\right)^{(k-1)/k}$$

or

$$P_1 = P_4\left(\frac{T_1}{T_4}\right)^{k/(k-1)}$$

$$800 \text{ kPa} = P_4\left(\frac{750 \text{ K}}{350 \text{ K}}\right)^{1.4/0.4} \longrightarrow P_4 = 110.1 \text{ kPa}$$



The minimum pressure in the cycle is determined from

$$\Delta s_{12} = -\Delta s_{34} = c_p \ln\frac{T_4}{T_3}^{\nearrow 0} - R\ln\frac{P_4}{P_3}$$

$$-0.25 \text{ kJ/kg} \cdot \text{K} = -(0.287 \text{ kJ/kg} \cdot \text{K})\ln\frac{110.1 \text{ kPa}}{P_3} \longrightarrow P_3 = \mathbf{46.1 \text{ kPa}}$$

(*b*) The heat rejection from the cycle is

$$q_{\text{out}} = T_L \Delta s_{12} = (350 \text{ K})(0.25 \text{ kJ/kg.K}) = \mathbf{87.5 \text{ kJ/kg}}$$

(*c*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{350 \text{ K}}{750 \text{ K}} = \mathbf{0.533}$$

(*d*) The power output for the Carnot cycle is

$$\dot{W}_{\text{Carnot}} = \dot{m} w_{\text{net}} = (90 \text{ kg/s})(100 \text{ kJ/kg}) = 9000 \text{ kW}$$

Then, the second-law efficiency of the actual cycle becomes

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{actual}}}{\dot{W}_{\text{Carnot}}} = \frac{5200 \text{ kW}}{9000 \text{ kW}} = \mathbf{0.578}$$

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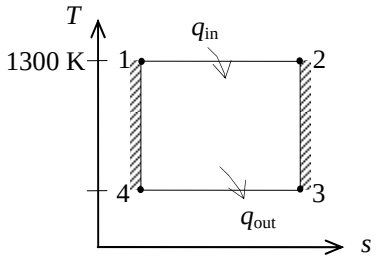
9-23 An ideal gas Carnot cycle with air as the working fluid is considered. The maximum temperature of the low-temperature energy reservoir, the cycle's thermal efficiency, and the amount of heat that must be supplied per cycle are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The temperature of the low-temperature reservoir can be found by applying the isentropic expansion process relation

$$T_1 = T_2\left(\frac{v_2}{v_1}\right)^{k-1} = (1027+273\,\text{K})\left(\frac{1}{12}\right)^{1.4-1} = \mathbf{481.1\,K}$$



Since the Carnot engine is completely reversible, its efficiency is

$$\eta_{\text{th,Carnot}} = 1-\frac{T_L}{T_H} = 1-\frac{481.1\,\text{K}}{(1027+273)\,\text{K}} = \mathbf{0.630}$$

The work output per cycle is

$$W_{\text{net}} = \frac{\dot{W}_{\text{net}}}{\dot{n}} = \frac{500\,\text{kJ/s}}{1500\,\text{cycle/min}}\left(\frac{60\,\text{s}}{1\,\text{min}}\right) = 20\,\text{kJ/cycle}$$

According to the definition of the cycle efficiency,

$$\eta_{\text{th,Carnot}} = \frac{W_{\text{net}}}{Q_{\text{in}}} \longrightarrow Q_{\text{in}} = \frac{W_{\text{net}}}{\eta_{\text{th,Carnot}}} = \frac{20\,\text{kJ/cycle}}{0.63} = \mathbf{31.75\ kJ/cycle}$$

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9-24 An air-standard cycle executed in a piston-cylinder system is composed of three specified processes. The cycle is to be sketcehed on the *P*-v and *T*-*s* diagrams; the heat and work interactions and the thermal efficiency of the cycle are to be determined; and an expression for thermal efficiency as functions of compression ratio and specific heat ratio is to be obtained.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

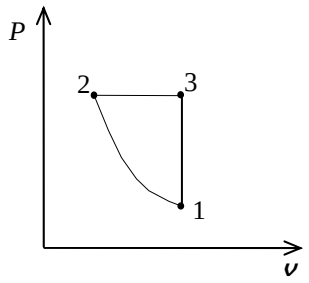
Properties The properties of air are given as R = 0.3 kJ/kg·K and c_v = 0.3 kJ/kg·K.

Analysis (a) The *P*-v and *T*-*s* diagrams of the cycle are shown in the figures.

(b) Noting that

$$c_p = c_v + R = 0.7 + 0.3 = 1.0 \text{ kJ/kg} \cdot \text{K}$$

$$k = \frac{c_p}{c_v} = \frac{1.0}{0.7} = 1.429$$

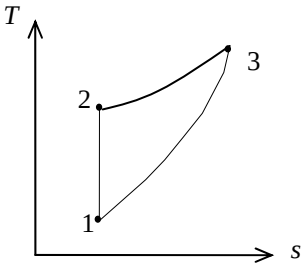


Process 1-2: Isentropic compression

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (293 \text{ K})(5)^{0.429} = 584.4 \text{ K}$$

$$w_{1-2,\text{in}} = c_v(T_2 - T_1) = (0.7 \text{ kJ/kg} \cdot \text{K})(584.4 - 293) \text{ K} = \mathbf{204.0 \text{ kJ/kg}}$$

$$q_{1-2} = \mathbf{0}$$



From ideal gas relation,

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = \frac{v_1}{v_2} = r \longrightarrow T_3 = (584.4)(5) = 2922$$

Process 2-3: Constant pressure heat addition

$$w_{2-3,\text{out}} = \int_2^3 P dv = P_2(v_3 - v_2) = R(T_3 - T_2)$$
$$= (0.3 \text{ kJ/kg} \cdot \text{K})(2922 - 584.4) \text{ K} = \mathbf{701.3 \text{ kJ/kg}}$$

$$q_{2-3,\text{in}} = w_{2-3,out} + \Delta u_{2-3} = \Delta h_{2-3}$$
$$= c_p(T_3 - T_2) = (1 \text{ kJ/kg} \cdot \text{K})(2922 - 584.4) \text{ K} = \mathbf{2338 \text{ kJ/kg}}$$

Process 3-1: Constant volume heat rejection

$$q_{3-1,\text{out}} = \Delta u_{1-3} = c_v(T_3 - T_1) = (0.7 \text{ kJ/kg} \cdot \text{K})(2922 - 293) \text{ K} = \mathbf{1840.3 \text{ kJ/kg}}$$

$$w_{3-1} = \mathbf{0}$$

(c) Net work is

$$w_{\text{net}} = w_{2-3,\text{out}} - w_{1-2,\text{in}} = 701.3 - 204.0 = 497.3 \text{ kJ/kg} \cdot \text{K}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{497.3 \text{ kJ}}{2338 \text{ kJ}} = 0.213 = \mathbf{21.3\%}$$

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(d) The expression for the cycle thermal efficiency is obtained as follows:

$$\begin{aligned}
\eta_{\text{th}} &= \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{w_{2-3,\text{out}} - w_{1-2,\text{in}}}{q_{\text{in}}} \\
&= \frac{R(T_3 - T_2) - c_v(T_2 - T_1)}{c_p(T_3 - T_2)} \\
&= \frac{R}{c_p} - \frac{c_v(T_1 r^{k-1} - T_1)}{c_p(rT_1 r^{k-1} - T_1 r^{k-1})} \\
&= \frac{R}{c_p} - \frac{c_v T_1 r^{k-1}\left(1 - \dfrac{T_1}{T_1 r^{k-1}}\right)}{c_p T_1 r^{k-1}(r-1)} \\
&= \frac{R}{c_p} - \frac{1}{k(r-1)}\left(1 - \frac{T_1}{T_1 r^{k-1}}\right) \\
&= \frac{R}{c_p} - \frac{1}{k(r-1)}\left(1 - \frac{1}{r^{k-1}}\right) \\
&= \left(1 - \frac{1}{k}\right) - \frac{1}{k(r-1)}\left(1 - \frac{1}{r^{k-1}}\right)
\end{aligned}$$

since

$$\frac{R}{c_p} = \frac{c_p - c_v}{c_p} = 1 - \frac{c_v}{c_p} = 1 - \frac{1}{k}$$

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Otto Cycle

9-25C For actual four-stroke engines, the rpm is twice the number of thermodynamic cycles; for two-stroke engines, it is equal to the number of thermodynamic cycles.

9-26C The ideal Otto cycle involves external irreversibilities, and thus it has a lower thermal efficiency.

9-27C The four processes that make up the Otto cycle are (1) isentropic compression, (2) v = constant heat addition, (3) isentropic expansion, and (4) v = constant heat rejection.

9-28C They are analyzed as closed system processes because no mass crosses the system boundaries during any of the processes.

9-29C It increases with both of them.

9-30C Because high compression ratios cause engine knock.

9-31C The thermal efficiency will be the highest for argon because it has the highest specific heat ratio, $k = 1.667$.

9-32C The fuel is injected into the cylinder in both engines, but it is ignited with a spark plug in gasoline engines.

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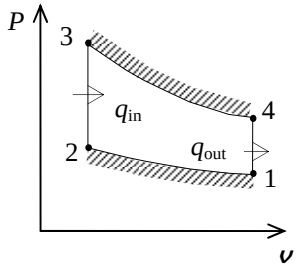
9-33 An ideal Otto cycle is considered. The thermal efficiency and the rate of heat input are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The definition of cycle thermal efficiency reduces to

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{10.5^{1.4-1}} = 0.6096 = \mathbf{61.0\%}$$



The rate of heat addition is then

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th}}} = \frac{90\ \text{kW}}{0.6096} = \mathbf{148\ kW}$$

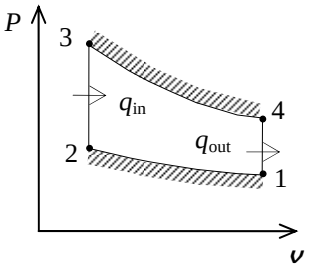
9-34 An ideal Otto cycle is considered. The thermal efficiency and the rate of heat input are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The definition of cycle thermal efficiency reduces to

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{8.5^{1.4-1}} = 0.5752 = \mathbf{57.5\%}$$



The rate of heat addition is then

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th}}} = \frac{90\ \text{kW}}{0.5752} = \mathbf{157\ kW}$$

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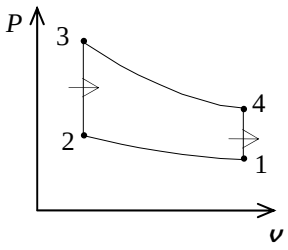
9-35 The two isentropic processes in an Otto cycle are replaced with polytropic processes. The heat added to and rejected from this cycle, and the cycle's thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The temperature at the end of the compression is

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = T_1 r^{n-1} = (288\text{ K})(8)^{1.3-1} = 537.4\text{ K}$$



And the temperature at the end of the expansion is

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = T_3\left(\frac{1}{r}\right)^{n-1} = (1473\text{ K})\left(\frac{1}{8}\right)^{1.3-1} = 789.4\text{ K}$$

The integral of the work expression for the polytropic compression gives

$$w_{1-2} = \frac{RT_1}{n-1}\left[\left(\frac{v_1}{v_2}\right)^{n-1} - 1\right] = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(288\text{ K})}{1.3-1}(8^{1.3-1} - 1) = 238.6\text{ kJ/kg}$$

Similarly, the work produced during the expansion is

$$w_{3-4} = -\frac{RT_3}{n-1}\left[\left(\frac{v_3}{v_4}\right)^{n-1} - 1\right] = -\frac{(0.287\text{ kJ/kg}\cdot\text{K})(1473\text{ K})}{1.3-1}\left[\left(\frac{1}{8}\right)^{1.3-1} - 1\right] = 654.0\text{ kJ/kg}$$

Application of the first law to each of the four processes gives

$$q_{1-2} = w_{1-2} - c_v(T_2 - T_1) = 238.6\text{ kJ/kg} - (0.718\text{ kJ/kg}\cdot\text{K})(537.4 - 288)\text{K} = 59.53\text{ kJ/kg}$$

$$q_{2-3} = c_v(T_3 - T_2) = (0.718\text{ kJ/kg}\cdot\text{K})(1473 - 537.4)\text{K} = 671.8\text{ kJ/kg}$$

$$q_{3-4} = w_{3-4} - c_v(T_3 - T_4) = 654.0\text{ kJ/kg} - (0.718\text{ kJ/kg}\cdot\text{K})(1473 - 789.4)\text{K} = 163.2\text{ kJ/kg}$$

$$q_{4-1} = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(789.4 - 288)\text{K} = 360.0\text{ kJ/kg}$$

The head added and rejected from the cycle are

$$q_{\text{in}} = q_{2-3} + q_{3-4} = 671.8 + 163.2 = \mathbf{835.0\text{ kJ/kg}}$$
$$q_{\text{out}} = q_{1-2} + q_{4-1} = 59.53 + 360.0 = \mathbf{419.5\text{ kJ/kg}}$$

The thermal efficiency of this cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{419.5}{835.0} = \mathbf{0.498}$$

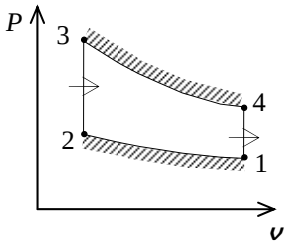
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9-36 An ideal Otto cycle is considered. The heat added to and rejected from this cycle, and the cycle's thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The temperature at the end of the compression is



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (288\,\text{K})(8)^{1.4-1} = 661.7\,\text{K}$$

and the temperature at the end of the expansion is

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1} = (1473\,\text{K})\left(\frac{1}{8}\right)^{1.4-1} = 641.2\,\text{K}$$

Application of the first law to the heat addition process gives

$$q_{\text{in}} = c_v(T_3 - T_2) = (0.718\,\text{kJ/kg}\cdot\text{K})(1473 - 661.7)\text{K} = \mathbf{582.5\ kJ/kg}$$

Similarly, the heat rejected is

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.718\,\text{kJ/kg}\cdot\text{K})(641.2 - 288)\text{K} = \mathbf{253.6\ kJ/kg}$$

The thermal efficiency of this cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{253.6}{582.5} = \mathbf{0.565}$$

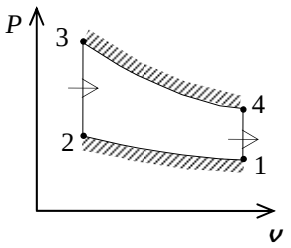
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9-37E A six-cylinder, four-stroke, spark-ignition engine operating on the ideal Otto cycle is considered. The power produced by the engine is to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis From the data specified in the problem statement,



$$r = \frac{v_1}{v_2} = \frac{v_1}{0.14 v_1} = 7.143$$

Since the compression and expansion processes are isentropic,

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (525\,\text{R})(7.143)^{1.4-1} = 1153\,\text{R}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1} = (2060\,\text{R})\left(\frac{1}{7.143}\right)^{1.4-1} = 938.2\,\text{R}$$

Application of the first law to the compression and expansion processes gives

$$\begin{aligned} w_{\text{net}} &= c_v(T_3 - T_4) - c_v(T_2 - T_1) \\ &= (0.171\,\text{Btu/lbm}\cdot\text{R})(2060 - 938.2)\text{R} - (0.171\,\text{Btu/lbm}\cdot\text{R})(1153 - 525)\text{R} \\ &= 84.44\,\text{Btu/lbm} \end{aligned}$$

When each cylinder is charged with the air-fuel mixture,

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(525\,\text{R})}{14\,\text{psia}} = 13.89\,\text{ft}^3/\text{lbm}$$

The total air mass taken by all 6 cylinders when they are charged is

$$m = N_{\text{cyl}}\frac{\Delta V}{v_1} = N_{\text{cyl}}\frac{\pi B^2 S/4}{v_1} = (6)\frac{\pi(3.5/12\,\text{ft})^2(3.9/12\,\text{ft})/4}{13.89\,\text{ft}^3/\text{lbm}} = 0.009380\,\text{lbm}$$

The net work produced per cycle is

$$W_{\text{net}} = m w_{\text{net}} = (0.009380\,\text{lbm})(84.44\,\text{Btu/lbm}) = 0.7920\,\text{Btu/cycle}$$

The power produced is determined from

$$\dot{W}_{\text{net}} = \frac{W_{\text{net}}\dot{n}}{N_{\text{rev}}} = \frac{(0.7920\,\text{Btu/cycle})(2500/60\,\text{rev/s})}{2\,\text{rev/cycle}}\left(\frac{1\,\text{hp}}{0.7068\,\text{Btu/s}}\right) = \mathbf{23.3\ hp}$$

since there are two revolutions per cycle in a four-stroke engine.

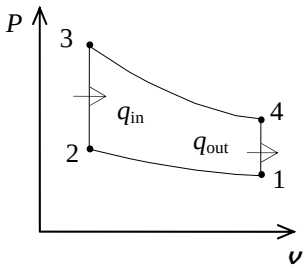
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9-38E An Otto cycle with non-isentropic compression and expansion processes is considered. The thermal efficiency, the heat addition, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis We begin by determining the temperatures of the cycle states using the process equations and component efficiencies. The ideal temperature at the end of the compression is then



$$T_{2s} = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (520\,\text{R})(8)^{1.4-1} = 1195\,\text{R}$$

With the isentropic compression efficiency, the actual temperature at the end of the compression is

$$\eta = \frac{T_{2s} - T_1}{T_2 - T_1} \longrightarrow T_2 = T_1 + \frac{T_{2s} - T_1}{\eta} = (520\,\text{R}) + \frac{(1195 - 520)\,\text{R}}{0.85} = 1314\,\text{R}$$

Similarly for the expansion,

$$T_{4s} = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1} = (2300 + 460\,\text{R})\left(\frac{1}{8}\right)^{1.4-1} = 1201\,\text{R}$$

$$\eta = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow T_4 = T_3 - \eta(T_3 - T_{4s}) = (2760\,\text{R}) - (0.95)(2760 - 1201)\,\text{R} = 1279\,\text{R}$$

The specific heat addition is that of process 2-3,

$$q_{\text{in}} = c_v(T_3 - T_2) = (0.171\,\text{Btu/lbm}\cdot\text{R})(2760 - 1314)\text{R} = \mathbf{247.3\ Btu/lbm}$$

The net work production is the difference between the work produced by the expansion and that used by the compression,

$$\begin{aligned} w_{\text{net}} &= c_v(T_3 - T_4) - c_v(T_2 - T_1) \\ &= (0.171\,\text{Btu/lbm}\cdot\text{R})(2760 - 1279)\text{R} - (0.171\,\text{Btu/lbm}\cdot\text{R})(1314 - 520)\text{R} \\ &= 117.5\,\text{Btu/lbm} \end{aligned}$$

The thermal efficiency of this cycle is then

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{117.5\,\text{Btu/lbm}}{247.3\,\text{Btu/lbm}} = \mathbf{0.475}$$

At the beginning of compression, the maximum specific volume of this cycle is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520\,\text{R})}{13\,\text{psia}} = 14.82\,\text{ft}^3/\text{lbm}$$

while the minimum specific volume of the cycle occurs at the end of the compression

$$v_2 = \frac{v_1}{r} = \frac{14.82\,\text{ft}^3/\text{lbm}}{8} = 1.852\,\text{ft}^3/\text{lbm}$$

The engine's mean effective pressure is then

$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{117.5\,\text{Btu/lbm}}{(14.82 - 1.852)\,\text{ft}^3/\text{lbm}}\left(\frac{5.404\,\text{psia}\cdot\text{ft}^3}{1\,\text{Btu}}\right) = \mathbf{49.0\ psia}$$

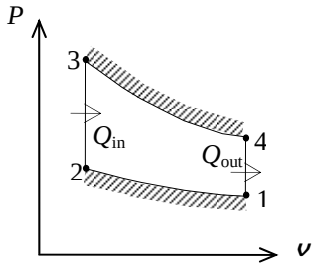
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9-39 An ideal Otto cycle with air as the working fluid has a compression ratio of 9.5. The highest pressure and temperature in the cycle, the amount of heat transferred, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (308\ \text{K})(9.5)^{0.4} = 757.9\ \text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{757.9\ \text{K}}{308\ \text{K}}\right)(100\ \text{kPa}) = 2338\ \text{kPa}$$

Process 3-4: isentropic expansion.

$$T_3 = T_4\left(\frac{v_4}{v_3}\right)^{k-1} = (800\ \text{K})(9.5)^{0.4} = \mathbf{1969\ K}$$

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1969\ \text{K}}{757.9\ \text{K}}\right)(2338\ \text{kPa}) = \mathbf{6072\ kPa}$$

(*b*) $$m = \frac{P_1 V_1}{RT_1} = \frac{(100\ \text{kPa})(0.0006\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\ \text{K})} = 6.788\times10^{-4}\ \text{kg}$$

$$Q_{in} = m(u_3 - u_2) = mc_v(T_3 - T_2) = (6.788\times10^{-4}\ \text{kg})(0.718\ \text{kJ/kg}\cdot\text{K})(1969 - 757.9)\text{K} = \mathbf{0.590\ kJ}$$

(c) Process 4-1: v = constant heat rejection.

$$Q_{out} = m(u_4 - u_1) = mc_v(T_4 - T_1) = -(6.788\times10^{-4}\ \text{kg})(0.718\ \text{kJ/kg}\cdot\text{K})(800 - 308)\text{K} = 0.240\ \text{kJ}$$

$$W_{net} = Q_{in} - Q_{out} = 0.590 - 0.240 = 0.350\ \text{kJ}$$

$$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{0.350\ \text{kJ}}{0.590\ \text{kJ}} = \mathbf{59.4\%}$$

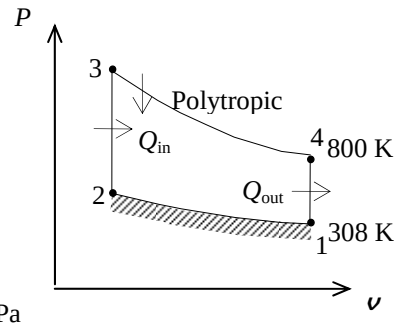
(*d*) $$V_{min} = V_2 = \frac{V_{max}}{r}$$

$$\text{MEP} = \frac{W_{net,out}}{V_1 - V_2} = \frac{W_{net,out}}{V_1(1 - 1/r)} = \frac{0.350\ \text{kJ}}{(0.0006\ \text{m}^3)(1 - 1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{652\ kPa}$$

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9-40 An Otto cycle with air as the working fluid has a compression ratio of 9.5. The highest pressure and temperature in the cycle, the amount of heat transferred, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.



Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (308\ \text{K})(9.5)^{0.4} = 757.9\ \text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{757.9\ \text{K}}{308\ \text{K}}\right)(100\ \text{kPa}) = 2338\ \text{kPa}$$

Process 3-4: polytropic expansion.

$$m = \frac{P_1 V_1}{RT_1} = \frac{(100\ \text{kPa})(0.0006\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\ \text{K})} = 6.788\times10^{-4}\ \text{kg}$$

$$T_3 = T_4\left(\frac{v_4}{v_3}\right)^{n-1} = (800\ \text{K})(9.5)^{0.35} = \mathbf{1759\ K}$$

$$W_{34} = \frac{mR(T_4 - T_3)}{1-n} = \frac{(6.788\times10^{-4})(0.287\ \text{kJ/kg}\cdot\text{K})(800-1759)\text{K}}{1-1.35} = 0.5338\ \text{kJ}$$

Then energy balance for process 3-4 gives

$$E_{in} - E_{out} = \Delta E_{system}$$
$$Q_{34,in} - W_{34,out} = m(u_4 - u_3)$$
$$Q_{34,in} = m(u_4 - u_3) + W_{34,out} = mc_v(T_4 - T_3) + W_{34,out}$$
$$Q_{34,in} = (6.788\times10^{-4}\ \text{kg})(0.718\ \text{kJ/kg}\cdot\text{K})(800-1759)\text{K} + 0.5338\ \text{kJ} = 0.0664\ \text{kJ}$$

That is, 0.066 kJ of heat is added to the air during the expansion process (This is not realistic, and probably is due to assuming constant specific heats at room temperature).

(*b*) Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1759\ \text{K}}{757.9\ \text{K}}\right)(2338\ \text{kPa}) = \mathbf{5426\ kPa}$$

$$Q_{23,in} = m(u_3 - u_2) = mc_v(T_3 - T_2)$$
$$Q_{23,in} = (6.788\times10^{-4}\ \text{kg})(0.718\ \text{kJ/kg}\cdot\text{K})(1759 - 757.9)\text{K} = 0.4879\ \text{kJ}$$

Therefore, $Q_{in} = Q_{23,in} + Q_{34,in} = 0.4879 + 0.0664 = \mathbf{0.5543\ kJ}$

(*c*) Process 4-1: v = constant heat rejection.

$$Q_{out} = m(u_4 - u_1) = mc_v(T_4 - T_1) = (6.788\times10^{-4}\ \text{kg})(0.718\ \text{kJ/kg}\cdot\text{K})(800-308)\text{K} = 0.2398\ \text{kJ}$$

$$W_{net,out} = Q_{in} - Q_{out} = 0.5543 - 0.2398 = 0.3145\ \text{kJ}$$

$$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{0.3145\ \text{kJ}}{0.5543\ \text{kJ}} = \mathbf{56.7\%}$$

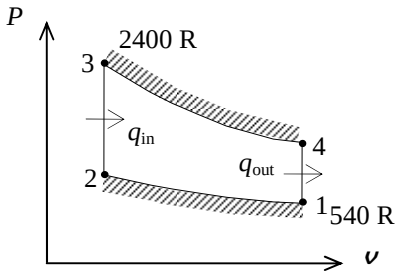
(*d*) $$V_{min} = V_2 = \frac{V_{max}}{r}$$

$$\text{MEP} = \frac{W_{net,out}}{V_1 - V_2} = \frac{W_{net,out}}{V_1(1-1/r)} = \frac{0.3145\ \text{kJ}}{(0.0006\ \text{m}^3)(1-1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{586\ kPa}$$

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9-41E An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The amount of heat transferred to the air during the heat addition process, the thermal efficiency, and the thermal efficiency of a Carnot cycle operating between the same temperature limits are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.



Properties The properties of air are given in Table A-17E.

Analysis (a) Process 1-2: isentropic compression.

$$T_1 = 540\text{R} \longrightarrow \begin{matrix} u_1 = 92.04\text{Btu/lbm} \\ v_{r_1} = 144.32 \end{matrix}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_2} = \frac{1}{r} v_{r_2} = \frac{1}{8}(144.32) = 18.04 \longrightarrow u_2 = 211.28 \text{ Btu/lbm}$$

Process 2-3: v = constant heat addition.

$$T_3 = 2400\text{R} \longrightarrow \begin{matrix} u_3 = 452.70 \text{ Btu/lbm} \\ v_{r_3} = 2.419 \end{matrix}$$

$$q_{in} = u_3 - u_2 = 452.70 - 211.28 = \mathbf{241.42\ Btu/lbm}$$

(b) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (8)(2.419) = 19.35 \longrightarrow u_4 = 205.54 \text{ Btu/lbm}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 205.54 - 92.04 = 113.50 \text{ Btu/lbm}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{113.50 \text{ Btu/lbm}}{241.42 \text{ Btu/lbm}} = \mathbf{53.0\%}$$

(c) The thermal efficiency of a Carnot cycle operating between the same temperature limits is

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540 \text{ R}}{2400 \text{ R}} = \mathbf{77.5\%}$$

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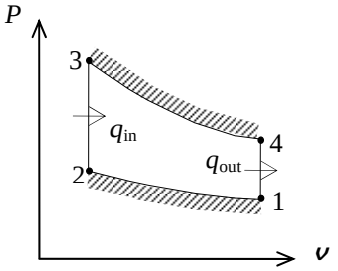
9-42E An ideal Otto cycle with argon as the working fluid has a compression ratio of 8. The amount of heat transferred to the argon during the heat addition process, the thermal efficiency, and the thermal efficiency of a Carnot cycle operating between the same temperature limits are to be determined.

Assumptions **1** The air-standard assumptions are applicable with argon as the working fluid. **2** Kinetic and potential energy changes are negligible. **3** Argon is an ideal gas with constant specific heats.

Properties The properties of argon are c_p = 0.1253 Btu/lbm.R, c_v = 0.0756 Btu/lbm.R, and k = 1.667 (Table A-2E).

Analysis (*a*) Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (540\ \text{R})(8)^{0.667} = 2161\ \text{R}$$



Process 2-3: v = constant heat addition.

$$\begin{aligned} q_{in} &= u_3 - u_2 = c_v(T_3 - T_2) \\ &= (0.0756\ \text{Btu/lbm.R})(2400 - 2161)\ \text{R} \\ &= \mathbf{18.07\ Btu/lbm} \end{aligned}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (2400\ \text{R})\left(\frac{1}{8}\right)^{0.667} = 600\ \text{R}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.0756\ \text{Btu/lbm.R})(600 - 540)\text{R} = 4.536\ \text{Btu/lbm}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{4.536\ \text{Btu/lbm}}{18.07\ \text{Btu/lbm}} = \mathbf{74.9\%}$$

(*c*) The thermal efficiency of a Carnot cycle operating between the same temperature limits is

$$\eta_{th,C} = 1 - \frac{T_L}{T_H} = 1 - \frac{540\ \text{R}}{2400\ \text{R}} = \mathbf{77.5\%}$$

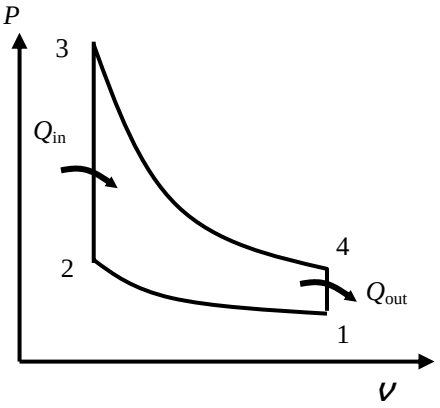
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9-43 A gasoline engine operates on an Otto cycle. The compression and expansion processes are modeled as polytropic. The temperature at the end of expansion process, the net work output, the thermal efficiency, the mean effective pressure, the engine speed for a given net power, and the specific fuel consumption are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at 850 K are c_p = 1.110 kJ/kg·K, c_v = 0.823 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.349 (Table A-2b).

Analysis (*a*) Process 1-2: polytropic compression



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = (310\text{ K})(11)^{1.3\text{-}1} = 636.5\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{n} = (100\text{ kPa})(11)^{1.3} = 2258\text{ kPa}$$

$$w_{12} = \frac{R(T_2 - T_1)}{1-n} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(636.5-310)\text{K}}{1-1.3} = -312.3\text{ kJ/kg}$$

Process 2-3: constant volume heat addition

$$T_3 = T_2\left(\frac{P_3}{P_2}\right) = (636.5\text{ K})\left(\frac{8000\text{ kPa}}{2258\text{ kPa}}\right) = 2255\text{ K}$$

$$q_{\text{in}} = u_3 - u_2 = c_v(T_3 - T_2)$$
$$= (0.823\text{ kJ/kg}\cdot\text{K})(2255-636.5)\text{K} = 1332\text{ kJ/kg}$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = (2255\text{ K})\left(\frac{1}{11}\right)^{1.3\text{-}1} = \mathbf{1098\text{ K}}$$

$$P_4 = P_3\left(\frac{v_2}{v_1}\right)^{n} = (8000\text{ kPa})\left(\frac{1}{11}\right)^{1.3} = 354.2\text{ kPa}$$

$$w_{34} = \frac{R(T_4 - T_3)}{1-n} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(1098-2255)\text{K}}{1-1.3} = 1106\text{ kJ/kg}$$

Process 4-1: constant volume heat rejection.

(*b*) The net work output and the thermal efficiency are

$$w_{\text{net,out}} = w_{34} - w_{12} = 1106 - 312.3 = \mathbf{794\text{ kJ/kg}}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{794\text{ kJ/kg}}{1332\text{ kJ/kg}} = 0.596 = \mathbf{59.6\%}$$

(*c*) The mean effective pressure is determined as follows

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(310\text{ K})}{100\text{ kPa}} = 0.8897\text{ m}^3/\text{kg} = v_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1-1/r)} = \frac{794\text{ kJ/kg}}{(0.8897\text{ m}^3/\text{kg})(1-1/11)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{982\text{ kPa}}$$

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(*d*) The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 11 = \frac{V_c + 0.0016\,\text{m}^3}{V_c} \longrightarrow V_c = 0.00016\,\text{m}^3$$

$$V_1 = V_c + V_d = 0.00016 + 0.0016 = 0.00176\,\text{m}^3$$

The total mass contained in the cylinder is

$$m_t = \frac{P_1 V_1}{RT_1} = \frac{(100\,\text{kPa})/0.00176\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(310\,\text{K})} = 0.001978\,\text{kg}$$

The engine speed for a net power output of 50 kW is

$$\dot{n} = 2\frac{\dot{W}_{\text{net}}}{m_t w_{\text{net}}} = (2\,\text{rev/cycle})\frac{50\,\text{kJ/s}}{(0.001978\,\text{kg})(794\,\text{kJ/kg}\cdot\text{cycle})}\left(\frac{60\,\text{s}}{1\,\text{min}}\right) = \mathbf{3820\ rev/min}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) The mass of fuel burned during one cycle is

$$\text{AF} = \frac{m_a}{m_f} = \frac{m_t - m_f}{m_f} \longrightarrow 16 = \frac{(0.001978\,\text{kg}) - m_f}{m_f} \longrightarrow m_f = 0.0001164\,\text{kg}$$

Finally, the specific fuel consumption is

$$\text{sfc} = \frac{m_f}{m_t w_{\text{net}}} = \frac{0.0001164\,\text{kg}}{(0.001978\,\text{kg})(794\,\text{kJ/kg})}\left(\frac{1000\,\text{g}}{1\,\text{kg}}\right)\left(\frac{3600\,\text{kJ}}{1\,\text{kWh}}\right) = \mathbf{267\ g/kWh}$$

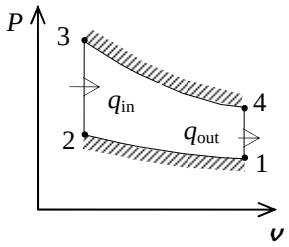
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9-44 The expressions for the maximum gas temperature and pressure of an ideal Otto cycle are to be determined when the compression ratio is doubled.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis The temperature at the end of the compression varies with the compression ratio as

$$T_2 = T_1\left(\frac{\nu_1}{\nu_2}\right)^{k-1} = T_1 r^{k-1}$$



since T_1 is fixed. The temperature rise during the combustion remains constant since the amount of heat addition is fixed. Then, the maximum cycle temperature is given by

$$T_3 = q_{in}/c_\nu + T_2 = q_{in}/c_\nu + T_1 r^{k-1}$$

The smallest gas specific volume during the cycle is

$$\nu_3 = \frac{\nu_1}{r}$$

When this is combined with the maximum temperature, the maximum pressure is given by

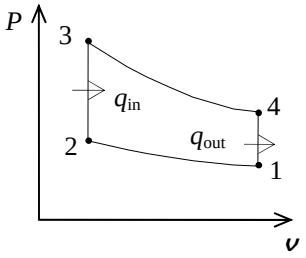
$$P_3 = \frac{RT_3}{\nu_3} = \frac{Rr}{\nu_1}\left(q_{in}/c_\nu + T_1 r^{k-1}\right)$$

9-45 It is to be determined if the polytropic exponent to be used in an Otto cycle model will be greater than or less than the isentropic exponent.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis During a polytropic process,

$$P\nu^n = \text{constant}$$
$$TP^{(n-1)/n} = \text{constant}$$



and for an isentropic process,

$$P\nu^k = \text{constant}$$
$$TP^{(k-1)/k} = \text{constant}$$

If heat is lost during the expansion of the gas,

$$T_4 > T_{4s}$$

where T_{4s} is the temperature that would occur if the expansion were reversible and adiabatic (n=k). This can only occur when

$$n \le k$$

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Diesel Cycle

9-46C A diesel engine differs from the gasoline engine in the way combustion is initiated. In diesel engines combustion is initiated by compressing the air above the self-ignition temperature of the fuel whereas it is initiated by a spark plug in a gasoline engine.

9-47C The Diesel cycle differs from the Otto cycle in the heat addition process only; it takes place at constant volume in the Otto cycle, but at constant pressure in the Diesel cycle.

9-48C The gasoline engine.

9-49C Diesel engines operate at high compression ratios because the diesel engines do not have the engine knock problem.

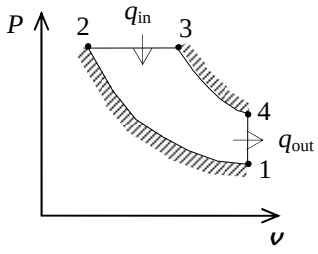
9-50C Cutoff ratio is the ratio of the cylinder volumes after and before the combustion process. As the cutoff ratio decreases, the efficiency of the diesel cycle increases.

9-51 An ideal diesel cycle has a compression ratio of 20 and a cutoff ratio of 1.3. The maximum temperature of the air and the rate of heat addition are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis We begin by using the process types to fix the temperatures of the states.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (288\text{ K})(20)^{1.4-1} = 954.6\text{ K}$$

$$T_3 = T_2\left(\frac{v_3}{v_2}\right) = T_2 r_c = (954.6\text{ K})(1.3) = \mathbf{1241\,K}$$

Combining the first law as applied to the various processes with the process equations gives

$$\eta_{th} = 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)} = 1 - \frac{1}{20^{1.4-1}}\frac{1.3^{1.4} - 1}{1.4(1.3-1)} = 0.6812$$

According to the definition of the thermal efficiency,

$$\dot{Q}_{in} = \frac{\dot{W}_{net}}{\eta_{th}} = \frac{250\text{ kW}}{0.6812} = \mathbf{367\,kW}$$

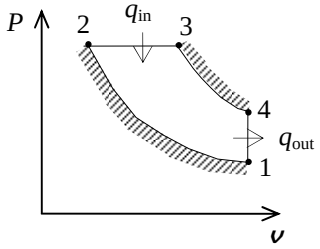
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9-52E An ideal diesel cycle has a a cutoff ratio of 1.4. The power produced is to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis The specific volume of the air at the start of the compression is



$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510\,\text{R})}{14.4\,\text{psia}} = 13.12\,\text{ft}^3/\text{lbm}$$

The total air mass taken by all 8 cylinders when they are charged is

$$m = N_{\text{cyl}}\frac{\Delta V}{v_1} = N_{\text{cyl}}\frac{\pi B^2 S/4}{v_1} = (8)\frac{\pi(4/12\,\text{ft})^2(4/12\,\text{ft})/4}{13.12\,\text{ft}^3/\text{lbm}} = 0.01774\,\text{lbm}$$

The rate at which air is processed by the engine is determined from

$$\dot{m} = \frac{m\dot{n}}{N_{\text{rev}}} = \frac{(0.01774\,\text{lbm/cycle})(1800/60\,\text{rev/s})}{2\,\text{rev/cycle}} = 0.2661\,\text{lbm/s} = 958.0\,\text{lbm/h}$$

since there are two revolutions per cycle in a four-stroke engine. The compression ratio is

$$r = \frac{1}{0.045} = 22.22$$

At the end of the compression, the air temperature is

$$T_2 = T_1 r^{k-1} = (510\,\text{R})(22.22)^{1.4-1} = 1763\,\text{R}$$

Application of the first law and work integral to the constant pressure heat addition gives

$$q_{\text{in}} = c_p(T_3 - T_2) = (0.240\,\text{Btu/lbm}\cdot\text{R})(2760-1763)\text{R} = 239.3\,\text{Btu/lbm}$$

while the thermal efficiency is

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)} = 1 - \frac{1}{22.22^{1.4-1}}\frac{1.4^{1.4}-1}{1.4(1.4-1)} = 0.6892$$

The power produced by this engine is then

$$\begin{aligned}\dot{W}_{\text{net}} &= \dot{m}w_{\text{net}} = \dot{m}\eta_{\text{th}}q_{\text{in}} \\ &= (958.0\,\text{lbm/h})(0.6892)(239.3\,\text{Btu/lbm})\left(\frac{1\,\text{hp}}{2544.5\,\text{Btu/h}}\right) \\ &= \mathbf{62.1\,hp}\end{aligned}$$

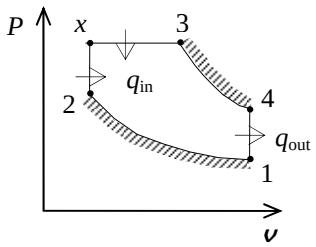
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9-53 An ideal dual cycle has a compression ratio of 14 and cutoff ratio of 1.2. The thermal efficiency, amount of heat added, and the maximum gas pressure and temperature are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The specific volume of the air at the start of the compression is



$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{80\,\text{kPa}} = 1.051\,\text{m}^3/\text{kg}$$

and the specific volume at the end of the compression is

$$v_2 = \frac{v_1}{r} = \frac{1.051\,\text{m}^3/\text{kg}}{14} = 0.07508\,\text{m}^3/\text{kg}$$

The pressure at the end of the compression is

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^k = P_1 r^k = (80\,\text{kPa})(14)^{1.4} = 3219\,\text{kPa}$$

and the maximum pressure is

$$P_x = P_3 = r_p P_2 = (1.5)(3219\,\text{kPa}) = \mathbf{4829\,kPa}$$

The temperature at the end of the compression is

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (293\,\text{K})(14)^{1.4-1} = 842.0\,\text{K}$$

and $$T_x = T_2\left(\frac{P_3}{P_2}\right) = (842.0\,\text{K})\left(\frac{4829\,\text{kPa}}{3219\,\text{kPa}}\right) = 1263\,\text{K}$$

From the definition of cutoff ratio

$$v_3 = r_c v_x = r_c v_2 = (1.2)(0.07508\,\text{m}^3/\text{kg}) = 0.09010\,\text{m}^3/\text{kg}$$

The remaining state temperatures are then

$$T_3 = T_x\left(\frac{v_3}{v_x}\right) = (1263\,\text{K})\left(\frac{0.09010}{0.07508}\right) = \mathbf{1516\,K}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1516\,\text{K})\left(\frac{0.09010}{1.051}\right)^{1.4-1} = 567.5\,\text{K}$$

Applying the first law and work expression to the heat addition processes gives

$$\begin{aligned} q_{\text{in}} &= c_v(T_x - T_2) + c_p(T_3 - T_x) \\ &= (0.718\,\text{kJ/kg}\cdot\text{K})(1263-842.0)\text{K} + (1.005\,\text{kJ/kg}\cdot\text{K})(1516-1263)\text{K} \\ &= \mathbf{556.5\,kJ/kg} \end{aligned}$$

The heat rejected is

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.718\,\text{kJ/kg}\cdot\text{K})(567.5-293)\text{K} = 197.1\,\text{kJ/kg}$$

Then, $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{197.1\,\text{kJ/kg}}{556.5\,\text{kJ/kg}} = \mathbf{0.646}$$

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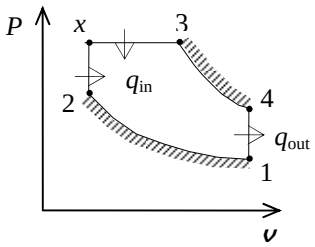
9-54 An ideal dual cycle has a compression ratio of 14 and cutoff ratio of 1.2. The thermal efficiency, amount of heat added, and the maximum gas pressure and temperature are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis The specific volume of the air at the start of the compression is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(253\ \text{K})}{80\ \text{kPa}} = 0.9076\ \text{m}^3/\text{kg}$$



and the specific volume at the end of the compression is

$$v_2 = \frac{v_1}{r} = \frac{0.9076\ \text{m}^3/\text{kg}}{14} = 0.06483\ \text{m}^3/\text{kg}$$

The pressure at the end of the compression is

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^k = P_1 r^k = (80\ \text{kPa})(14)^{1.4} = 3219\ \text{kPa}$$

and the maximum pressure is

$$P_x = P_3 = r_p P_2 = (1.5)(3219\ \text{kPa}) = \mathbf{4829\ kPa}$$

The temperature at the end of the compression is

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (253\ \text{K})(14)^{1.4-1} = 727.1\ \text{K}$$

and $$T_x = T_2\left(\frac{P_3}{P_2}\right) = (727.1\ \text{K})\left(\frac{4829\ \text{kPa}}{3219\ \text{kPa}}\right) = 1091\ \text{K}$$

From the definition of cutoff ratio

$$v_3 = r_c v_x = r_c v_2 = (1.2)(0.06483\ \text{m}^3/\text{kg}) = 0.07780\ \text{m}^3/\text{kg}$$

The remaining state temperatures are then

$$T_3 = T_x\left(\frac{v_3}{v_x}\right) = (1091\ \text{K})\left(\frac{0.07780}{0.06483}\right) = \mathbf{1309\ K}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1309\ \text{K})\left(\frac{0.07780}{0.9076}\right)^{1.4-1} = 490.0\ \text{K}$$

Applying the first law and work expression to the heat addition processes gives

$$\begin{aligned} q_{\text{in}} &= c_v(T_x - T_2) + c_p(T_3 - T_x) \\ &= (0.718\ \text{kJ/kg}\cdot\text{K})(1091 - 727.1)\text{K} + (1.005\ \text{kJ/kg}\cdot\text{K})(1309 - 1091)\text{K} \\ &= \mathbf{480.4\ kJ/kg} \end{aligned}$$

The heat rejected is

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.718\ \text{kJ/kg}\cdot\text{K})(490.0 - 253)\text{K} = 170.2\ \text{kJ/kg}$$

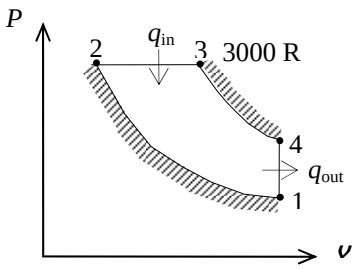
Then, $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{170.2\ \text{kJ/kg}}{480.4\ \text{kJ/kg}} = \mathbf{0.646}$$

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9-55E An air-standard Diesel cycle with a compression ratio of 18.2 is considered. The cutoff ratio, the heat rejection per unit mass, and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-17E.



Analysis (*a*) Process 1-2: isentropic compression.

$$T_1 = 540 \text{ R} \longrightarrow \begin{matrix} u_1 = 92.04 \text{ Btu/lbm} \\ v_{r_1} = 144.32 \end{matrix}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{18.2}(144.32) = 7.93 \longrightarrow \begin{matrix} T_2 = 1623.6 \text{ R} \\ h_2 = 402.05 \text{ Btu/lbm} \end{matrix}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{3000 \text{ R}}{1623.6 \text{ R}} = \mathbf{1.848}$$

(*b*) $$T_3 = 3000 \text{ R} \longrightarrow \begin{matrix} h_3 = 790.68 \text{ Btu/lbm} \\ v_{r_3} = 1.180 \end{matrix}$$

$$q_{\text{in}} = h_3 - h_2 = 790.68 - 402.05 = 388.63 \text{ Btu/lbm}$$

Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{1.848 v_2} v_{r_3} = \frac{r}{1.848} v_{r_3} = \frac{18.2}{1.848}(1.180) = 11.621 \longrightarrow u_4 = 250.91 \text{ Btu/lbm}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 250.91 - 92.04 = \mathbf{158.87 \ Btu/lbm}$$

(*c*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{158.87 \text{ Btu/lbm}}{388.63 \text{ Btu/lbm}} = \mathbf{59.1\%}$$

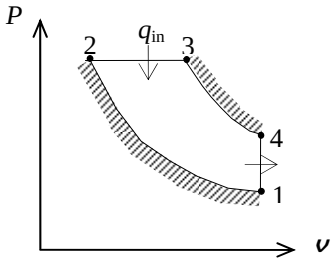
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9-56E An air-standard Diesel cycle with a compression ratio of 18.2 is considered. The cutoff ratio, the heat rejection per unit mass, and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm.R, c_v = 0.171 Btu/lbm.R, and k = 1.4 (Table A-2E).

Analysis (*a*) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (540\text{R})(18.2)^{0.4} = 1724\text{R}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{3000\text{ R}}{1724\text{ R}} = \mathbf{1.741}$$

(*b*) $$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) = (0.240\text{ Btu/lbm.R})(3000 - 1724)\text{R} = 306\text{ Btu/lbm}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1.741 v_2}{v_4}\right)^{k-1} = (3000\text{ R})\left(\frac{1.741}{18.2}\right)^{0.4} = 1173\text{ R}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$
$$= (0.171\text{ Btu/lbm.R})(1173 - 540)\text{R} = \mathbf{108\ Btu/lbm}$$

(*c*) $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{108\text{ Btu/lbm}}{306\text{ Btu/lbm}} = \mathbf{64.6\%}$$

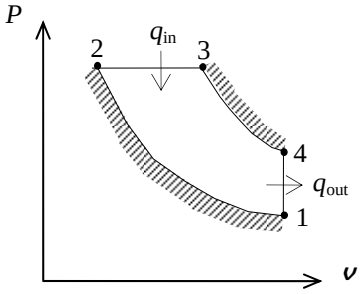
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9-57 An ideal diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (293\text{ K})(20)^{0.4} = 971.1\text{ K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2200\text{K}}{971.1\text{K}} = 2.265$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.265 v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.265}{r}\right)^{k-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.4} = 920.6\text{ K}$$

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(2200 - 971.1)\text{K} = 1235\text{ kJ/kg}$$

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(920.6 - 293)\text{K} = 450.6\text{ kJ/kg}$$

$$w_{net,out} = q_{in} - q_{out} = 1235 - 450.6 = 784.4\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{784.4\text{ kJ/kg}}{1235\text{ kJ/kg}} = \mathbf{63.5\%}$$

(*b*)
$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{95\text{ kPa}} = 0.885\text{ m}^3/\text{kg} = v_{max}$$

$$v_{min} = v_2 = \frac{v_{max}}{r}$$

$$\text{MEP} = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1 - 1/r)} = \frac{784.4\text{ kJ/kg}}{(0.885\text{ m}^3/\text{kg})(1 - 1/20)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{933\text{ kPa}}$$

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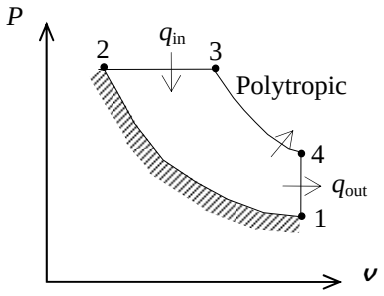
9-58 A diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (a) Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (293\ \text{K})(20)^{0.4} = 971.1\ \text{K}$$



Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2200\ \text{K}}{971.1\ \text{K}} = 2.265$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = T_3\left(\frac{2.265 v_2}{v_4}\right)^{n-1} = T_3\left(\frac{2.265}{\text{r}}\right)^{n-1} = (2200\ \text{K})\left(\frac{2.265}{20}\right)^{0.35} = 1026\ \text{K}$$

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) = (1.005\ \text{kJ/kg}\cdot\text{K})(2200 - 971.1)\ \text{K} = 1235\ \text{kJ/kg}$$

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\ \text{kJ/kg}\cdot\text{K})(1026 - 293)\ \text{K} = 526.3\ \text{kJ/kg}$$

Note that q_{out} in this case does not represent the entire heat rejected since some heat is also rejected during the polytropic process, which is determined from an energy balance on process 3-4:

$$w_{34,out} = \frac{R(T_4 - T_3)}{1-n} = \frac{(0.287\ \text{kJ/kg}\cdot\text{K})(1026 - 2200)\ \text{K}}{1 - 1.35} = 963\ \text{kJ/kg}$$

$$E_{in} - E_{out} = \Delta E_{system}$$

$$q_{34,in} - w_{34,out} = u_4 - u_3 \longrightarrow q_{34,in} = w_{34,out} + c_v(T_4 - T_3)$$
$$= 963\ \text{kJ/kg} + (0.718\ \text{kJ/kg}\cdot\text{K})(1026 - 2200)\ \text{K}$$
$$= 120.1\ \text{kJ/kg}$$

which means that 120.1 kJ/kg of heat is transferred to the combustion gases during the expansion process. This is unrealistic since the gas is at a much higher temperature than the surroundings, and a hot gas loses heat during polytropic expansion. The cause of this unrealistic result is the constant specific heat assumption. If we were to use u data from the air table, we would obtain

$$q_{34,in} = w_{34,out} + (u_4 - u_3) = 963 + (781.3 - 1872.4) = -128.1\ \text{kJ/kg}$$

which is a heat loss as expected. Then q_{out} becomes

$$q_{out} = q_{34,out} + q_{41,out} = 128.1 + 526.3 = 654.4\ \text{kJ/kg}$$

and

$$w_{net,out} = q_{in} - q_{out} = 1235 - 654.4 = 580.6\ \text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{580.6\ \text{kJ/kg}}{1235\ \text{kJ/kg}} = \mathbf{47.0\%}$$

(b) $$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})}{95\ \text{kPa}} = 0.885\ \text{m}^3/\text{kg} = v_{max}$$

$$v_{min} = v_2 = \frac{v_{max}}{r}$$

$$\text{MEP} = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1 - 1/r)} = \frac{580.6\ \text{kJ/kg}}{(0.885\ \text{m}^3/\text{kg})(1 - 1/20)}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{691\ kPa}$$

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9-59 Problem 9-58 is reconsidered. The effect of the compression ratio on the net work output, mean effective pressure, and thermal efficiency is to be investigated. Also, *T-s* and *P-v* diagrams for the cycle are to be plotted.

Analysis Using EES, the problem is solved as follows:

```
Procedure QTotal(q_12,q_23,q_34,q_41: q_in_total,q_out_total)
q_in_total = 0
q_out_total = 0
IF (q_12 > 0) THEN q_in_total = q_12 ELSE  q_out_total = - q_12
If q_23 > 0 then q_in_total = q_in_total + q_23 else q_out_total = q_out_total - q_23
If q_34 > 0 then q_in_total = q_in_total + q_34 else q_out_total = q_out_total - q_34
If q_41 > 0 then q_in_total = q_in_total + q_41 else q_out_total = q_out_total - q_41
END

"Input Data"
T[1]=293 [K]
P[1]=95 [kPa]
T[3] = 2200 [K]
n=1.35
{r_comp = 20}

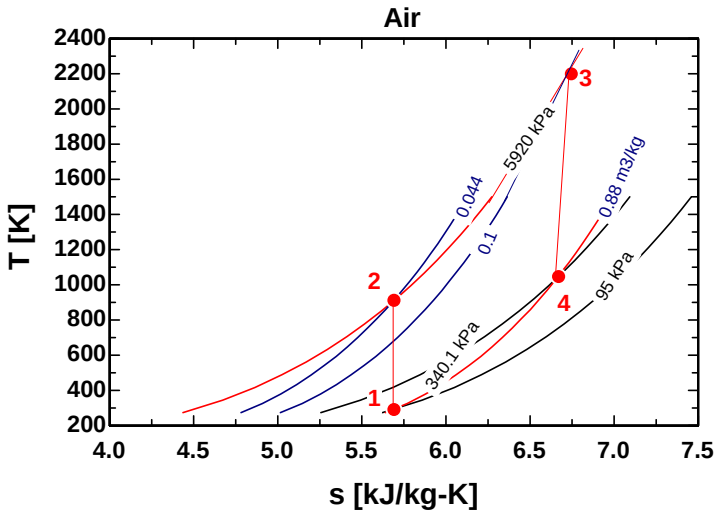
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant pressure heat addition"
P[3]=P[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =P[2]*(V[3] - V[2])"constant pressure process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is polytropic expansion"
P[3]/P[4] =(V[4]/V[3])^n
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 - w_34 = DELTAu_34    "q_34 is not 0 for the ploytropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
P[3]*V[3]^n = Const
w_34=(P[4]*V[4]-P[3]*V[3])/(1-n)
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0 "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

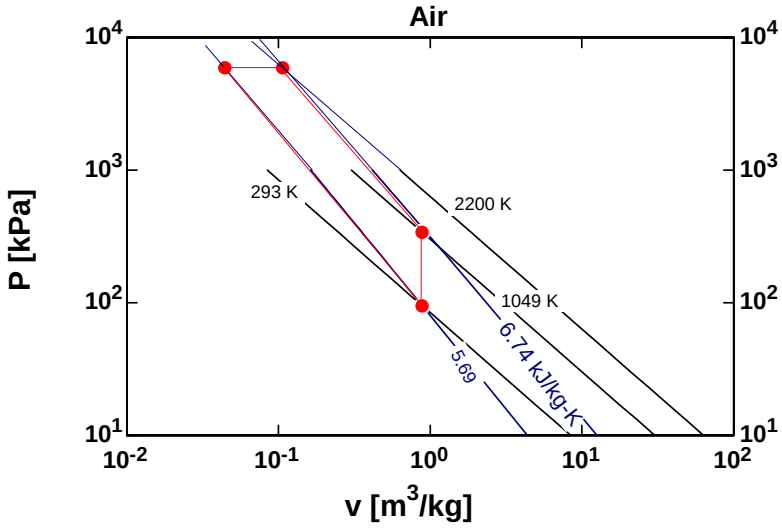
Call QTotal(q_12,q_23,q_34,q_41: q_in_total,q_out_total)
w_net = w_12+w_23+w_34+w_41
```

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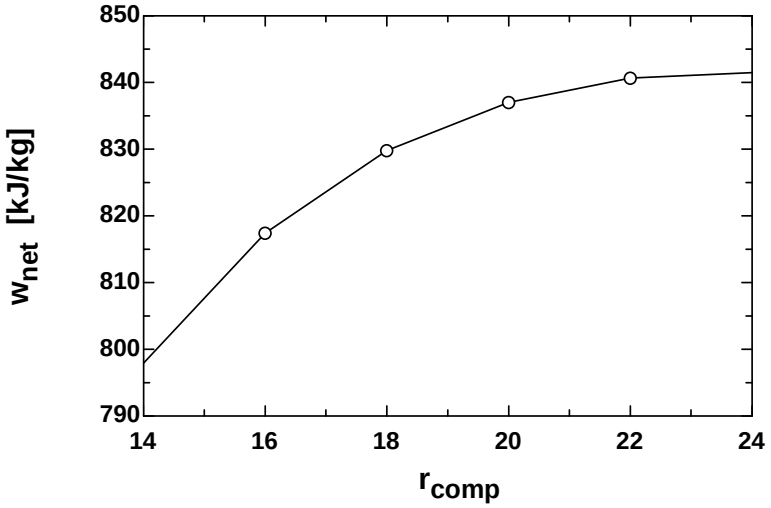
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
"The mean effective pressure is:"
MEP = w_net/(V[1]-V[2])

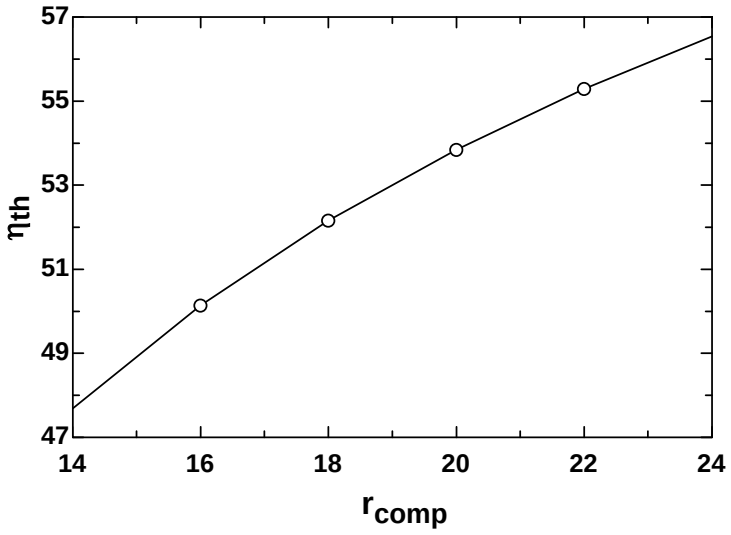
r_{comp}	η_{th}	MEP [kPa]	w_{net} [kJ/kg]
14	47.69	970.8	797.9
16	50.14	985	817.4
18	52.16	992.6	829.8
20	53.85	995.4	837.0
22	55.29	994.9	840.6
24	56.54	992	841.5

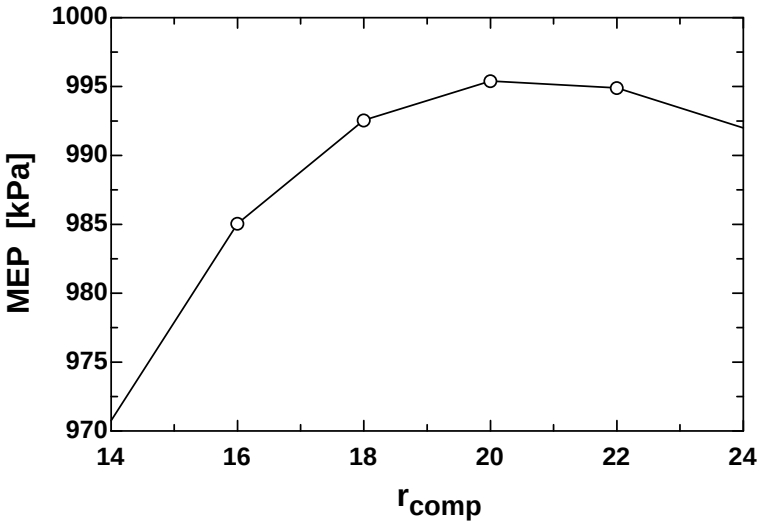




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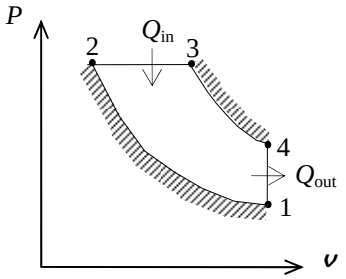
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9-60 A four-cylinder ideal diesel engine with air as the working fluid has a compression ratio of 22 and a cutoff ratio of 1.8. The power the engine will deliver at 2300 rpm is to be determined.

Assumptions **1** The cold air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (343\text{ K})(22)^{0.4} = 1181\text{ K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 1.8T_2 = (1.8)(1181\text{ K}) = 2126\text{ K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.2v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.2}{r}\right)^{k-1} = (2216\text{ K})\left(\frac{1.8}{22}\right)^{0.4} = 781\text{ K}$$

$$m = \frac{P_1 v_1}{RT_1} = \frac{(97\text{ kPa})(0.0020\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343\text{ K})} = 0.001971\text{ kg}$$

$$\begin{aligned} Q_{in} &= m(h_3 - h_2) = mc_p(T_3 - T_2) \\ &= (0.001971\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(2216 - 1181)\text{K} = 1.871\text{ kJ} \end{aligned}$$

$$\begin{aligned} Q_{out} &= m(u_4 - u_1) = mc_v(T_4 - T_1) \\ &= (0.001971\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(781 - 343)\text{K} = 0.6198\text{ kJ} \end{aligned}$$

$$W_{net,out} = Q_{in} - Q_{out} = 1.871 - 0.6198 = 1.251\text{ kJ/rev}$$

$$\dot{W}_{net,out} = \dot{n}W_{net,out} = (2300/60\text{ rev/s})(1.251\text{ kJ/rev}) = \mathbf{48.0\text{ kW}}$$

Discussion Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

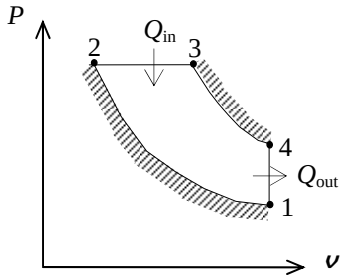
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9-61 A four-cylinder ideal diesel engine with nitrogen as the working fluid has a compression ratio of 22 and a cutoff ratio of 1.8. The power the engine will deliver at 2300 rpm is to be determined.

Assumptions **1** The air-standard assumptions are applicable with nitrogen as the working fluid. **2** Kinetic and potential energy changes are negligible. **3** Nitrogen is an ideal gas with constant specific heats.

Properties The properties of nitrogen at room temperature are c_p = 1.039 kJ/kg·K, c_v = 0.743 kJ/kg·K, R = 0.2968 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (343\text{ K})(22)^{0.4} = 1181\text{ K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 1.8T_2 = (1.8)(1181\text{ K}) = 2126\text{ K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.2v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.2}{r}\right)^{k-1} = (2216\text{ K})\left(\frac{1.8}{22}\right)^{0.4} = 781\text{ K}$$

$$m = \frac{P_1 v_1}{RT_1} = \frac{(97\text{ kPa})(0.0020\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343\text{ K})} = 0.001906\text{ kg}$$

$$\begin{aligned} Q_{in} &= m(h_3 - h_2) = mc_p(T_3 - T_2) \\ &= (0.001906\text{ kg})(1.039\text{ kJ/kg}\cdot\text{K})(2216 - 1181)\text{K} = 1.871\text{ kJ} \end{aligned}$$

$$\begin{aligned} Q_{out} &= m(u_4 - u_1) = mc_v(T_4 - T_1) \\ &= (0.001906\text{ kg})(0.743\text{ kJ/kg}\cdot\text{K})(781 - 343)\text{K} = 0.6202\text{ kJ} \end{aligned}$$

$$W_{net,out} = Q_{in} - Q_{out} = 1.871 - 0.6202 = 1.251\text{ kJ/rev}$$

$$\dot{W}_{net,out} = \dot{n}W_{net,out} = (2300/60\text{ rev/s})(1.251\text{ kJ/rev}) = \mathbf{47.9\ kW}$$

Discussion Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

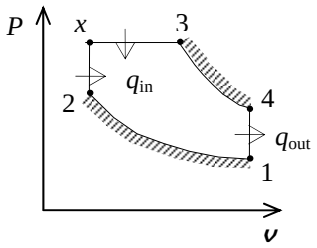
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9-62 An ideal dual cycle has a compression ratio of 18 and cutoff ratio of 1.1. The power produced by the cycle is to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis We begin by fixing the temperatures at all states.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (291\,\text{K})(18)^{1.4-1} = 924.7\,\text{K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1 r^{k} = (90\,\text{kPa})(18)^{1.4} = 5148\,\text{kPa}$$

$$P_x = P_3 = r_p P_2 = (1.1)(5148\,\text{kPa}) = 5663\,\text{kPa}$$

$$T_x = T_2\left(\frac{P_x}{P_2}\right) = (924.7\,\text{K})\left(\frac{5663\,\text{kPa}}{5148\,\text{kPa}}\right) = 1017\,\text{K}$$

$$T_3 = r_c T_x = (1.1)(1017\,\text{K}) = 1119\,\text{K}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{r_c}{r}\right)^{k-1} = (1119\,\text{K})\left(\frac{1.1}{18}\right)^{1.4-1} = 365.8\,\text{K}$$

Applying the first law to each of the processes gives

$$w_{1-2} = c_v(T_2 - T_1) = (0.718\,\text{kJ/kg}\cdot\text{K})(924.7 - 291)\text{K} = 455.0\,\text{kJ/kg}$$

$$q_{x-3} = c_p(T_3 - T_x) = (1.005\,\text{kJ/kg}\cdot\text{K})(1119 - 1017)\text{K} = 102.5\,\text{kJ/kg}$$

$$w_{x-3} = q_{x-3} - c_v(T_3 - T_x) = 102.5 - (0.718\,\text{kJ/kg}\cdot\text{K})(1119 - 1017)\text{K} = 29.26\,\text{kJ/kg}$$

$$w_{3-4} = c_v(T_3 - T_4) = (0.718\,\text{kJ/kg}\cdot\text{K})(1119 - 365.8)\text{K} = 540.8\,\text{kJ/kg}$$

The net work of the cycle is

$$w_{\text{net}} = w_{3-4} + w_{x-3} - w_{1-2} = 540.8 + 29.26 - 455.0 = 115.1\,\text{kJ/kg}$$

The mass in the device is given by

$$m = \frac{P_1 V_1}{RT_1} = \frac{(90\,\text{kPa})(0.003\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(291\,\text{K})} = 0.003233\,\text{kg}$$

The net power produced by this engine is then

$$\dot{W}_{\text{net}} = m w_{\text{net}} \dot{n} = (0.003233\,\text{kg/cycle})(115.1\,\text{kJ/kg})(4000/60\,\text{cycle/s}) = \mathbf{24.8\,kW}$$

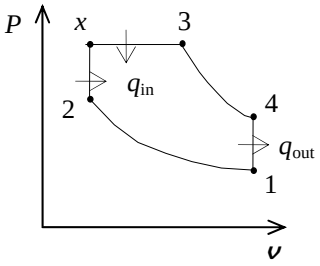
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9-63 A dual cycle with non-isentropic compression and expansion processes is considered. The power produced by the cycle is to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis We begin by fixing the temperatures at all states.



$$T_{2s} = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (291\,\text{K})(18)^{1.4-1} = 924.7\,\text{K}$$

$$\eta = \frac{T_{2s} - T_1}{T_2 - T_1} \longrightarrow T_2 = T_1 + \frac{T_{2s} - T_1}{\eta} = (291\,\text{K}) + \frac{(924.7 - 291)\,\text{K}}{0.85} = 1037\,\text{K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1 r^{k} = (90\,\text{kPa})(18)^{1.4} = 5148\,\text{kPa}$$

$$P_x = P_3 = r_p P_2 = (1.1)(5148\,\text{kPa}) = 5663\,\text{kPa}$$

$$T_x = T_2\left(\frac{P_x}{P_2}\right) = (1037\,\text{K})\left(\frac{5663\,\text{kPa}}{5148\,\text{kPa}}\right) = 1141\,\text{K}$$

$$T_3 = r_c T_x = (1.1)(1141\,\text{K}) = 1255\,\text{K}$$

$$T_{4s} = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{r_c}{r}\right)^{k-1} = (1255\,\text{K})\left(\frac{1.1}{18}\right)^{1.4-1} = 410.3\,\text{K}$$

$$\eta = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow T_4 = T_3 - \eta(T_3 - T_{4s}) = (1255\,\text{K}) - (0.90)(1255 - 410.3)\,\text{K} = 494.8\,\text{K}$$

Applying the first law to each of the processes gives

$$w_{1-2} = c_v(T_2 - T_1) = (0.718\,\text{kJ/kg}\cdot\text{K})(1037 - 291)\text{K} = 535.6\,\text{kJ/kg}$$

$$q_{x-3} = c_p(T_3 - T_x) = (1.005\,\text{kJ/kg}\cdot\text{K})(1255 - 1141)\text{K} = 114.6\,\text{kJ/kg}$$

$$w_{x-3} = q_{x-3} - c_v(T_3 - T_x) = 114.6 - (0.718\,\text{kJ/kg}\cdot\text{K})(1255 - 1141)\text{K} = 32.75\,\text{kJ/kg}$$

$$w_{3-4} = c_v(T_3 - T_4) = (0.718\,\text{kJ/kg}\cdot\text{K})(1255 - 494.8)\text{K} = 545.8\,\text{kJ/kg}$$

The net work of the cycle is

$$w_{\text{net}} = w_{3-4} + w_{x-3} - w_{1-2} = 545.8 + 32.75 - 535.6 = 42.95\,\text{kJ/kg}$$

The mass in the device is given by

$$m = \frac{P_1 V_1}{RT_1} = \frac{(90\,\text{kPa})(0.003\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(291\,\text{K})} = 0.003233\,\text{kg}$$

The net power produced by this engine is then

$$\dot{W}_{\text{net}} = m w_{\text{net}} \dot{n} = (0.003233\,\text{kg/cycle})(42.95\,\text{kJ/kg})(4000/60\,\text{cycle/s}) = \mathbf{9.26\,kW}$$

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9-64E An ideal dual cycle has a compression ratio of 15 and cutoff ratio of 1.4. The net work, heat addition, and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis Working around the cycle, the germane properties at the various states are

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (535\,\text{R})(15)^{1.4-1} = 1580\,\text{R}$$

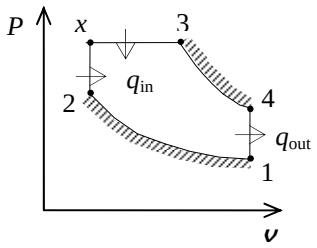
$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1 r^{k} = (14.2\,\text{psia})(15)^{1.4} = 629.2\,\text{psia}$$

$$P_x = P_3 = r_p P_2 = (1.1)(629.2\,\text{psia}) = 692.1\,\text{psia}$$

$$T_x = T_2\left(\frac{P_x}{P_2}\right) = (1580\,\text{R})\left(\frac{692.1\,\text{psia}}{629.2\,\text{psia}}\right) = 1738\,\text{R}$$

$$T_3 = T_x\left(\frac{v_3}{v_x}\right) = T_x r_c = (1738\,\text{R})(1.4) = 2433\,\text{R}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{r_c}{r}\right)^{k-1} = (2433\,\text{R})\left(\frac{1.4}{15}\right)^{1.4-1} = 942.2\,\text{R}$$



Applying the first law to each of the processes gives

$$w_{1-2} = c_v(T_2 - T_1) = (0.171\,\text{Btu/lbm}\cdot\text{R})(1580-535)\text{R} = 178.7\,\text{Btu/lbm}$$

$$q_{2-x} = c_v(T_x - T_2) = (0.171\,\text{Btu/lbm}\cdot\text{R})(1738-1580)\text{R} = 27.02\,\text{Btu/lbm}$$

$$q_{x-3} = c_p(T_3 - T_x) = (0.240\,\text{Btu/lbm}\cdot\text{R})(2433-1738)\text{R} = 166.8\,\text{Btu/lbm}$$

$$w_{x-3} = q_{x-3} - c_v(T_3 - T_x) = 166.8\,\text{Btu/lbm} - (0.171\,\text{Btu/lbm}\cdot\text{R})(2433-1738)\text{R} = 47.96\,\text{Btu/lbm}$$

$$w_{3-4} = c_v(T_3 - T_4) = (0.171\,\text{Btu/lbm}\cdot\text{R})(2433-942.2)\text{R} = 254.9\,\text{Btu/lbm}$$

The net work of the cycle is

$$w_{\text{net}} = w_{3-4} + w_{x-3} - w_{1-2} = 254.9 + 47.96 - 178.7 = \mathbf{124.2\ Btu/lbm}$$

and the net heat addition is

$$q_{\text{in}} = q_{2-x} + q_{x-3} = 27.02 + 166.8 = \mathbf{193.8\ Btu/lbm}$$

Hence, the thermal efficiency is

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{124.2\,\text{Btu/lbm}}{193.8\,\text{Btu/lbm}} = \mathbf{0.641}$$

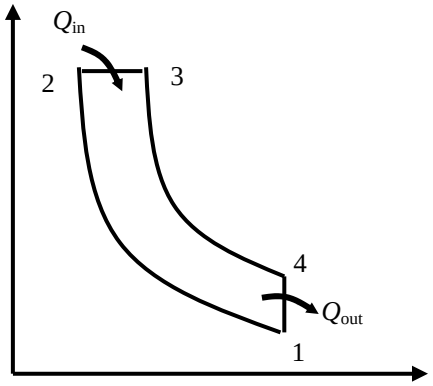
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9-65 A six-cylinder compression ignition engine operates on the ideal Diesel cycle. The maximum temperature in the cycle, the cutoff ratio, the net work output per cycle, the thermal efficiency, the mean effective pressure, the net power output, and the specific fuel consumption are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at 850 K are c_p = 1.110 kJ/kg·K, c_v = 0.823 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.349 (Table A-2b).

Analysis (*a*) Process 1-2: Isentropic compression



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (340\text{ K})(19)^{1.349\text{-}1} = 950.1\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = (95\text{ kPa})(19)^{1.349} = 5044\text{ kPa}$$

The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 19 = \frac{V_c + 0.0045\text{ m}^3}{V_c}$$

$$V_c = 0.0001778\text{ m}^3$$

$$V_1 = V_c + V_d = 0.0001778 + 0.0032 = 0.003378\text{ m}^3$$

The total mass contained in the cylinder is

$$m = \frac{P_1V_1}{RT_1} = \frac{(95\text{ kPa})(0.003378\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(340\text{ K})} = 0.003288\text{ kg}$$

The mass of fuel burned during one cycle is

$$AF = \frac{m_a}{m_f} = \frac{m - m_f}{m_f} \longrightarrow 28 = \frac{(0.003288\text{ kg}) - m_f}{m_f} \longrightarrow m_f = 0.0001134\text{ kg}$$

Process 2-3: constant pressure heat addition

$$Q_{\text{in}} = m_f q_{\text{HV}}\eta_c = (0.0001134\text{ kg})(42{,}500\text{ kJ/kg})(0.98) = 4.723\text{ kJ}$$

$$Q_{\text{in}} = mc_v(T_3 - T_2) \longrightarrow 4.723\text{ kJ} = (0.003288\text{ kg})(0.823\text{ kJ/kg.K})(T_3 - 950.1)\text{K} \longrightarrow T_3 = \mathbf{2244\text{ K}}$$

The cutoff ratio is

$$\beta = \frac{T_3}{T_2} = \frac{2244\text{ K}}{950.1\text{ K}} = \mathbf{2.362}$$

(*b*) $$V_2 = \frac{V_1}{r} = \frac{0.003378\text{ m}^3}{19} = 0.0001778\text{ m}^3$$

$$V_3 = \beta V_2 = (2.362)(0.0001778\text{ m}^3) = 0.0004199\text{ m}^3$$

$$V_4 = V_1$$

$$P_3 = P_2$$

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Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (2244\ \text{K})\left(\frac{0.0004199\ \text{m}^3}{0.003378\ \text{m}^3}\right)^{1.349\text{-}1} = 1084\ \text{K}$$

$$P_4 = P_3\left(\frac{v_3}{v_4}\right)^{k} = (5044\ \text{kPa})\left(\frac{0.0004199\ \text{m}^3}{0.003378\ \text{m}^3}\right)^{1.349} = 302.9\ \text{kPa}$$

Process 4-1: constant voume heat rejection.

$$Q_{\text{out}} = mc_v(T_4 - T_1) = (0.003288\ \text{kg})(0.823\ \text{kJ/kg}\cdot\text{K})(1084 - 340)\text{K} = 2.013\ \text{kJ}$$

The net work output and the thermal efficiency are

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 4.723 - 2.013 = \mathbf{2.710\ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{2.710\ \text{kJ}}{4.723\ \text{kJ}} = 0.5737 = \mathbf{57.4\%}$$

(*c*) The mean effective pressure is determined to be

$$\text{MEP} = \frac{W_{\text{net,out}}}{v_1 - v_2} = \frac{2.710\ \text{kJ}}{(0.003378 - 0.0001778)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{847\ kPa}$$

(*d*) The power for engine speed of 1750 rpm is

$$\dot{W}_{\text{net}} = W_{\text{net}}\frac{\dot{n}}{2} = (2.710\ \text{kJ/cycle})\frac{1750\ (\text{rev/min})}{(2\ \text{rev/cycle})}\left(\frac{1\,\text{min}}{60\ \text{s}}\right) = \mathbf{39.5\ kW}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) Finally, the specific fuel consumption is

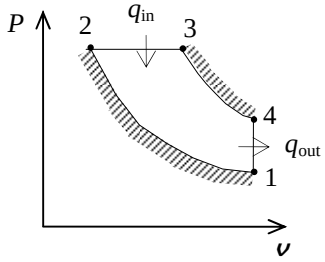
$$\text{sfc} = \frac{m_f}{W_{\text{net}}} = \frac{0.0001134\ \text{kg}}{2.710\ \text{kJ/kg}}\left(\frac{1000\ \text{g}}{1\ \text{kg}}\right)\left(\frac{3600\ \text{kJ}}{1\ \text{kWh}}\right) = \mathbf{151\ g/kWh}$$

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9-66 An expression for cutoff ratio of an ideal diesel cycle is to be developed.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis Employing the isentropic process equations,



$$T_2 = T_1 r^{k-1}$$

while the ideal gas law gives

$$T_3 = T_2 r_c = r_c r^{k-1} T_1$$

When the first law and the closed system work integral is applied to the constant pressure heat addition, the result is

$$q_{in} = c_p (T_3 - T_2) = c_p (r_c r^{k-1} T_1 - r^{k-1} T_1)$$

When this is solved for cutoff ratio, the result is

$$r_c = 1 + \frac{q_{in}}{c_p r^{k-1} T_1}$$

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9-67 An expression for the thermal efficiency of a dual cycle is to be developed and the thermal efficiency for a given case is to be calculated.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

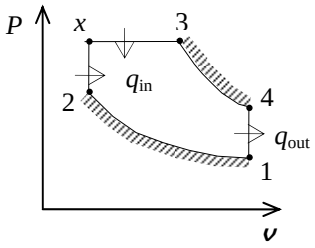
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2)

Analysis The thermal efficiency of a dual cycle may be expressed as

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_x - T_2) + c_p(T_3 - T_x)}$$

By applying the isentropic process relations for ideal gases with constant specific heats to the processes 1-2 and 3-4, as well as the ideal gas equation of state, the temperatures may be eliminated from the thermal efficiency expression. This yields the result

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_p r_c^k - 1}{k r_p (r_c - 1) + r_p - 1}\right]$$



where

$$r_p = \frac{P_x}{P_2} \quad \text{and} \quad r_c = \frac{v_3}{v_x}$$

When $r_c = r_p$, we obtain

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\left(\frac{r_p^{k+1} - 1}{k(r_p^2 - r_p) + r_p - 1}\right)$$

For the case r = 20 and r_p = 2,

$$\eta_{\text{th}} = 1 - \frac{1}{20^{1.4-1}}\left(\frac{2^{1.4+1} - 1}{1.4(2^2 - 2) + 2 - 1}\right) = \mathbf{0.660}$$

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9-68 An expression regarding the thermal efficiency of a dual cycle for a special case is to be obtained.

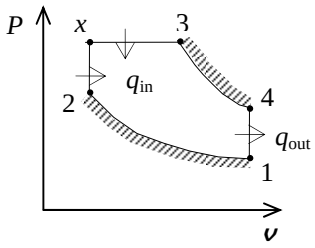
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis The thermal efficiency of a dual cycle may be expressed as

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_x - T_2) + c_p(T_3 - T_x)}$$

By applying the isentropic process relations for ideal gases with constant specific heats to the processes 1-2 and 3-4, as well as the ideal gas equation of state, the temperatures may be eliminated from the thermal efficiency expression. This yields the result

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_p r_c^k - 1}{k r_p (r_c - 1) + r_p - 1}\right]$$



where

$$r_p = \frac{P_x}{P_2} \quad \text{and} \quad r_c = \frac{v_3}{v_x}$$

When $r_c = r_p$, we obtain

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\left(\frac{r_p^{k+1} - 1}{k(r_p^2 - r_p) + r_p - 1}\right)$$

Rearrangement of this result gives

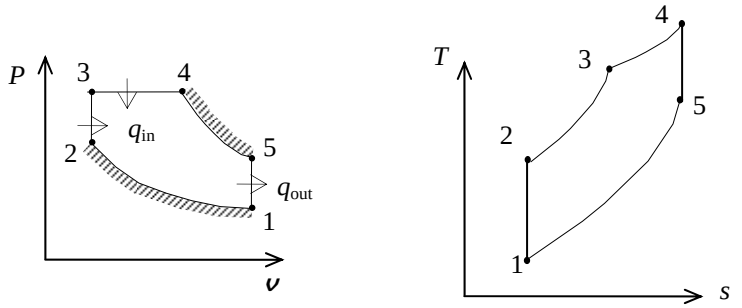
$$\frac{r_p^{k+1} - 1}{k(r_p^2 - r_p) + r_p - 1} = (1 - \eta_{\text{th}}) r^{k-1}$$

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9-69 The five processes of the dual cycle is described. The *P-v* and *T-s* diagrams for this cycle is to be sketched. An expression for the cycle thermal efficiency is to be obtained and the limit of the efficiency is to be evaluated for certain cases.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis (a) The *P-v* and *T-s* diagrams for this cycle are as shown.



(b) Apply first law to the closed system for processes 2-3, 3-4, and 5-1 to show:

$$q_{in} = C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right)$$
$$q_{out} = C_v\left(T_5 - T_1\right)$$

The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_v\left(T_5 - T_1\right)}{C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right)} = 1 - \frac{T_1\left(T_5/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right) + kT_3\left(T_4/T_3 - 1\right)}$$
$$\eta_{th} = 1 - \frac{\left(T_5/T_1 - 1\right)}{\frac{T_2}{T_1}\left(T_3/T_2 - 1\right) + k\frac{T_3}{T_1}\left(T_4/T_3 - 1\right)}$$

Process 1-2 is isentropic; therefore,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = r^{k-1}$$

Process 2-3 is constant volume; therefore,

$$\frac{T_3}{T_2} = \frac{P_3V_3}{P_2V_2} = \frac{P_3}{P_2} = r_p$$

Process 3-4 is constant pressure; therefore,

$$\frac{P_4V_4}{T_4} = \frac{P_3V_3}{T_3} \Rightarrow \frac{T_4}{T_3} = \frac{V_4}{V_3} = r_c$$

Process 4-5 is isentropic; therefore,

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{k-1} = \left(\frac{V_4}{V_1}\right)^{k-1} = \left(\frac{r_cV_3}{V_1}\right)^{k-1} = \left(\frac{r_cV_2}{V_1}\right)^{k-1} = \left(\frac{r_c}{r}\right)^{k-1}$$

Process 5-1 is constant volume; however T_5/T_1 is found from the following.

$$\frac{T_5}{T_1} = \frac{T_5}{T_4}\frac{T_4}{T_3}\frac{T_3}{T_2}\frac{T_2}{T_1} = \left(\frac{r_c}{r}\right)^{k-1} r_c\, r_p r^{k-1} = r_c^k r_p$$

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The ratio T_3/T_1 is found from the following.

$$\frac{T_3}{T_1} = \frac{T_3}{T_2}\frac{T_2}{T_1} = r_p r^{k-1}$$

The efficiency becomes

$$\eta_{th} = 1 - \frac{r_c^k r_p - 1}{r^{k-1}\left(r_p - 1\right) + k r_p r^{k-1}\left(r_c - 1\right)}$$

(c) In the limit as r_p approaches unity, the cycle thermal efficiency becomes

$$\lim_{r_p \to 1} \eta_{th} = 1 - \left\{ \lim_{r_p \to 1} \frac{r_c^k r_p - 1}{r^{k-1}\left(r_p - 1\right) + k r_p r^{k-1}\left(r_c - 1\right)} \right\}$$

$$\lim_{r_p \to 1} \eta_{th} = 1 - \frac{1}{r^{k-1}} \left[\frac{r_c^k - 1}{k\left(r_c - 1\right)} \right] = \eta_{th\ Diesel}$$

(d) In the limit as r_c approaches unity, the cycle thermal efficiency becomes

$$\lim_{r_c \to 1} \eta_{th} = 1 - \left\{ \lim_{r_p \to 1} \frac{r_c^k r_p - 1}{r^{k-1}\left(r_p - 1\right) + k r_p r^{k-1}\left(r_c - 1\right)} \right\} = 1 - \left\{ \frac{r_p - 1}{r^{k-1}\left(r_p - 1\right)} \right\}$$

$$\lim_{r_c \to 1} \eta_{th} = 1 - \frac{1}{r^{k-1}} = \eta_{th\ Otto}$$

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Stirling and Ericsson Cycles

9-70C The Stirling cycle.

9-71C The two isentropic processes of the Carnot cycle are replaced by two constant pressure regeneration processes in the Ericsson cycle.

9-72C The efficiencies of the Carnot and the Stirling cycles would be the same, the efficiency of the Otto cycle would be less.

9-73C The efficiencies of the Carnot and the Ericsson cycles would be the same, the efficiency of the Diesel cycle would be less.

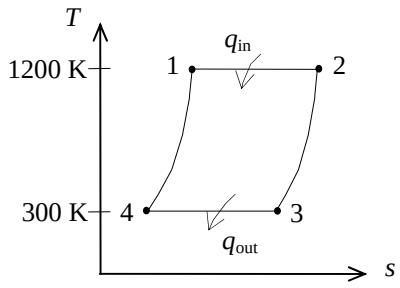
9-74 An ideal steady-flow Ericsson engine with air as the working fluid is considered. The maximum pressure in the cycle, the net work output, and the thermal efficiency of the cycle are to be determined.

Assumptions Air is an ideal gas.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

Analysis (*a*) The entropy change during process 3-4 is

$$s_4 - s_3 = -\frac{q_{34,\text{out}}}{T_0} = -\frac{150 \text{ kJ/kg}}{300 \text{ K}} = -0.5 \text{ kJ/kg} \cdot \text{K}$$



and

$$s_4 - s_3 = c_p \ln \frac{T_4}{T_3}^{\nearrow 0} - R \ln \frac{P_4}{P_3}$$

$$= -(0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{P_4}{120 \text{ kPa}} = -0.5 \text{ kJ/kg} \cdot \text{K}$$

It yields $P_4 =$ **685.2 kPa**

(*b*) For reversible cycles,

$$\frac{q_{\text{out}}}{q_{\text{in}}} = \frac{T_L}{T_H} \longrightarrow q_{\text{in}} = \frac{T_H}{T_L} q_{\text{out}} = \frac{1200 \text{ K}}{300 \text{ K}} (150 \text{ kJ/kg}) = 600 \text{ kJ/kg}$$

Thus,

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 600 - 150 = \mathbf{450 \text{ kJ/kg}}$$

(*c*) The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{K}}{1200\text{K}} = \mathbf{75.0\%}$$

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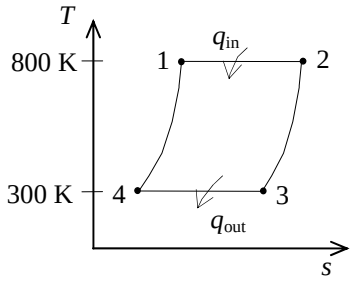
9-75 An ideal Stirling engine with air as the working fluid operates between the specified temperature and pressure limits. The net work produced per cycle and the thermal efficiency of the cycle are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis Since the specific volume is constant during process 2-3,

$$P_2 = P_3 \frac{T_2}{T_3} = (100 \text{ kPa})\left(\frac{800 \text{ K}}{300 \text{ K}}\right) = 266.7 \text{ kPa}$$



Heat is only added to the system during reversible process 1-2. Then,

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1}$$

$$= 0 - (0.287 \text{ kJ/kg} \cdot \text{K})\ln\left(\frac{266.7 \text{ kPa}}{2000 \text{ kPa}}\right)$$

$$= 0.5782 \text{ kJ/kg} \cdot \text{K}$$

$$q_{in} = T_1(s_2 - s_1) = (800 \text{ K})(0.5782 \text{ kJ/kg} \cdot \text{K}) = 462.6 \text{ kJ/kg}$$

The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{800 \text{ K}} = \mathbf{0.625}$$

Then,

$$W_{net} = \eta_{th} m q_{in} = (0.625)(1 \text{ kg})(462.6 \text{ kJ/kg}) = \mathbf{289.1 kJ}$$

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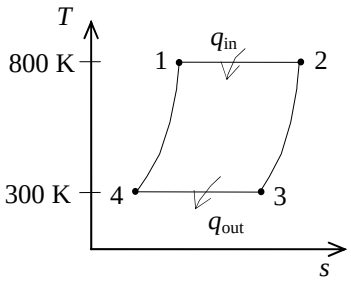
9-76 An ideal Stirling engine with air as the working fluid operates between the specified temperature and pressure limits. The power produced and the rate of heat input are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis Since the specific volume is constant during process 2-3,

$$P_2 = P_3 \frac{T_2}{T_3} = (100 \text{ kPa})\left(\frac{800 \text{ K}}{300 \text{ K}}\right) = 266.7 \text{ kPa}$$



Heat is only added to the system during reversible process 1-2. Then,

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} \\ &= 0 - (0.287 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{266.7 \text{ kPa}}{2000 \text{ kPa}}\right) \\ &= 0.5782 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$q_{\text{in}} = T_1(s_2 - s_1) = (800 \text{ K})(0.5782 \text{ kJ/kg}\cdot\text{K}) = 462.6 \text{ kJ/kg}$$

The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{800 \text{ K}} = 0.625$$

Then,

$$W_{\text{net}} = \eta_{\text{th}} m q_{\text{in}} = (0.625)(1 \text{ kg})(462.6 \text{ kJ/kg}) = 289.1 \text{ kJ}$$

The rate at which heat is added to this engine is

$$\dot{Q}_{\text{in}} = m q_{\text{in}} \dot{n} = (1 \text{ kg/cycle})(462.6 \text{ kJ/kg})(1300/60 \text{ cycle/s}) = \mathbf{10{,}020 \text{ kW}}$$

while the power produced by the engine is

$$\dot{W}_{\text{net}} = W_{\text{net}} \dot{n} =)(289.1 \text{ kJ/cycle})(1300/60 \text{ cycle/s}) = \mathbf{6264 \text{ kW}}$$

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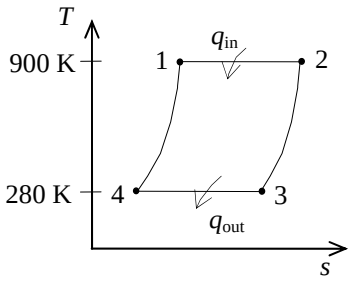
9-77 An ideal Ericsson cycle operates between the specified temperature limits. The rate of heat addition is to be determined.

Analysis The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{280\ \text{K}}{900\ \text{K}} = 0.6889$$

According to the general definition of the thermal efficiency, the rate of heat addition is

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th}}} = \frac{500\ \text{kW}}{0.6889} = \mathbf{726\ kW}$$



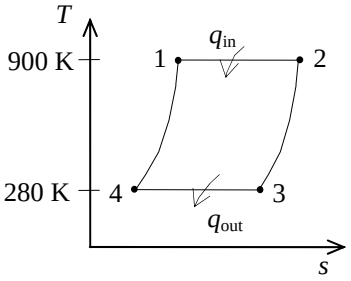
9-78 An ideal Ericsson cycle operates between the specified temperature limits. The power produced by the cycle is to be determined.

Analysis The power output is 500 kW when the cycle is repeated 2000 times per minute. Then the work per cycle is

$$W_{\text{net}} = \frac{\dot{W}_{\text{net}}}{\dot{n}} = \frac{500\ \text{kJ/s}}{(2000/60)\ \text{cycle/s}} = 15\ \text{kJ/cycle}$$

When the cycle is repeated 3000 times per minute, the power output will be

$$\dot{W}_{\text{net}} = \dot{n} W_{\text{net}} = (3000/60\ \text{cycle/s})(15\ \text{kJ/cycle}) = \mathbf{750\ kW}$$



9-79E An ideal Stirling engine with air as the working fluid is considered. The temperature of the source-energy reservoir, the amount of air contained in the engine, and the maximum air pressure during the cycle are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis From the thermal efficiency relation,

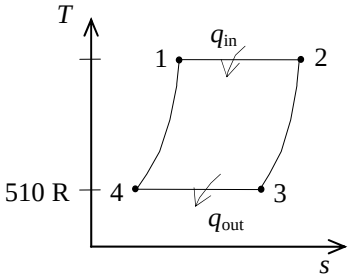
$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = 1 - \frac{T_L}{T_H} \longrightarrow \frac{2\ \text{Btu}}{6\ \text{Btu}} = 1 - \frac{510\ \text{R}}{T_H} \longrightarrow T_H = \mathbf{765\ R}$$

State 3 may be used to determine the mass of air in the system,

$$m = \frac{P_3 V_3}{R T_3} = \frac{(10\ \text{psia})(0.5\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510\ \text{R})} = \mathbf{0.02647\ lbm}$$

The maximum pressure occurs at state 1,

$$P_1 = \frac{mRT_1}{V_1} = \frac{(0.02647\ \text{lbm})(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(765\ \text{R})}{0.06\ \text{ft}^3} = \mathbf{125\ psia}$$



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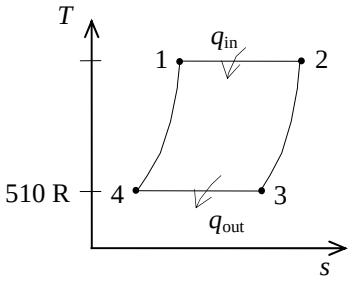
9-80E An ideal Stirling engine with air as the working fluid is considered. The temperature of the source-energy reservoir, the amount of air contained in the engine, and the maximum air pressure during the cycle are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $R = 0.3704$ psia·ft^3/lbm.R, $c_p = 0.240$ Btu/lbm·R, $c_v = 0.171$ Btu/lbm·R, and $k = 1.4$ (Table A-2E).

Analysis From the thermal efficiency relation,

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = 1 - \frac{T_L}{T_H} \longrightarrow \frac{2.5\ \text{Btu}}{6\ \text{Btu}} = 1 - \frac{510\ \text{R}}{T_H} \longrightarrow T_H = \mathbf{874\ R}$$



State 3 may be used to determine the mass of air in the system,

$$m = \frac{P_3 V_3}{RT_3} = \frac{(10\ \text{psia})(0.5\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510\ \text{R})} = \mathbf{0.02647\ lbm}$$

The maximum pressure occurs at state 1,

$$P_1 = \frac{mRT_1}{V_1} = \frac{(0.02647\ \text{lbm})(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(874\ \text{R})}{0.06\ \text{ft}^3} = \mathbf{143\ psia}$$

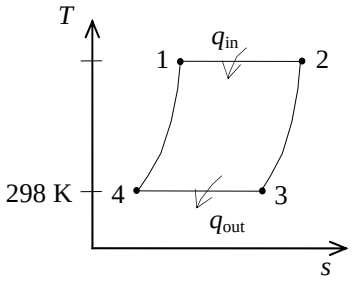
9-81 An ideal Stirling engine with air as the working fluid operates between specified pressure limits. The heat added to the cycle and the net work produced by the cycle are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $R = 0.287$ kPa·m^3/kg.K, $c_p = 1.005$ kJ/kg·K, $c_v = 0.718$ kJ/kg·K, and $k = 1.4$ (Table A-2a).

Analysis Applying the ideal gas equation to the isothermal process 3-4 gives

$$P_4 = P_3 \frac{v_3}{v_4} = (50\ \text{kPa})(12) = 600\ \text{kPa}$$



Since process 4-1 is one of constant volume,

$$T_1 = T_4\left(\frac{P_1}{P_4}\right) = (298\ \text{K})\left(\frac{3600\ \text{kPa}}{600\ \text{kPa}}\right) = 1788\ \text{K}$$

Adapting the first law and work integral to the heat addition process gives

$$q_{\text{in}} = w_{1-2} = RT_1 \ln\frac{v_2}{v_1} = (0.287\ \text{kJ/kg}\cdot\text{K})(1788\ \text{K})\ln(12) = \mathbf{1275\ kJ/kg}$$

Similarly,

$$q_{\text{out}} = w_{3-4} = RT_3 \ln\frac{v_4}{v_3} = (0.287\ \text{kJ/kg}\cdot\text{K})(298\ \text{K})\ln\left(\frac{1}{12}\right) = \mathbf{212.5\ kJ/kg}$$

The net work is then

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1275 - 212.5 = \mathbf{1063\ kJ/kg}$$

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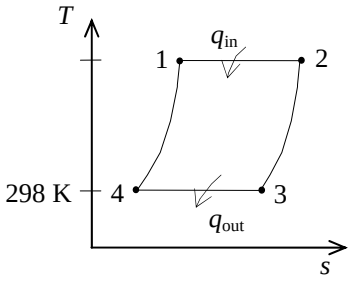
9-82 An ideal Stirling engine with air as the working fluid operates between specified pressure limits. The heat transfer in the regenerator is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg.K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis Applying the ideal gas equation to the isothermal process 3-4 gives

$$P_4 = P_3 \frac{v_3}{v_4} = (50\ \text{kPa})(12) = 600\ \text{kPa}$$



Since process 4-1 is one of constant volume,

$$T_1 = T_4\left(\frac{P_1}{P_4}\right) = (298\ \text{K})\left(\frac{3600\ \text{kPa}}{600\ \text{kPa}}\right) = 1788\ \text{K}$$

Application of the first law to process 4-1 gives

$$q_{\text{regen}} = c_v(T_1 - T_4) = (0.718\ \text{kJ/kg}\cdot\text{K})(1788 - 298)\text{K} = \mathbf{1070\ kJ/kg}$$

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Ideal and Actual Gas-Turbine (Brayton) Cycles

9-83C They are (1) isentropic compression (in a compressor), (2) P = constant heat addition, (3) isentropic expansion (in a turbine), and (4) P = constant heat rejection.

9-84C For fixed maximum and minimum temperatures, (a) the thermal efficiency increases with pressure ratio, (b) the net work first increases with pressure ratio, reaches a maximum, and then decreases.

9-85C Back work ratio is the ratio of the compressor (or pump) work input to the turbine work output. It is usually between 0.40 and 0.6 for gas turbine engines.

9-86C In gas turbine engines a gas is compressed, and thus the compression work requirements are very large since the steady-flow work is proportional to the specific volume.

9-87C As a result of turbine and compressor inefficiencies, (a) the back work ratio increases, and (b) the thermal efficiency decreases.

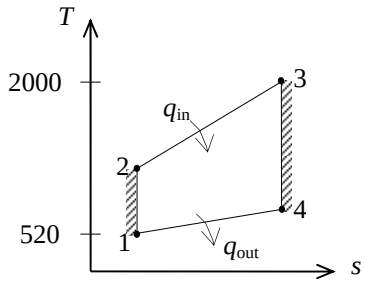
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9-88E A simple ideal Brayton cycle with air as the working fluid has a pressure ratio of 10. The air temperature at the compressor exit, the back work ratio, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-17E.

Analysis (*a*) Noting that process 1-2 is isentropic,



$$T_1 = 520\ \text{R} \longrightarrow \begin{matrix} h_1 = 124.27\ \text{Btu/lbm} \\ P_{r_1} = 1.2147 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.2147) = 12.147 \longrightarrow \begin{matrix} T_2 = \mathbf{996.5\ R} \\ h_2 = 240.11\ \text{Btu/lbm} \end{matrix}$$

(*b*) Process 3-4 is isentropic, and thus

$$T_3 = 2000\ \text{R} \longrightarrow \begin{matrix} h_3 = 504.71\ \text{Btu/lbm} \\ P_{r_3} = 174.0 \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{10}\right)(174.0) = 17.4 \longrightarrow h_4 = 265.83\ \text{Btu/lbm}$$

$$w_{\text{C,in}} = h_2 - h_1 = 240.11 - 124.27 = 115.84\ \text{Btu/lbm}$$

$$w_{\text{T,out}} = h_3 - h_4 = 504.71 - 265.83 = 238.88\ \text{Btu/lbm}$$

Then the back-work ratio becomes

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{115.84\ \text{Btu/lbm}}{238.88\ \text{Btu/lbm}} = \mathbf{48.5\%}$$

(*c*) $$q_{\text{in}} = h_3 - h_2 = 504.71 - 240.11 = 264.60\ \text{Btu/lbm}$$

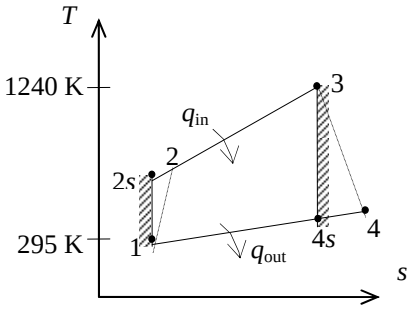
$$w_{\text{net,out}} = w_{\text{T,out}} - w_{\text{C,in}} = 238.88 - 115.84 = 123.04\ \text{Btu/lbm}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{123.04\ \text{Btu/lbm}}{264.60\ \text{Btu/lbm}} = \mathbf{46.5\%}$$

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9-89 A simple Brayton cycle with air as the working fluid has a pressure ratio of 10. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.



Properties The properties of air are given in Table A-17.

Analysis (*a*) Noting that process 1-2s is isentropic,

$$T_1 = 295\text{ K} \longrightarrow \begin{matrix} h_1 = 295.17\text{ kJ/kg} \\ P_{r_1} = 1.3068 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.3068) = 13.07 \longrightarrow h_{2s} = 570.26\text{ kJ/kg} \text{ and } T_{2s} = 564.9\text{ K}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C}$$

$$= 295.17 + \frac{570.26 - 295.17}{0.83} = 626.60\text{ kJ/kg}$$

$$T_3 = 1240\text{ K} \longrightarrow \begin{matrix} h_3 = 1324.93\text{ kJ/kg} \\ P_{r_3} = 272.3 \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{10}\right)(272.3) = 27.23 \longrightarrow h_{4s} = 702.07\text{ kJ/kg} \text{ and } T_{4s} = 689.6\text{ K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

$$= 1324.93 - (0.87)(1324.93 - 702.07)$$

$$= 783.04\text{ kJ/kg}$$

Thus,

$$T_4 = \mathbf{764.4\text{ K}}$$

(*b*) $$q_{in} = h_3 - h_2 = 1324.93 - 626.60 = 698.3\text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = 783.04 - 295.17 = 487.9\text{ kJ/kg}$$

$$w_{net,out} = q_{in} - q_{out} = 698.3 - 487.9 = \mathbf{210.4\text{ kJ/kg}}$$

(*c*) $$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{210.4\text{ kJ/kg}}{698.3\text{ kJ/kg}} = 0.3013 = \mathbf{30.1\%}$$

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9-90 Problem 9-89 is reconsidered. The mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic efficiencies of the turbine and compressor are to be varied and a general solution for the problem by taking advantage of the diagram window method for supplying data to EES is to be developed.

Analysis Using EES, the problem is solved as follows:

```
"Input data - from diagram window"
{P_ratio = 10}
{T[1] = 295 [K]
P[1]= 100 [kPa]
T[3] = 1240 [K]
m_dot = 20 [kg/s]
Eta_c = 83/100
Eta_t = 87/100}

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c = W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]  "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0"

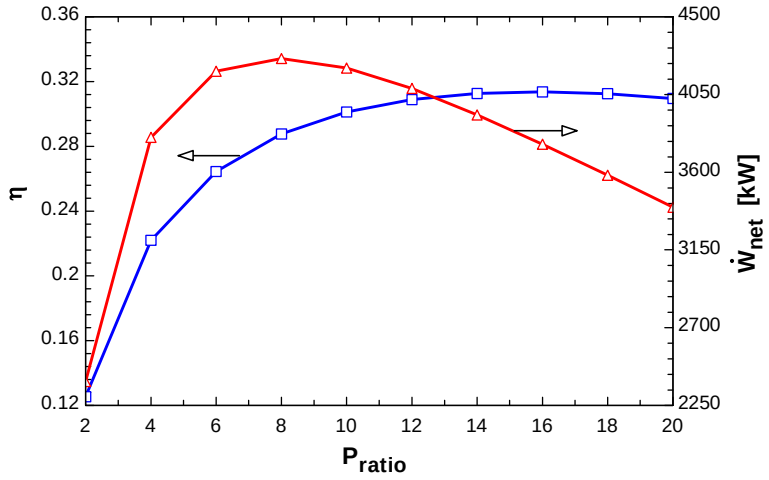
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency, Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"

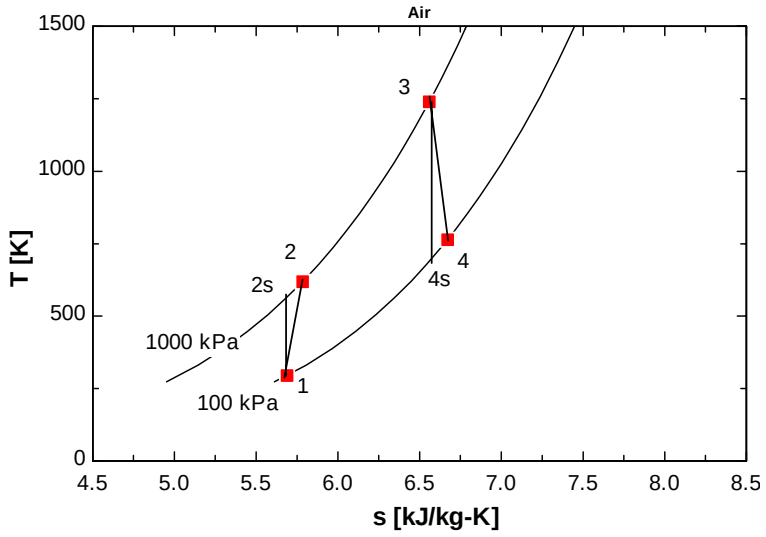
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature(air,h=h[2])
T[4]=temperature(air,h=h[4])
s[2]=entropy(air,T=T[2],P=P[2])
s[4]=entropy(air,T=T[4],P=P[4])
```

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Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.5229	0.1	2	1818	1659	3477	16587
0.6305	0.1644	4	4033	2364	6396	14373
0.7038	0.1814	6	5543	2333	7876	12862
0.7611	0.1806	8	6723	2110	8833	11682
0.8088	0.1702	10	7705	1822	9527	10700
0.85	0.1533	12	8553	1510	10063	9852
0.8864	0.131	14	9304	1192	10496	9102
0.9192	0.1041	16	9980	877.2	10857	8426
0.9491	0.07272	18	10596	567.9	11164	7809
0.9767	0.03675	20	11165	266.1	11431	7241





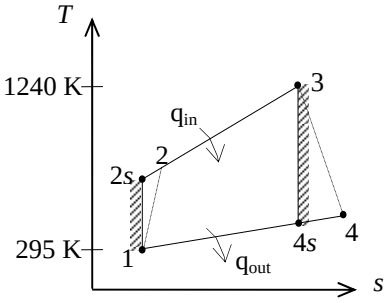
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9-91 A simple Brayton cycle with air as the working fluid has a pressure ratio of 10. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2).

Analysis (*a*) Using the compressor and turbine efficiency relations,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (295\text{ K})(10)^{0.4/1.4} = 569.6\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1240\text{ K})\left(\frac{1}{10}\right)^{0.4/1.4} = 642.3\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta_C}$$

$$= 295 + \frac{569.6-295}{0.83} = 625.8\text{ K}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} = \frac{c_p(T_3-T_4)}{c_p(T_3-T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3-T_{4s})$$

$$= 1240 - (0.87)(1240-642.3)$$

$$= \mathbf{720\text{ K}}$$

(*b*) $$q_{\text{in}} = h_3 - h_2 = c_p(T_3-T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(1240-625.8)\text{K} = 617.3\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = c_p(T_4-T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(720-295)\text{K} = 427.1\text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 617.3 - 427.1 = \mathbf{190.2\text{ kJ/kg}}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{190.2\text{ kJ/kg}}{617.3\text{ kJ/kg}} = 0.3081 = \mathbf{30.8\%}$$

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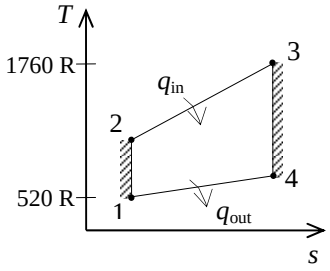
9-92E A simple ideal Brayton cycle with helium has a pressure ratio of 14. The power output is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Helium is an ideal gas with constant specific heats.

Properties The properties of helium are c_p = 1.25 Btu/lbm·R and k = 1.667 (Table A-2Ea).

Analysis Using the isentropic relations for an ideal gas,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1 r_p^{(k-1)/k} = (520\text{ R})(14)^{0.667/1.667} = 1495\text{ R}$$



Similarly,

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1760\text{ R})\left(\frac{1}{14}\right)^{0.667/1.667} = 612.2\text{ K}$$

Applying the first law to the constant-pressure heat addition process 2-3 produces

$$q_{in} = c_p(T_3 - T_2) = (1.25\text{ Btu/lbm}\cdot\text{R})(1760 - 1495)\text{R} = 331.3\text{ Btu/lbm}$$

Similarly,

$$q_{out} = c_p(T_4 - T_1) = (1.25\text{ Btu/lbm}\cdot\text{R})(612.2 - 520)\text{R} = 115.3\text{ Btu/lbm}$$

The net work production is then

$$w_{net} = q_{in} - q_{out} = 331.3 - 115.3 = 216.0\text{ Btu/lbm}$$

and

$$\dot{W}_{net} = \dot{m}w_{net} = (100\text{ lbm/min})(216.0\text{ Btu/lbm})\left(\frac{1\text{ hp}}{42.41\text{ Btu/min}}\right) = \mathbf{509.3\text{ hp}}$$

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9-93E A simple Brayton cycle with helium has a pressure ratio of 14. The power output is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Helium is an ideal gas with constant specific heats.

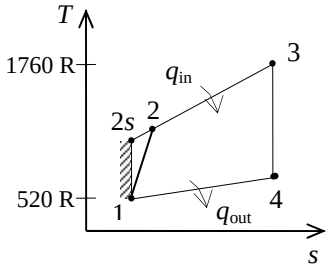
Properties The properties of helium at room temperature are c_p = 1.25 Btu/lbm·R and k = 1.667 (Table A-2Ea).

Analysis For the compression process,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1 r_p^{(k-1)/k} = (520\text{ R})(14)^{0.667/1.667} = 1495\text{ R}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta_C}$$

$$= 520 + \frac{1495-520}{0.95} = 1546\text{ R}$$



For the isentropic expansion process,

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1760\text{ R})\left(\frac{1}{14}\right)^{0.667/1.667} = 612.2\text{ R}$$

Applying the first law to the constant-pressure heat addition process 2-3 produces

$$q_{in} = c_p(T_3 - T_2) = (1.25\text{ Btu/lbm}\cdot\text{R})(1760-1546)\text{R} = 267.5\text{ Btu/lbm}$$

Similarly,

$$q_{out} = c_p(T_4 - T_1) = (1.25\text{ Btu/lbm}\cdot\text{R})(612.2-520)\text{R} = 115.3\text{ Btu/lbm}$$

The net work production is then

$$w_{net} = q_{in} - q_{out} = 267.5 - 115.3 = 152.2\text{ Btu/lbm}$$

and

$$\dot{W}_{net} = \dot{m}w_{net} = (100\text{ lbm/min})(152.2\text{ Btu/lbm})\left(\frac{1\text{ hp}}{42.41\text{ Btu/min}}\right) = \mathbf{358.9\text{ hp}}$$

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9-94 A simple Brayton cycle with air as the working fluid operates between the specified temperature and pressure limits. The effects of non-isentropic compressor and turbine on the back-work ratio is to be compared.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

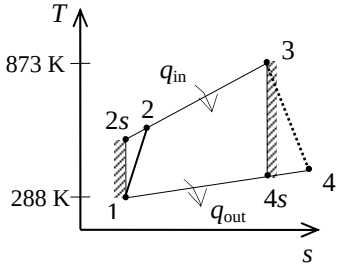
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis For the compression process,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (288\,\text{K})(12)^{0.4/1.4} = 585.8\,\text{K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta_C}$$

$$= 288 + \frac{585.8-288}{0.90} = 618.9\,\text{K}$$



For the expansion process,

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (873\,\text{K})\left(\frac{1}{12}\right)^{0.4/1.4} = 429.2\,\text{K}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} = \frac{c_p(T_3-T_4)}{c_p(T_3-T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3-T_{4s})$$

$$= 873-(0.90)(873-429.2)$$

$$= 473.6\,\text{K}$$

The isentropic and actual work of compressor and turbine are

$$W_{\text{Comp},s} = c_p(T_{2s}-T_1) = (1.005\,\text{kJ/kg}\cdot\text{K})(585.8-288)\text{K} = 299.3\,\text{kJ/kg}$$

$$W_{\text{Comp}} = c_p(T_2-T_1) = (1.005\,\text{kJ/kg}\cdot\text{K})(618.9-288)\text{K} = 332.6\,\text{kJ/kg}$$

$$W_{\text{Turb},s} = c_p(T_3-T_{4s}) = (1.005\,\text{kJ/kg}\cdot\text{K})(873-429.2)\text{K} = 446.0\,\text{kJ/kg}$$

$$W_{\text{Turb}} = c_p(T_3-T_4) = (1.005\,\text{kJ/kg}\cdot\text{K})(873-473.6)\text{K} = 401.4\,\text{kJ/kg}$$

The back work ratio for 90% efficient compressor and isentropic turbine case is

$$r_{\text{bw}} = \frac{W_{\text{Comp}}}{W_{\text{Turb},s}} = \frac{332.6\,\text{kJ/kg}}{446.0\,\text{kJ/kg}} = \mathbf{0.7457}$$

The back work ratio for 90% efficient turbine and isentropic compressor case is

$$r_{\text{bw}} = \frac{W_{\text{Comp},s}}{W_{\text{Turb}}} = \frac{299.3\,\text{kJ/kg}}{401.4\,\text{kJ/kg}} = \mathbf{0.7456}$$

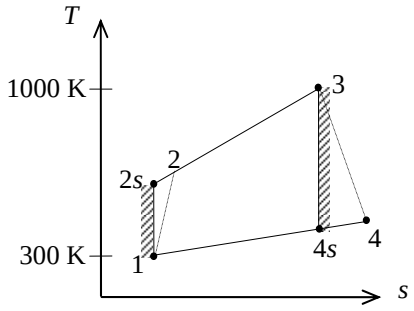
The two results are almost identical.

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9-95 A gas turbine power plant that operates on the simple Brayton cycle with air as the working fluid has a specified pressure ratio. The required mass flow rate of air is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2).



Analysis (*a*) Using the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(12)^{0.4/1.4} = 610.2\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1000\text{ K})\left(\frac{1}{12}\right)^{0.4/1.4} = 491.7\text{ K}$$

$$w_{s,C,in} = h_{2s} - h_1 = c_p(T_{2s} - T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(610.2 - 300)\text{K} = 311.75\text{ kJ/kg}$$

$$w_{s,T,out} = h_3 - h_{4s} = c_p(T_3 - T_{4s}) = (1.005\text{ kJ/kg}\cdot\text{K})(1000 - 491.7)\text{K} = 510.84\text{ kJ/kg}$$

$$w_{s,net,out} = w_{s,T,out} - w_{s,C,in} = 510.84 - 311.75 = 199.1\text{ kJ/kg}$$

$$\dot{m}_s = \frac{\dot{W}_{net,out}}{w_{s,net,out}} = \frac{70{,}000\text{ kJ/s}}{199.1\text{ kJ/kg}} = \mathbf{352\ \ kg/s}$$

(*b*) The net work output is determined to be

$$w_{a,net,out} = w_{a,T,out} - w_{a,C,in} = \eta_T w_{s,T,out} - w_{s,C,in}/\eta_C$$
$$= (0.85)(510.84) - 311.75/0.85 = 67.5\text{ kJ/kg}$$

$$\dot{m}_a = \frac{\dot{W}_{net,out}}{w_{a,net,out}} = \frac{70{,}000\text{ kJ/s}}{67.5\text{ kJ/kg}} = \mathbf{1037\ \ kg/s}$$

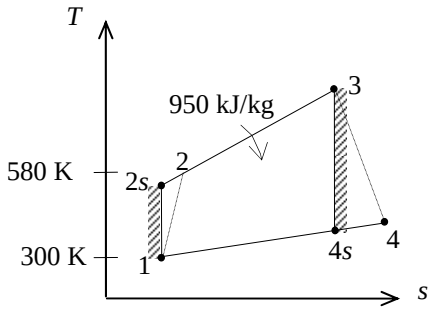
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9-96 An actual gas-turbine power plant operates at specified conditions. The fraction of the turbine work output used to drive the compressor and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-17.

Analysis (a) Using the isentropic relations,



$$T_1 = 300\ \text{K} \longrightarrow h_1 = 300.19\ \text{kJ/kg}$$

$$T_2 = 580\ \text{K} \longrightarrow h_2 = 586.04\ \text{kJ/kg}$$

$$r_p = \frac{P_2}{P_1} = \frac{700}{100} = 7$$

$$q_{\text{in}} = h_3 - h_2 \longrightarrow h_3 = 950 + 586.04 = 1536.04\text{kJ/kg}$$

$$\rightarrow P_{r_3} = 474.11$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{7}\right)(474.11) = 67.73 \longrightarrow h_{4s} = 905.83\ \text{kJ/kg}$$

$$w_{\text{C,in}} = h_2 - h_1 = 586.04 - 300.19 = 285.85\ \text{kJ/kg}$$

$$w_{\text{T,out}} = \eta_T (h_3 - h_{4s}) = (0.86)(1536.04 - 905.83) = 542.0\ \text{kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{285.85\ \text{kJ/kg}}{542.0\ \text{kJ/kg}} = \mathbf{52.7\%}$$

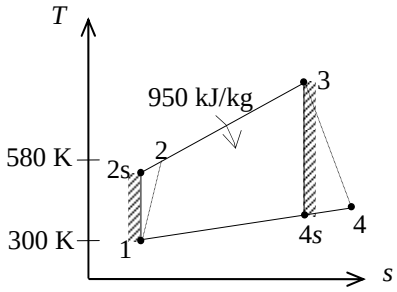
(b) $$w_{\text{net.out}} = w_{\text{T,out}} - w_{\text{C,in}} = 542.0 - 285.85 = 256.15\ \text{kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{256.15\ \text{kJ/kg}}{950\ \text{kJ/kg}} = \mathbf{27.0\%}$$

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9-97 A gas-turbine power plant operates at specified conditions. The fraction of the turbine work output used to drive the compressor and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.



Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2).

Analysis (*a*) Using constant specific heats,

$$r_p = \frac{P_2}{P_1} = \frac{700}{100} = 7$$

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2) \longrightarrow T_3 = T_2 + q_{\text{in}}/c_p$$
$$= 580\ \text{K} + (950\ \text{kJ/kg})/(1.005\ \text{kJ/kg}\cdot\text{K})$$
$$= 1525.3\ \text{K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1525.3\ \text{K})\left(\frac{1}{7}\right)^{0.4/1.4} = 874.8\ \text{K}$$

$$w_{\text{C,in}} = h_2 - h_1 = c_p(T_2 - T_1) = (1.005\text{kJ/kg}\cdot\text{K})(580 - 300)\text{K} = 281.4\ \text{kJ/kg}$$

$$w_{\text{T,out}} = \eta_T(h_3 - h_{4s}) = \eta_T c_p(T_3 - T_{4s}) = (0.86)(1.005\ \text{kJ/kg}\cdot\text{K})(1525.3 - 874.8)\text{K} = 562.2\ \text{kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{281.4\ \text{kJ/kg}}{562.2\ \text{kJ/kg}} = \mathbf{50.1\%}$$

(b) $$w_{\text{net,out}} = w_{\text{T,out}} - w_{\text{C,in}} = 562.2 - 281.4 = 280.8\ \text{kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{280.8\ \text{kJ/kg}}{950\ \text{kJ/kg}} = \mathbf{29.6\%}$$

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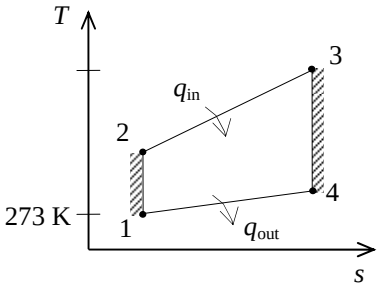
9-98 An aircraft engine operates as a simple ideal Brayton cycle with air as the working fluid. The pressure ratio and the rate of heat input are given. The net power and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis For the isentropic compression process,

$$T_2 = T_1 r_p^{(k-1)/k} = (273\,\text{K})(10)^{0.4/1.4} = 527.1\,\text{K}$$



The heat addition is

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = \frac{500\,\text{kW}}{1\,\text{kg/s}} = 500\,\text{kJ/kg}$$

Applying the first law to the heat addition process,

$$q_{in} = c_p(T_3 - T_2)$$

$$T_3 = T_2 + \frac{q_{in}}{c_p} = 527.1\,\text{K} + \frac{500\,\text{kJ/kg}}{1.005\,\text{kJ/kg}\cdot\text{K}} = 1025\,\text{K}$$

The temperature at the exit of the turbine is

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1025\,\text{K})\left(\frac{1}{10}\right)^{0.4/1.4} = 530.9\,\text{K}$$

Applying the first law to the adiabatic turbine and the compressor produce

$$w_T = c_p(T_3 - T_4) = (1.005\,\text{kJ/kg}\cdot\text{K})(1025 - 530.9)\text{K} = 496.6\,\text{kJ/kg}$$

$$w_C = c_p(T_2 - T_1) = (1.005\,\text{kJ/kg}\cdot\text{K})(527.1 - 273)\text{K} = 255.4\,\text{kJ/kg}$$

The net power produced by the engine is then

$$\dot{W}_{net} = \dot{m}(w_T - w_C) = (1\,\text{kg/s})(496.6 - 255.4)\text{kJ/kg} = \mathbf{241.2\,kW}$$

Finally the thermal efficiency is

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{241.2\,\text{kW}}{500\,\text{kW}} = \mathbf{0.482}$$

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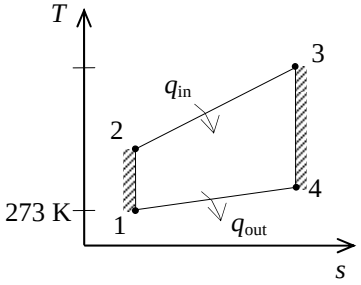
9-99 An aircraft engine operates as a simple ideal Brayton cycle with air as the working fluid. The pressure ratio and the rate of heat input are given. The net power and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis For the isentropic compression process,

$$T_2 = T_1 r_p^{(k-1)/k} = (273\text{ K})(15)^{0.4/1.4} = 591.8\text{ K}$$



The heat addition is

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = \frac{500\text{ kW}}{1\text{ kg/s}} = 500\text{ kJ/kg}$$

Applying the first law to the heat addition process,

$$q_{in} = c_p(T_3 - T_2)$$

$$T_3 = T_2 + \frac{q_{in}}{c_p} = 591.8\text{ K} + \frac{500\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 1089\text{ K}$$

The temperature at the exit of the turbine is

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1089\text{ K})\left(\frac{1}{15}\right)^{0.4/1.4} = 502.3\text{ K}$$

Applying the first law to the adiabatic turbine and the compressor produce

$$w_T = c_p(T_3 - T_4) = (1.005\text{ kJ/kg}\cdot\text{K})(1089 - 502.3)\text{K} = 589.6\text{ kJ/kg}$$

$$w_C = c_p(T_2 - T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(591.8 - 273)\text{K} = 320.4\text{ kJ/kg}$$

The net power produced by the engine is then

$$\dot{W}_{net} = \dot{m}(w_T - w_C) = (1\text{ kg/s})(589.6 - 320.4)\text{kJ/kg} = \mathbf{269.2\text{ kW}}$$

Finally the thermal efficiency is

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{269.2\text{ kW}}{500\text{ kW}} = \mathbf{0.538}$$

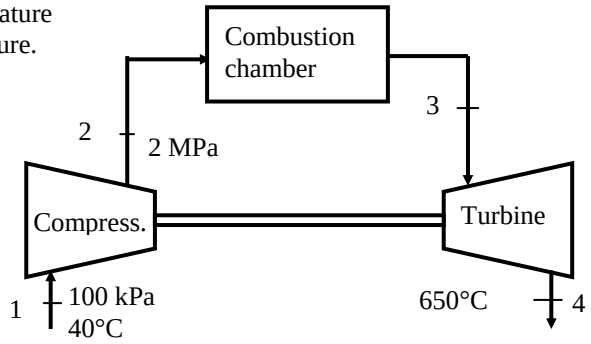
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9-100 A gas-turbine plant operates on the simple Brayton cycle. The net power output, the back work ratio, and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is R = 0.287 kJ/kg·K (Table A-1).

Analysis (*a*) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.



Process 1-2: Compression

$$T_1 = 40°\text{C} \longrightarrow h_1 = 313.6\ \text{kJ/kg}$$

$$\left.\begin{matrix} T_1 = 40°\text{C} \\ P_1 = 100\ \text{kPa} \end{matrix}\right\} s_1 = 5.749\ \text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{matrix} P_2 = 2000\ \text{kPa} \\ s_2 = s_1 = 5.749\ \text{kJ/kg.K} \end{matrix}\right\} h_{2s} = 736.7\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.85 = \frac{736.7 - 313.6}{h_2 - 313.6} \longrightarrow h_2 = 811.4\ \text{kJ/kg}$$

Process 3-4: Expansion

$$T_4 = 650°\text{C} \longrightarrow h_4 = 959.2\ \text{kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.88 = \frac{h_3 - 959.2}{h_3 - h_{4s}}$$

We cannot find the enthalpy at state 3 directly. However, using the following lines in EES together with the isentropic efficiency relation, we find h_3 = 1873 kJ/kg, T_3 = 1421ºC, s_3 = 6.736 kJ/kg.K. The solution by hand would require a trial-error approach.

```
h_3=enthalpy(Air, T=T_3)
s_3=entropy(Air, T=T_3, P=P_2)
h_4s=enthalpy(Air, P=P_1, s=s_3)
```

The mass flow rate is determined from

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(100\ \text{kPa})(700/60\ \text{m}^3/\text{s})}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(40 + 273\ \text{K})} = 12.99\ \text{kg/s}$$

The net power output is

$$\dot{W}_{C,in} = \dot{m}(h_2 - h_1) = (12.99\ \text{kg/s})(811.4 - 313.6)\text{kJ/kg} = 6464\ \text{kW}$$

$$\dot{W}_{T,out} = \dot{m}(h_3 - h_4) = (12.99\ \text{kg/s})(1873 - 959.2)\text{kJ/kg} = 11{,}868\ \text{kW}$$

$$\dot{W}_{net} = \dot{W}_{T,out} - \dot{W}_{C,in} = 11{,}868 - 6464 = \mathbf{5404\ kW}$$

(*b*) The back work ratio is

$$r_{bw} = \frac{\dot{W}_{C,in}}{\dot{W}_{T,out}} = \frac{6464\ \text{kW}}{11{,}868\ \text{kW}} = \mathbf{0.545}$$

(*c*) The rate of heat input and the thermal efficiency are

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2) = (12.99\ \text{kg/s})(1873 - 811.4)\text{kJ/kg} = 13{,}788\ \text{kW}$$

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{5404\ \text{kW}}{13{,}788\ \text{kW}} = 0.392 = \mathbf{39.2\%}$$

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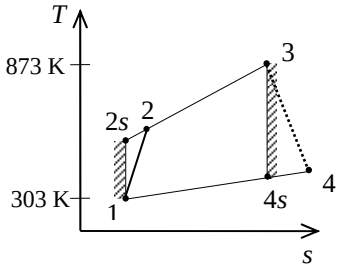
9-101 A simple Brayton cycle with air as the working fluid operates between the specified temperature and pressure limits. The cycle is to be sketched on the *T*-*s* cycle and the isentropic efficiency of the turbine and the cycle thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air are given as c_v = 0.718 kJ/kg·K, c_p = 1.005 kJ/kg·K, R = 0.287 kJ/kg·K, k = 1.4.

Analysis (b) For the compression process,

$$\begin{aligned}\dot{W}_{\text{Comp}} &= \dot{m}c_p(T_2 - T_1)\\ &= (200\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})(330-30)\text{K}\\ &= 60{,}300\ \text{kW}\end{aligned}$$



For the turbine during the isentropic process,

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1400\ \text{K})\left(\frac{100\ \text{kPa}}{800\ \text{kPa}}\right)^{0.4/1.4} = 772.9\ \text{K}$$

$$\dot{W}_{\text{Turb,s}} = \dot{m}c_p(T_3 - T_{4s}) = (200\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})(1400-772.9)\text{K} = 126{,}050\ \text{kW}$$

The actual power output from the turbine is

$$\dot{W}_{\text{net}} = \dot{W}_{\text{Turb}} - \dot{W}_{\text{Comp}}$$
$$\dot{W}_{\text{Turb}} = \dot{W}_{\text{net}} + \dot{W}_{\text{Turb}} = 60{,}000 + 60{,}300 = 120{,}300\ \text{kW}$$

The isentropic efficiency of the turbine is then

$$\eta_{\text{Turb}} = \frac{\dot{W}_{\text{Turb}}}{\dot{W}_{\text{Turb,s}}} = \frac{120{,}300\ \text{kW}}{126{,}050\ \text{kW}} = 0.954 = \mathbf{95.4\%}$$

(c) The rate of heat input is

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_3 - T_2) = (200\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})[(1400-(330+273)]\text{K} = 160{,}200\ \text{kW}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{60{,}000\ \text{kW}}{160{,}200\ \text{kW}} = 0.375 = \mathbf{37.5\%}$$

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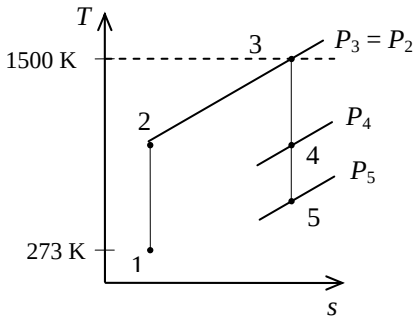
9-102 A modified Brayton cycle with air as the working fluid operates at a specified pressure ratio. The *T-s* diagram is to be sketched and the temperature and pressure at the exit of the high-pressure turbine and the mass flow rate of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air are given as c_v = 0.718 kJ/kg·K, c_p = 1.005 kJ/kg·K, R = 0.287 kJ/kg·K, k = 1.4.

Analysis (b) For the compression process,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (273\,\text{K})(8)^{0.4/1.4} = 494.5\,\text{K}$$



The power input to the compressor is equal to the power output from the high-pressure turbine. Then,

$$\dot{W}_{\text{Comp,in}} = \dot{W}_{\text{HP Turb,out}}$$
$$\dot{m}c_p(T_2 - T_1) = \dot{m}c_p(T_3 - T_4)$$
$$T_2 - T_1 = T_3 - T_4$$
$$T_4 = T_3 + T_1 - T_2 = 1500 + 273 - 494.5 = \mathbf{1278.5\,K}$$

The pressure at this state is

$$\frac{P_4}{P_3} = \left(\frac{T_4}{T_3}\right)^{k/(k-1)} \longrightarrow P_4 = rP_1\left(\frac{T_4}{T_3}\right)^{k/(k-1)} = 8(100\,\text{kPa})\left(\frac{1278.5\,\text{K}}{1500\,\text{K}}\right)^{1.4/0.4} = \mathbf{457.3\,kPa}$$

(c) The temperature at state 5 is determined from

$$T_5 = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = (1278.5\,\text{K})\left(\frac{100\,\text{kPa}}{457.3\,\text{kPa}}\right)^{0.4/1.4} = 828.1\,\text{K}$$

The net power is that generated by the low-pressure turbine since the power output from the high-pressure turbine is equal to the power input to the compressor. Then,

$$\dot{W}_{\text{LP Turb}} = \dot{m}c_p(T_4 - T_5)$$
$$\dot{m} = \frac{\dot{W}_{\text{LP Turb}}}{c_p(T_4 - T_5)} = \frac{200{,}000\,\text{kW}}{(1.005\,\text{kJ/kg}\cdot\text{K})(1278.5 - 828.1)\text{K}} = \mathbf{441.8\,kg/s}$$

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9-103 A simple Brayton cycle with air as the working fluid operates at a specified pressure ratio and between the specified temperature and pressure limits. The cycle is to be sketched on the *T-s* cycle and the volume flow rate of the air into the compressor is to be determined. Also, the effect of compressor inlet temperature on the mass flow rate and the net power output are to be investigated.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

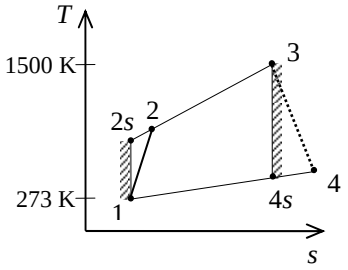
Properties The properties of air are given as c_v = 0.718 kJ/kg·K, c_p = 1.005 kJ/kg·K, R = 0.287 kJ/kg·K, k = 1.4.

Analysis (b) For the compression process,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (273\,\text{K})(7)^{0.4/1.4} = 476.0\,\text{K}$$

$$\eta_{\text{Comp}} = \frac{\dot{W}_{\text{Comp,s}}}{\dot{W}_{\text{Comp}}} = \frac{\dot{m}c_p(T_{2s}-T_1)}{\dot{m}c_p(T_2-T_1)} = \frac{T_{2s}-T_1}{T_2-T_1}$$

$$0.80 = \frac{476.0-273}{T_2-273} \longrightarrow T_2 = 526.8\,\text{K}$$



For the expansion process,

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1500\,\text{K})\left(\frac{1}{7}\right)^{0.4/1.4} = 860.3\,\text{K}$$

$$\eta_{\text{Turb}} = \frac{\dot{W}_{\text{Turb}}}{\dot{W}_{\text{Turb,s}}} = \frac{\dot{m}c_p(T_3-T_4)}{\dot{m}c_p(T_3-T_{4s})} = \frac{T_3-T_4}{T_3-T_{4s}}$$

$$0.90 = \frac{1500-T_4}{1500-860.3} \longrightarrow T_4 = 924.3\,\text{K}$$

Given the net power, the mass flow rate is determined from

$$\dot{W}_{\text{net}} = \dot{W}_{\text{Turb}} - \dot{W}_{\text{Comp}} = \dot{m}c_p(T_3-T_4) - \dot{m}c_p(T_2-T_1)$$

$$\dot{W}_{\text{net}} = \dot{m}c_p\left[(T_3-T_4)-(T_2-T_1)\right]$$

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{c_p\left[(T_3-T_4)-(T_2-T_1)\right]}$$

$$= \frac{150{,}000\,\text{kW}}{(1.005\,\text{kJ/kg}\cdot\text{K})\left[(1500-924.3)-(526.8-273)\right]}$$

$$= 463.7\,\text{kg/s}$$

The specific volume and the volume flow rate at the inlet of the compressor are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\,\text{kJ/kg}\cdot\text{K})(273\,\text{K})}{100\,\text{kPa}} = 0.7835\,\text{m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}v_1 = (463.7\,\text{kg/s})(0.7835\,\text{m}^3/\text{kg}) = \mathbf{363.2\,m^3/s}$$

(c) For a fixed compressor inlet velocity and flow area, when the compressor inlet temperature increases, the specific volume increases since $v = \frac{RT}{P}$. When specific volume increases, the mass flow rate decreases since $\dot{m} = \frac{\dot{V}}{v}$. Note that volume flow rate is the same since inlet velocity and flow area are fixed ($\dot{V} = AV$). When mass flow rate decreases, the net power decreases since $\dot{W}_{\text{net}} = \dot{m}(w_{\text{Turb}} - w_{\text{Comp}})$. Therefore, when the inlet temperature increases, both mass flow rate and the net power decrease.

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Brayton Cycle with Regeneration

9-104C Regeneration increases the thermal efficiency of a Brayton cycle by capturing some of the waste heat from the exhaust gases and preheating the air before it enters the combustion chamber.

9-105C Yes. At very high compression ratios, the gas temperature at the turbine exit may be lower than the temperature at the compressor exit. Therefore, if these two streams are brought into thermal contact in a regenerator, heat will flow to the exhaust gases instead of from the exhaust gases. As a result, the thermal efficiency will decrease.

9-106C The extent to which a regenerator approaches an ideal regenerator is called the effectiveness ε, and is defined as $\varepsilon = q_{\text{regen, act}} / q_{\text{regen, max}}$.

9-107C (b) turbine exit.

9-108C The steam injected increases the mass flow rate through the turbine and thus the power output. This, in turn, increases the thermal efficiency since $\eta = W / Q_{\text{in}}$ and W increases while Q_{in} remains constant. Steam can be obtained by utilizing the hot exhaust gases.

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9-109 A Brayton cycle with regeneration produces 150 kW power. The rates of heat addition and rejection are to be determined.

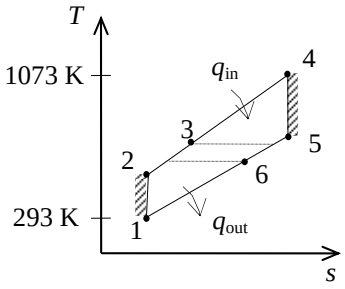
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a).

Analysis According to the isentropic process expressions for an ideal gas,

$$T_2 = T_1 r_p^{(k-1)/k} = (293\,\text{K})(8)^{0.4/1.4} = 530.8\,\text{K}$$

$$T_5 = T_4\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1073\,\text{K})\left(\frac{1}{8}\right)^{0.4/1.4} = 592.3\,\text{K}$$



When the first law is applied to the heat exchanger, the result is

$$T_3 - T_2 = T_5 - T_6$$

while the regenerator temperature specification gives

$$T_3 = T_5 - 10 = 592.3 - 10 = 582.3\,\text{K}$$

The simultaneous solution of these two results gives

$$T_6 = T_5 - (T_3 - T_2) = 592.3 - (582.3 - 530.8) = 540.8\,\text{K}$$

Application of the first law to the turbine and compressor gives

$$\begin{aligned} w_{\text{net}} &= c_p(T_4 - T_5) - c_p(T_2 - T_1) \\ &= (1.005\,\text{kJ/kg}\cdot\text{K})(1073 - 592.3)\,\text{K} - (1.005\,\text{kJ/kg}\cdot\text{K})(530.8 - 293)\,\text{K} \\ &= 244.1\,\text{kJ/kg} \end{aligned}$$

Then,

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150\,\text{kW}}{244.1\,\text{kJ/kg}} = 0.6145\,\text{kg/s}$$

Applying the first law to the combustion chamber produces

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_4 - T_3) = (0.6145\,\text{kg/s})(1.005\,\text{kJ/kg}\cdot\text{K})(1073 - 582.3)\text{K} = \mathbf{303.0\ kW}$$

Similarly,

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_6 - T_1) = (0.6145\,\text{kg/s})(1.005\,\text{kJ/kg}\cdot\text{K})(540.8 - 293)\text{K} = \mathbf{153.0\ kW}$$

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9-110 A Brayton cycle with regeneration produces 150 kW power. The rates of heat addition and rejection are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

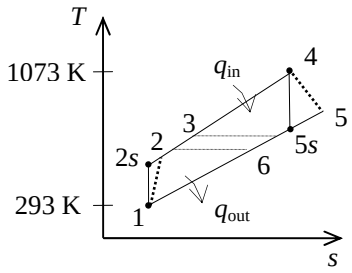
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a).

Analysis For the compression and expansion processes we have

$$T_{2s} = T_1 r_p^{(k-1)/k} = (293\,\text{K})(8)^{0.4/1.4} = 530.8\,\text{K}$$

$$\eta_C = \frac{c_p(T_{2s} - T_1)}{c_p(T_2 - T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_C}$$

$$= 293 + \frac{530.8 - 293}{0.87} = 566.3\,\text{K}$$



$$T_{5s} = T_4\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1073\,\text{K})\left(\frac{1}{8}\right)^{0.4/1.4} = 592.3\,\text{K}$$

$$\eta_T = \frac{c_p(T_4 - T_5)}{c_p(T_4 - T_{5s})} \longrightarrow T_5 = T_4 - \eta_T(T_4 - T_{5s})$$

$$= 1073 - (0.93)(1073 - 592.3)$$

$$= 625.9\,\text{K}$$

When the first law is applied to the heat exchanger, the result is

$$T_3 - T_2 = T_5 - T_6$$

while the regenerator temperature specification gives

$$T_3 = T_5 - 10 = 625.9 - 10 = 615.9\,\text{K}$$

The simultaneous solution of these two results gives

$$T_6 = T_5 - (T_3 - T_2) = 625.9 - (615.9 - 566.3) = 576.3\,\text{K}$$

Application of the first law to the turbine and compressor gives

$$w_{\text{net}} = c_p(T_4 - T_5) - c_p(T_2 - T_1)$$

$$= (1.005\,\text{kJ/kg}\cdot\text{K})(1073 - 625.9)\,\text{K} - (1.005\,\text{kJ/kg}\cdot\text{K})(566.3 - 293)\,\text{K}$$

$$= 174.7\,\text{kJ/kg}$$

Then,

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150\,\text{kW}}{174.7\,\text{kJ/kg}} = 0.8586\,\text{kg/s}$$

Applying the first law to the combustion chamber produces

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_4 - T_3) = (0.8586\,\text{kg/s})(1.005\,\text{kJ/kg}\cdot\text{K})(1073 - 615.9)\text{K} = \mathbf{394.4\ kW}$$

Similarly,

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_6 - T_1) = (0.8586\,\text{kg/s})(1.005\,\text{kJ/kg}\cdot\text{K})(576.3 - 293)\text{K} = \mathbf{244.5\ kW}$$

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9-111 A Brayton cycle with regeneration is considered. The thermal efficiencies of the cycle for parallel-flow and counter-flow arrangements of the regenerator are to be compared.

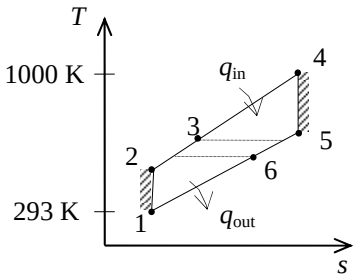
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis According to the isentropic process expressions for an ideal gas,

$$T_2 = T_1 r_p^{(k-1)/k} = (293\text{ K})(7)^{0.4/1.4} = 510.9\text{ K}$$

$$T_5 = T_4\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1000\text{ K})\left(\frac{1}{7}\right)^{0.4/1.4} = 573.5\text{ K}$$



When the first law is applied to the heat exchanger as originally arranged, the result is

$$T_3 - T_2 = T_5 - T_6$$

while the regenerator temperature specification gives

$$T_3 = T_5 - 6 = 573.5 - 6 = 567.5\text{ K}$$

The simultaneous solution of these two results gives

$$T_6 = T_5 - T_3 + T_2 = 573.5 - 567.5 + 510.9 = 516.9\text{ K}$$

The thermal efficiency of the cycle is then

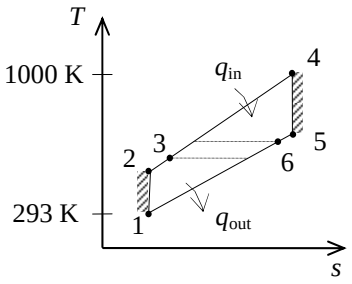
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_4 - T_3} = 1 - \frac{516.9 - 293}{1000 - 567.5} = \mathbf{0.482}$$

For the rearranged version of this cycle,

$$T_3 = T_6 - 6$$

An energy balance on the heat exchanger gives

$$T_3 - T_2 = T_5 - T_6$$



The solution of these two equations is

$$T_3 = 539.2\text{ K}$$
$$T_6 = 545.2\text{ K}$$

The thermal efficiency of the cycle is then

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_4 - T_3} = 1 - \frac{545.2 - 293}{1000 - 539.2} = \mathbf{0.453}$$

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9-112E An ideal Brayton cycle with regeneration has a pressure ratio of 11. The thermal efficiency of the cycle is to be determined with and without regenerator cases.

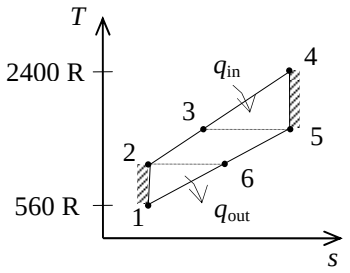
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis According to the isentropic process expressions for an ideal gas,

$$T_2 = T_1 r_p^{(k-1)/k} = (560\ \text{R})(11)^{0.4/1.4} = 1111\ \text{R}$$

$$T_5 = T_4\left(\frac{1}{r_p}\right)^{(k-1)/k} = (2400\ \text{R})\left(\frac{1}{11}\right)^{0.4/1.4} = 1210\ \text{R}$$



The regenerator is ideal (i.e., the effectiveness is 100%) and thus,

$$T_3 = T_5 = 1210\ \text{R}$$
$$T_6 = T_2 = 1111\ \text{R}$$

The thermal efficiency of the cycle is then

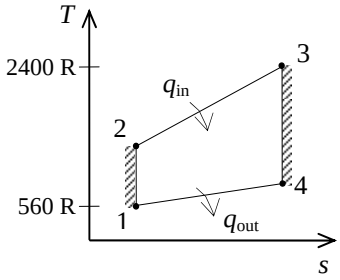
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_4 - T_3} = 1 - \frac{1111 - 560}{2400 - 1210} = 0.537 = \mathbf{53.7\%}$$

The solution without a regenerator is as follows:

$$T_2 = T_1 r_p^{(k-1)/k} = (560\ \text{R})(11)^{0.4/1.4} = 1111\ \text{R}$$

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (2400\ \text{R})\left(\frac{1}{11}\right)^{0.4/1.4} = 1210\ \text{R}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{1210 - 560}{2400 - 1111} = 0.496 = \mathbf{49.6\%}$$



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9-113E A car is powered by a gas turbine with a pressure ratio of 4. The thermal efficiency of the car and the mass flow rate of air for a net power output of 95 hp are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** The ambient air is 540 R and 14.5 psia. **4** The effectiveness of the regenerator is 0.9, and the isentropic efficiencies for both the compressor and the turbine are 80%. **5** The combustion gases can be treated as air.

Properties The properties of air at the compressor and turbine inlet temperatures can be obtained from Table A-17E.

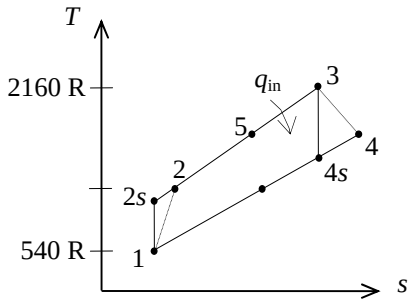
Analysis The gas turbine cycle with regeneration can be analyzed as follows:

$$T_1 = 540\ \text{R} \longrightarrow \begin{matrix} h_1 = 129.06\ \text{Btu/lbm} \\ P_{r_1} = 1.386 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (4)(1.386) = 5.544 \longrightarrow h_{2s} = 192.0\ \text{Btu/lbm}$$

$$T_3 = 2160\ \text{R} \longrightarrow \begin{matrix} h_3 = 549.35\ \text{Btu/lbm} \\ P_{r_3} = 230.12 \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{4}\right)(230.12) = 57.53 \longrightarrow h_{4s} = 372.2\ \text{Btu/lbm}$$



and

$$\eta_{comp} = \frac{h_{2s} - h_1}{h_2 - h_1} \rightarrow 0.80 = \frac{192.0 - 129.06}{h_2 - 129.06} \rightarrow h_2 = 207.74\ \text{Btu/lbm}$$

$$\eta_{turb} = \frac{h_3 - h_4}{h_3 - h_{4s}} \rightarrow 0.80 = \frac{549.35 - h_4}{549.35 - 372.2} \rightarrow h_4 = 407.63\ \text{Btu/lbm}$$

Then the thermal efficiency of the gas turbine cycle becomes

$$q_{regen} = \varepsilon(h_4 - h_2) = 0.9(407.63 - 207.74) = 179.9\ \text{Btu/lbm}$$

$$q_{in} = (h_3 - h_2) - q_{regen} = (549.35 - 207.74) - 179.9 = 161.7\ \text{Btu/lbm}$$

$$w_{net,out} = w_{T,out} - w_{C,in} = (h_3 - h_4) - (h_2 - h_1) = (549.35 - 407.63) - (207.74 - 129.06) = 63.0\ \text{Btu/lbm}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{63.0\ \text{Btu/lbm}}{161.7\ \text{Btu/lbm}} = 0.39 = \mathbf{39\%}$$

Finally, the mass flow rate of air through the turbine becomes

$$\dot{m}_{air} = \frac{\dot{W}_{net}}{w_{net}} = \frac{95\ \text{hp}}{63.0\ \text{Btu/lbm}} \left(\frac{0.7068\ \text{Btu/s}}{1\ \text{hp}}\right) = \mathbf{1.07\ lbm/s}$$

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9-114 The thermal efficiency and power output of an actual gas turbine are given. The isentropic efficiency of the turbine and of the compressor, and the thermal efficiency of the gas turbine modified with a regenerator are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible. **3** The mass flow rates of air and of the combustion gases are the same, and the properties of combustion gases are the same as those of air.

Properties The properties of air are given in Table A-17.

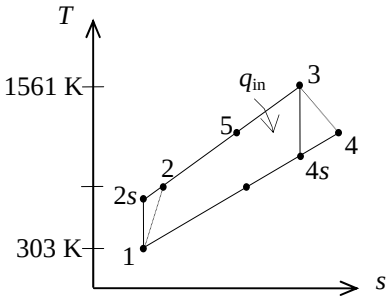
Analysis The properties at various states are

$$T_1 = 30°\text{C} = 303\ \text{K} \longrightarrow \begin{matrix} h_1 = 303.21\ \text{kJ/kg} \\ P_{r_1} = 1.4356 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (14.7)(1.4356) = 21.10 \longrightarrow h_{2s} = 653.25\ \text{kJ/kg}$$

$$T_3 = 1288°\text{C} = 1561\ \text{K} \longrightarrow \begin{matrix} h_3 = 1710.0\ \text{kJ/kg} \\ P_{r_3} = 712.5 \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{14.7}\right)(712.5) = 48.47 \longrightarrow h_{4s} = 825.23\ \text{kJ/kg}$$



The net work output and the heat input per unit mass are

$$w_{net} = \frac{\dot{W}_{net}}{\dot{m}} = \frac{159{,}000\ \text{kW}}{1{,}536{,}000\ \text{kg/h}}\left(\frac{3600\ \text{s}}{1\ \text{h}}\right) = 372.66\ \text{kJ/kg}$$

$$q_{in} = \frac{w_{net}}{\eta_{th}} = \frac{372.66\ \text{kJ/kg}}{0.359} = 1038.0\ \text{kJ/kg}$$

$$q_{in} = h_3 - h_2 \ \rightarrow \ h_2 = h_3 - q_{in} = 1710 - 1038 = 672.0\ \text{kJ/kg}$$

$$q_{out} = q_{in} - w_{net} = 1038.0 - 372.66 = 665.34\ \text{kJ/kg}$$

$$q_{out} = h_4 - h_1 \ \rightarrow \ h_4 = q_{out} + h_1 = 665.34 + 303.21 = 968.55\ \text{kJ/kg} \ \rightarrow \ T_4 = 931.7\ \text{K} = 658.7°\text{C}$$

Then the compressor and turbine efficiencies become

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1710 - 968.55}{1710 - 825.23} = 0.838 = \mathbf{83.8\%}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{653.25 - 303.21}{672 - 303.21} = 0.949 = \mathbf{94.9\%}$$

When a regenerator is added, the new heat input and the thermal efficiency become

$$q_{regen} = \varepsilon(h_4 - h_2) = (0.65)(968.55 - 672.0) = 192.8\ \text{kJ/kg}$$

$$q_{in,new} = q_{in} - q_{regen} = 1038 - 192.8 = 845.2\ \text{kJ/kg}$$

$$\eta_{th,new} = \frac{w_{net}}{q_{in,new}} = \frac{372.66\ \text{kJ/kg}}{845.2\ \text{kJ/kg}} = 0.441 = \mathbf{44.1\%}$$

Discussion Note a 65% efficient regenerator would increase the thermal efficiency of this gas turbine from 35.9% to 44.1%.

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9-115 Problem 9-114 is reconsidered. A solution that allows different isentropic efficiencies for the compressor and turbine is to be developed and the effect of the isentropic efficiencies on net work done and the heat supplied to the cycle is to be studied. Also, the *T-s* diagram for the cycle is to be plotted.

Analysis Using EES, the problem is solved as follows:

```
"Input data"
T[3] = 1288 [C]
Pratio = 14.7
T[1] = 30 [C]
P[1]= 100 [kPa]
{T[4]=659 [C]}
{W_dot_net=159 [MW] }"We omit the information about the cycle net work"
m_dot = 1536000 [kg/h]*Convert(kg/h,kg/s)
{Eta_th_noreg=0.359} "We omit the information about the cycle efficiency."
Eta_reg = 0.65
Eta_c = 0.84 "Compressor isentropic efficiency"
Eta_t = 0.95 "Turbien isentropic efficiency"

"Isentropic Compressor anaysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = W_dot_compisen/W_dot_comp
"compressor adiabatic efficiency, W_dot_comp > W_dot_compisen"

"Conservation of energy for the compressor for the isentropic case:
E_dot_in - E_dot_out = DELTAE_dot=0 for steady-flow"
m_dot*h[1] + W_dot_compisen = m_dot*h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])

"Actual compressor analysis:"
m_dot*h[1] + W_dot_comp = m_dot*h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
E_dot_in - E_dot_out =DELTAE_dot_cv =0 for steady flow"
m_dot*h[2] + Q_dot_in_noreg = m_dot*h[3]
q_in_noreg=Q_dot_in_noreg/m_dot
h[3]=ENTHALPY(Air,T=T[3])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = W_dot_turb /W_dot_turbisen "turbine adiabatic efficiency, W_dot_turbisen > W_dot_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
E_dot_in -E_dot_out = DELTAE_dot_cv = 0 for steady-flow"
m_dot*h[3] = W_dot_turbisen + m_dot*h_s[4]
h_s[4]=ENTHALPY(Air,T=T_s[4])
```

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"Actual Turbine analysis:"
m_dot*h[3] = W_dot_turb + m_dot*h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])

"Cycle analysis"
"Using the definition of the net cycle work and 1 MW = 1000 kW:"
W_dot_net*1000=W_dot_turb-W_dot_comp "kJ/s"
Eta_th_noreg=W_dot_net*1000/Q_dot_in_noreg"Cycle thermal efficiency"
Bwr=W_dot_comp/W_dot_turb"Back work ratio"

"With the regenerator the heat added in the external heat exchanger is"
m_dot*h[5] + Q_dot_in_withreg = m_dot*h[3]
q_in_withreg=Q_dot_in_withreg/m_dot

h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[2]

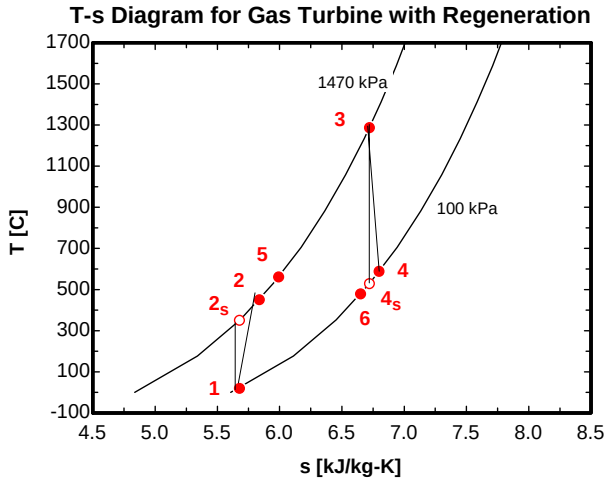
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[2])/(h[4]-h[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
m_dot*h[2] + m_dot*h[4]=m_dot*h[5] + m_dot*h[6]
h[6]=ENTHALPY(Air, T=T[6])
s[6]=ENTROPY(Air,T=T[6], P=P[6])
P[6]=P[4]

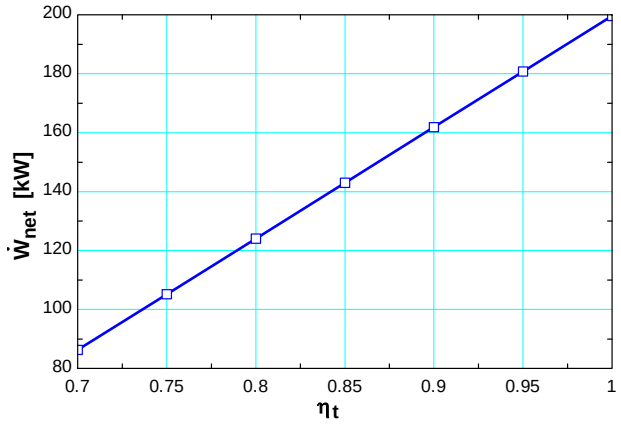
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=W_dot_net*1000/Q_dot_in_withreg

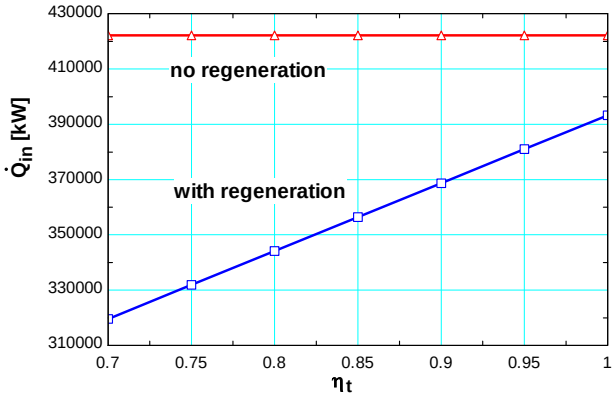
"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]

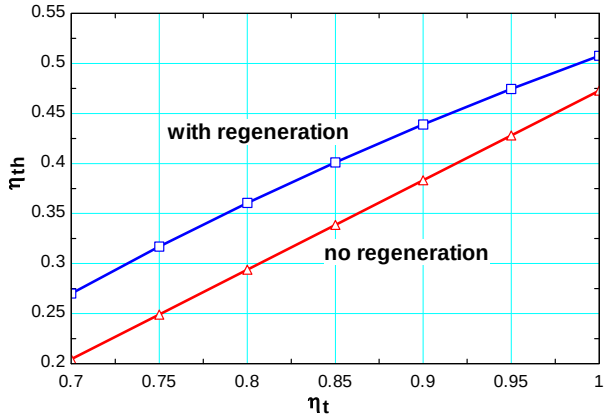
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η_t	η_c	$\eta_{th,noreg}$	$\eta_{th,withreg}$	$Q_{innoreg}$ [kW]	$Q_{inwithreg}$ [kW]	W_{net} [kW]
0.7	0.84	0.2044	0.27	422152	319582	86.3
0.75	0.84	0.2491	0.3169	422152	331856	105.2
0.8	0.84	0.2939	0.3605	422152	344129	124.1
0.85	0.84	0.3386	0.4011	422152	356403	142.9
0.9	0.84	0.3833	0.4389	422152	368676	161.8
0.95	0.84	0.4281	0.4744	422152	380950	180.7
1	0.84	0.4728	0.5076	422152	393223	199.6





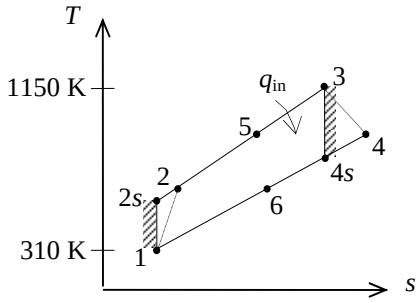




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9-116 A Brayton cycle with regeneration using air as the working fluid is considered. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.



Properties The properties of air are given in Table A-17.

Analysis (a) The properties of air at various states are

$$T_1 = 310\text{ K} \longrightarrow \begin{matrix} h_1 = 310.24\text{ kJ/kg} \\ P_{r_1} = 1.5546 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (7)(1.5546) = 10.88 \longrightarrow h_{2s} = 541.26\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + (h_{2s} - h_1)/\eta_C = 310.24 + (541.26 - 310.24)/(0.75) = 618.26\text{ kJ/kg}$$

$$T_3 = 1150\text{ K} \longrightarrow \begin{matrix} h_3 = 1219.25\text{ kJ/kg} \\ P_{r_3} = 200.15 \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{7}\right)(200.15) = 28.59 \longrightarrow h_{4s} = 711.80\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s}) = 1219.25 - (0.82)(1219.25 - 711.80) = 803.14\text{ kJ/kg}$$

Thus,

$$T_4 = \mathbf{782.8\ K}$$

(*b*)
$$\begin{aligned} w_{\text{net}} &= w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1) \\ &= (1219.25 - 803.14) - (618.26 - 310.24) \\ &= \mathbf{108.09\ kJ/kg} \end{aligned}$$

(*c*)
$$\begin{aligned} \varepsilon = \frac{h_5 - h_2}{h_4 - h_2} \longrightarrow h_5 &= h_2 + \varepsilon(h_4 - h_2) \\ &= 618.26 + (0.65)(803.14 - 618.26) \\ &= 738.43\text{ kJ/kg} \end{aligned}$$

Then,

$$q_{\text{in}} = h_3 - h_5 = 1219.25 - 738.43 = 480.82\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{108.09\text{ kJ/kg}}{480.82\text{ kJ/kg}} = \mathbf{22.5\%}$$

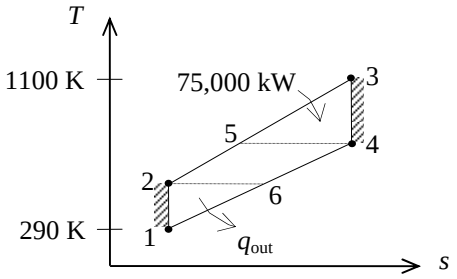
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9-117 A stationary gas-turbine power plant operating on an ideal regenerative Brayton cycle with air as the working fluid is considered. The power delivered by this plant is to be determined for two cases.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas. **3** Kinetic and potential energy changes are negligible.

Properties When assuming constant specific heats, the properties of air at room temperature are c_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a). When assuming variable specific heats, the properties of air are obtained from Table A-17.

Analysis (*a*) Assuming constant specific heats,



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (290\text{ K})(8)^{0.4/1.4} = 525.3\text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1100\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 607.2\text{ K}$$

$$\varepsilon = 100\% \longrightarrow T_5 = T_4 = 607.2\text{ K and } T_6 = T_2 = 525.3\text{ K}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_6 - T_1)}{c_p(T_3 - T_5)} = 1 - \frac{T_6 - T_1}{T_3 - T_5} = 1 - \frac{525.3 - 290}{1100 - 607.2} = 0.5225$$

$$\dot{W}_{net} = \eta_T \dot{Q}_{in} = (0.5225)(75{,}000\text{ kW}) = \mathbf{39{,}188\ kW}$$

(*b*) Assuming variable specific heats,

$$T_1 = 290\text{K} \longrightarrow \begin{matrix} h_1 = 290.16\text{ kJ/kg} \\ P_{r_1} = 1.2311 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.2311) = 9.8488 \longrightarrow h_2 = 526.12\text{ kJ/kg}$$

$$T_3 = 1100\text{K} \longrightarrow \begin{matrix} h_3 = 1161.07\text{ kJ/kg} \\ P_{r_3} = 167.1 \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(167.1) = 20.89 \longrightarrow h_4 = 651.37\text{ kJ/kg}$$

$$\varepsilon = 100\% \longrightarrow h_5 = h_4 = 651.37\text{ kJ/kg and } h_6 = h_2 = 526.12\text{ kJ/kg}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{h_6 - h_1}{h_3 - h_5} = 1 - \frac{526.12 - 290.16}{1161.07 - 651.37} = 0.5371$$

$$\dot{W}_{net} = \eta_T \dot{Q}_{in} = (0.5371)(75{,}000\text{ kW}) = \mathbf{40{,}283\ kW}$$

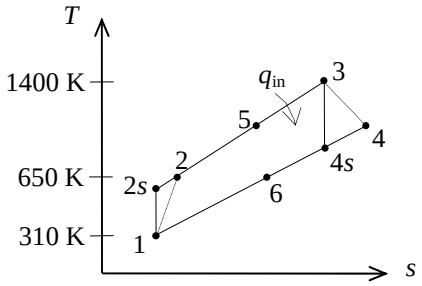
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9-118 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air are given in Table A-17.

Analysis (*a*) The properties at various states are



$$r_p = P_2 / P_1 = 900/100 = 9$$

$$T_1 = 310\text{ K} \longrightarrow h_1 = 310.24\text{ kJ/kg}$$
$$T_2 = 650\text{ K} \longrightarrow h_2 = 659.84\text{ kJ/kg}$$
$$T_3 = 1400\text{ K} \longrightarrow h_3 = 1515.42\text{ kJ/kg}$$
$$P_{r_3} = 450.5$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{9}\right)(450.5) = 50.06 \longrightarrow h_{4s} = 832.44\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 1515.42 - (0.90)(1515.42 - 832.44)$$
$$= 900.74\text{ kJ/kg}$$

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = (0.80)(900.74 - 659.84) = \mathbf{192.7\ kJ/kg}$$

(*b*)
$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1)$$
$$= (1515.42 - 900.74) - (659.84 - 310.24) = 265.08\text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = (1515.42 - 659.84) - 192.7 = 662.88\text{ kJ/kg}$$

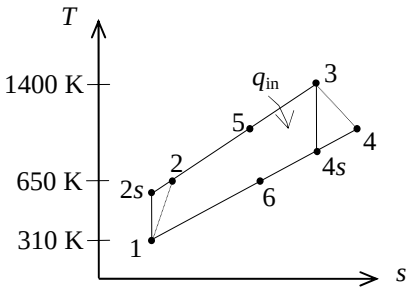
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{265.08\text{ kJ/kg}}{662.88\text{ kJ/kg}} = 0.400 = \mathbf{40.0\%}$$

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9-119 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a).



Analysis (a) Using the isentropic relations and turbine efficiency,

$$r_p = P_2 / P_1 = 900/100 = 9$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{1}{9}\right)^{0.4/1.4} = 747.3\text{ K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{c_p(T_3 - T_4)}{c_p(T_3 - T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3 - T_{4s})$$
$$= 1400 - (0.90)(1400 - 747.3)$$
$$= 812.6\text{ K}$$

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = \varepsilon c_p(T_4 - T_2) = (0.80)(1.005\text{ kJ/kg}\cdot\text{K})(812.6 - 650)\text{K} = \mathbf{130.7\ kJ/kg}$$

(b) $$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = c_p(T_3 - T_4) - c_p(T_2 - T_1)$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})[(1400 - 812.6) - (650 - 310)]\text{K} = 248.7\text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = c_p(T_3 - T_2) - q_{\text{regen}}$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})(1400 - 650)\text{K} - 130.7 = 623.1\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{248.7\text{ kJ/kg}}{623.1\text{ kJ/kg}} = 0.399 = \mathbf{39.9\%}$$

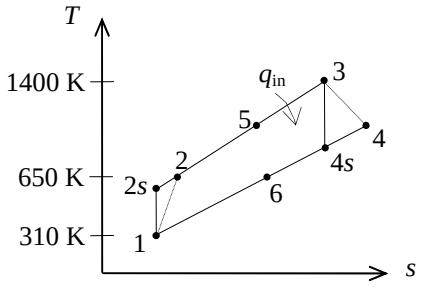
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9-120 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air are given in Table A-17.

Analysis (*a*) The properties at various states are



$$r_p = P_2 / P_1 = 900/100 = 9$$

$$T_1 = 310\text{ K} \longrightarrow h_1 = 310.24\text{ kJ/kg}$$
$$T_2 = 650\text{ K} \longrightarrow h_2 = 659.84\text{ kJ/kg}$$
$$T_3 = 1400\text{ K} \longrightarrow h_3 = 1515.42\text{ kJ/kg}$$
$$P_{r_3} = 450.5$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{9}\right)(450.5) = 50.06 \longrightarrow h_{4s} = 832.44\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 1515.42 - (0.90)(1515.42 - 832.44)$$
$$= 900.74\text{ kJ/kg}$$

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = (0.70)(900.74 - 659.84) = \mathbf{168.6\ kJ/kg}$$

(*b*)
$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1)$$
$$= (1515.42 - 900.74) - (659.84 - 310.24) = 265.08\text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = (1515.42 - 659.84) - 168.6 = 687.18\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{265.08\text{ kJ/kg}}{687.18\text{ kJ/kg}} = 0.386 = \mathbf{38.6\%}$$

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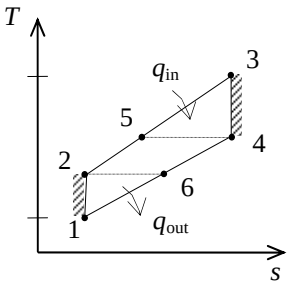
9-121 An expression for the thermal efficiency of an ideal Brayton cycle with an ideal regenerator is to be developed.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Analysis The expressions for the isentropic compression and expansion processes are

$$T_2 = T_1 r_p^{(k-1)/k}$$

$$T_4 = T_3 \left(\frac{1}{r_p} \right)^{(k-1)/k}$$



For an ideal regenerator,

$$T_5 = T_4$$
$$T_6 = T_2$$

The thermal efficiency of the cycle is

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_3 - T_5} = 1 - \frac{T_1}{T_3} \frac{(T_6/T_1) - 1}{1 - (T_5/T_3)}$$
$$= 1 - \frac{T_1}{T_3} \frac{(T_2/T_1) - 1}{1 - (T_4/T_3)}$$
$$= 1 - \frac{T_1}{T_3} \frac{r_p^{(k-1)/k} - 1}{1 - r_p^{-(k-1)/k}}$$
$$= 1 - \frac{T_1}{T_3} r_p^{(k-1)/k}$$

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Brayton Cycle with Intercooling, Reheating, and Regeneration

9-122C As the number of compression and expansion stages are increased and regeneration is employed, the ideal Brayton cycle will approach the Ericsson cycle.

9-123C Because the steady-flow work is proportional to the specific volume of the gas. Intercooling decreases the average specific volume of the gas during compression, and thus the compressor work. Reheating increases the average specific volume of the gas, and thus the turbine work output.

9-124C (a) decrease, (b) decrease, and (c) decrease.

9-125C (a) increase, (b) decrease, and (c) decrease.

9-126C (a) increase, (b) decrease, (c) decrease, and (d) increase.

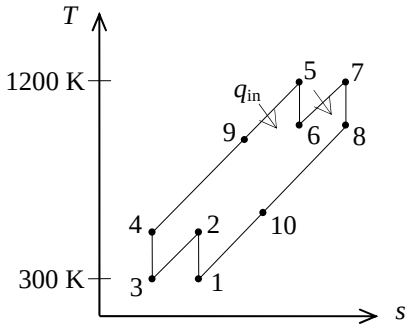
9-127C (a) increase, (b) decrease, (c) increase, and (d) decrease.

9-128C (c) The Carnot (or Ericsson) cycle efficiency.

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9-129 An ideal gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of with and without a regenerator.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.



Properties The properties of air are given in Table A-17.

Analysis (*a*) The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine since this is an ideal cycle. Then,

$$T_1 = 300\text{ K} \longrightarrow \begin{matrix} h_1 = 300.19\text{ kJ/kg} \\ P_{r_1} = 1.386 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_2 = h_4 = 411.26\text{ kJ/kg}$$

$$T_5 = 1200\text{ K} \longrightarrow \begin{matrix} h_5 = h_7 = 1277.79\text{ kJ/kg} \\ P_{r_5} = 238 \end{matrix}$$

$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_6 = h_8 = 946.36\text{ kJ/kg}$$

$$w_{C,in} = 2(h_2 - h_1) = 2(411.26 - 300.19) = 222.14\text{ kJ/kg}$$

$$w_{T,out} = 2(h_5 - h_6) = 2(1277.79 - 946.36) = 662.86\text{ kJ/kg}$$

Thus,

$$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{222.14\text{ kJ/kg}}{662.86\text{ kJ/kg}} = \mathbf{33.5\%}$$

$$q_{in} = (h_5 - h_4) + (h_7 - h_6) = (1277.79 - 411.26) + (1277.79 - 946.36) = 1197.96\text{ kJ/kg}$$

$$w_{net} = w_{T,out} - w_{C,in} = 662.86 - 222.14 = 440.72\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{440.72\text{ kJ/kg}}{1197.96\text{ kJ/kg}} = \mathbf{36.8\%}$$

(*b*) When a regenerator is used, r_{bw} remains the same. The thermal efficiency in this case becomes

$$q_{regen} = \varepsilon(h_8 - h_4) = (0.75)(946.36 - 411.26) = 401.33\text{ kJ/kg}$$

$$q_{in} = q_{in,old} - q_{regen} = 1197.96 - 401.33 = 796.63\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{440.72\text{ kJ/kg}}{796.63\text{ kJ/kg}} = \mathbf{55.3\%}$$

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9-130 A gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of with and without a regenerator.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air are given in Table A-17.

Analysis (*a*) The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine. Then,

$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.19\text{ kJ/kg}$$
$$P_{r_1} = 1.386$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_{2s} = h_{4s} = 411.26\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_4 = h_1 + (h_{2s} - h_1)/\eta_C$$
$$= 300.19 + (411.26 - 300.19)/(0.84)$$
$$= 432.42\text{ kJ/kg}$$

$$T_5 = 1200\text{ K} \longrightarrow h_5 = h_7 = 1277.79\text{ kJ/kg}$$
$$P_{r_5} = 238$$

$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_{6s} = h_{8s} = 946.36\text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_8 = h_5 - \eta_T (h_5 - h_{6s})$$
$$= 1277.79 - (0.88)(1277.79 - 946.36)$$
$$= 986.13\text{ kJ/kg}$$

$$w_{\text{C,in}} = 2(h_2 - h_1) = 2(432.42 - 300.19) = 264.46\text{ kJ/kg}$$
$$w_{\text{T,out}} = 2(h_5 - h_6) = 2(1277.79 - 986.13) = 583.32\text{ kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{264.46\text{ kJ/kg}}{583.32\text{ kJ/kg}} = 0.453 = \mathbf{45.3\%}$$

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (1277.79 - 432.42) + (1277.79 - 986.13) = 1137.03\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 583.32 - 264.46 = 318.86\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{318.86\text{ kJ/kg}}{1137.03\text{ kJ/kg}} = 0.280 = \mathbf{28.0\%}$$

(*b*) When a regenerator is used, r_{bw} remains the same. The thermal efficiency in this case becomes

$$q_{\text{regen}} = \varepsilon(h_8 - h_4) = (0.75)(986.13 - 432.42) = 415.28\text{ kJ/kg}$$
$$q_{\text{in}} = q_{\text{in,old}} - q_{\text{regen}} = 1137.03 - 415.28 = 721.75\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{318.86\text{ kJ/kg}}{721.75\text{ kJ/kg}} = 0.442 = \mathbf{44.2\%}$$

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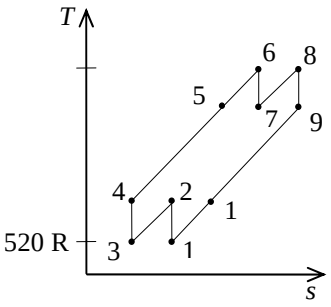
9-131E An ideal regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The power produced and consumed by each compression and expansion stage, and the rate of heat rejected are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis The pressure ratio for each stage is

$$r_p = \sqrt{12} = 3.464$$



According to the isentropic process expressions for an ideal gas,

$$T_2 = T_4 = T_1 r_p^{(k-1)/k} = (520\text{ R})(3.464)^{0.4/1.4} = 741.6\text{ R}$$

Since this is an ideal cycle,

$$T_5 = T_7 = T_9 = T_4 + 50 = 741.6 + 50 = 791.6\text{ R}$$

For the isentropic expansion processes,

$$T_6 = T_8 = T_7 r_p^{(k-1)/k} = (791.6\text{ R})(3.464)^{0.4/1.4} = 1129\text{ R}$$

The heat input is

$$q_{\text{in}} = 2c_p(T_6 - T_5) = 2(0.24\text{ Btu/lbm}\cdot\text{R})(1129 - 791.6)\text{ R} = 162.0\text{ Btu/lbm}$$

The mass flow rate is then

$$\dot{m} = \frac{\dot{Q}_{\text{in}}}{q_{\text{in}}} = \frac{500\text{ Btu/s}}{162.0\text{ Btu/lbm}} = 3.086\text{ lbm/s}$$

Application of the first law to the expansion process 6-7 gives

$$\begin{aligned}\dot{W}_{\text{6-7,out}} &= \dot{m}c_p(T_6 - T_7)\\ &= (3.086\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(1129 - 791.6)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right)\\ &= \mathbf{263.6\text{ kW}}\end{aligned}$$

The same amount of power is produced in process 8-9. When the first law is adapted to the compression process 1-2 it becomes

$$\begin{aligned}\dot{W}_{\text{12,in}} &= \dot{m}c_p(T_2 - T_1)\\ &= (3.086\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(741.6 - 520)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right)\\ &= \mathbf{173.2\text{ kW}}\end{aligned}$$

Compression process 3-4 uses the same amount of power. The rate of heat rejection from the cycle is

$$\begin{aligned}\dot{Q}_{\text{out}} &= 2\dot{m}c_p(T_2 - T_3)\\ &= 2(3.086\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(741.6 - 520)\text{ R}\\ &= \mathbf{328.3\text{ Btu/s}}\end{aligned}$$

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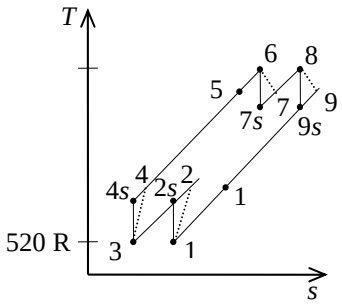
9-132E An ideal regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The power produced and consumed by each compression and expansion stage, and the rate of heat rejected are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis The pressure ratio for each stage is

$$r_p = \sqrt{12} = 3.464$$



For the compression processes,

$$T_{2s} = T_{4s} = T_1 r_p^{(k-1)/k} = (520\text{ R})(3.464)^{0.4/1.4} = 741.6\text{ R}$$

$$\eta_C = \frac{c_p(T_{2s} - T_1)}{c_p(T_2 - T_1)} \longrightarrow T_2 = T_4 = T_1 + \frac{T_{2s} - T_1}{\eta_C}$$

$$= 520 + \frac{741.6 - 520}{0.85} = 780.7\text{ R}$$

Since the regenerator is ideal,

$$T_5 = T_7 = T_9 = T_4 + 50 = 780.7 + 50 = 830.7\text{ R}$$

For the expansion processes,

$$T_{6s} = T_{8s} = T_7 r_p^{(k-1)/k} = (830.7\text{ R})(3.464)^{0.4/1.4} = 1185\text{ R}$$

$$\eta_T = \frac{c_p(T_6 - T_7)}{c_p(T_{6s} - T_7)} \longrightarrow T_6 = T_8 = T_7 + \eta_T(T_{6s} - T_7) = 830.7 + (0.90)(1185 - 830.7) = 1150\text{ R}$$

The heat input is

$$q_{\text{in}} = 2c_p(T_6 - T_5) = 2(0.24\text{ Btu/lbm}\cdot\text{R})(1150 - 830.7)\text{ R} = 153.3\text{ Btu/lbm}$$

The mass flow rate is then

$$\dot{m} = \frac{\dot{Q}_{\text{in}}}{q_{\text{in}}} = \frac{500\text{ Btu/s}}{153.3\text{ Btu/lbm}} = 3.262\text{ lbm/s}$$

Application of the first law to the expansion process 6-7 gives

$$\dot{W}_{6\text{-}7,\text{out}} = \dot{m}c_p(T_6 - T_7)$$

$$= (3.262\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(1150 - 830.7)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right) = \mathbf{263.7\text{ kW}}$$

The same amount of power is produced in process 8-9. When the first law is adapted to the compression process 1-2 it becomes

$$\dot{W}_{12,\text{in}} = \dot{m}c_p(T_2 - T_1)$$

$$= (3.262\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(780.7 - 520)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right) = \mathbf{215.3\text{ kW}}$$

Compression process 3-4 uses the same amount of power. The rate of heat rejection from the cycle is

$$\dot{Q}_{\text{out}} = 2\dot{m}c_p(T_2 - T_3)$$

$$= 2(3.262\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(780.7 - 520)\text{ R} = \mathbf{408.2\text{ Btu/s}}$$

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9-133 A regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The thermal efficiency of the cycle is to be determined.

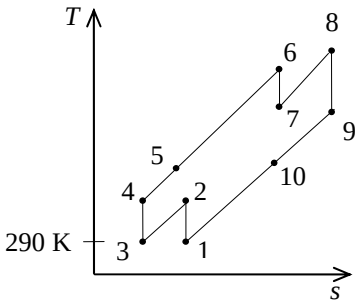
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis The temperatures at various states are obtained as follows

$$T_2 = T_4 = T_1 r_p^{(k-1)/k} = (290\text{ K})(4)^{0.4/1.4} = 430.9\text{ K}$$

$$T_5 = T_4 + 20 = 430.9 + 20 = 450.9\text{ K}$$



$$q_{\text{in}} = c_p(T_6 - T_5)$$

$$T_6 = T_5 + \frac{q_{\text{in}}}{c_p} = 450.9\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 749.4\text{ K}$$

$$T_7 = T_6\left(\frac{1}{r_p}\right)^{(k-1)/k} = (749.4\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 504.3\text{ K}$$

$$T_8 = T_7 + \frac{q_{\text{in}}}{c_p} = 504.3\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 802.8\text{ K}$$

$$T_9 = T_8\left(\frac{1}{r_p}\right)^{(k-1)/k} = (802.8\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 540.2\text{ K}$$

$$T_{10} = T_9 - 20 = 540.2 - 20 = 520.2\text{ K}$$

The heat input is

$$q_{\text{in}} = 300 + 300 = 600\text{ kJ/kg}$$

The heat rejected is

$$\begin{aligned} q_{\text{out}} &= c_p(T_{10} - T_1) + c_p(T_2 - T_3) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(520.2 - 290 + 430.9 - 290)\text{ R} \\ &= 373.0\text{ kJ/kg} \end{aligned}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{373.0}{600} = \mathbf{0.378}$$

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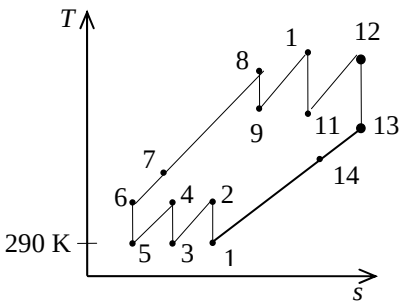
9-134 A regenerative gas-turbine cycle with three stages of compression and three stages of expansion is considered. The thermal efficiency of the cycle is to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis The temperatures at various states are obtained as follows

$$T_2 = T_4 = T_6 = T_1 r_p^{(k-1)/k} = (290\text{ K})(4)^{0.4/1.4} = 430.9\text{ K}$$



$$T_7 = T_6 + 20 = 430.9 + 20 = 450.9\text{ K}$$

$$q_{\text{in}} = c_p(T_8 - T_7)$$

$$T_8 = T_7 + \frac{q_{\text{in}}}{c_p} = 450.9\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 749.4\text{ K}$$

$$T_9 = T_8\left(\frac{1}{r_p}\right)^{(k-1)/k} = (749.4\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 504.3\text{ K}$$

$$T_{10} = T_9 + \frac{q_{\text{in}}}{c_p} = 504.3\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 802.8\text{ K}$$

$$T_{11} = T_{10}\left(\frac{1}{r_p}\right)^{(k-1)/k} = (802.8\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 540.2\text{ K}$$

$$T_{12} = T_{11} + \frac{q_{\text{in}}}{c_p} = 540.2\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 838.7\text{ K}$$

$$T_{13} = T_{12}\left(\frac{1}{r_p}\right)^{(k-1)/k} = (838.7\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 564.4\text{ K}$$

$$T_{14} = T_{13} - 20 = 564.4 - 20 = 544.4\text{ K}$$

The heat input is

$$q_{\text{in}} = 300 + 300 + 300 = 900\text{ kJ/kg}$$

The heat rejected is

$$\begin{aligned} q_{\text{out}} &= c_p(T_{14} - T_1) + c_p(T_2 - T_3) + c_p(T_4 - T_5) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(544.4 - 290 + 430.9 - 290 + 430.9 - 290)\text{ R} \\ &= 538.9\text{ kJ/kg} \end{aligned}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{538.9}{900} = 0.401 = \mathbf{40.1\%}$$

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9-135 A regenerative gas-turbine cycle with three stages of compression and three stages of expansion is considered. The thermal efficiency of the cycle is to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

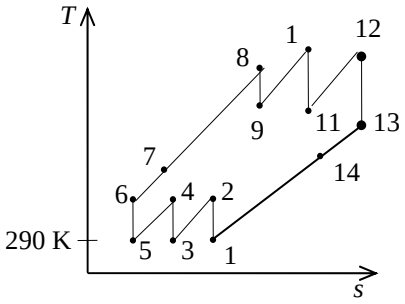
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis Since all compressors share the same compression ratio and begin at the same temperature,

$$T_2 = T_4 = T_6 = T_1 r_p^{(k-1)/k} = (290\text{ K})(4)^{0.4/1.4} = 430.9\text{ K}$$

From the problem statement,

$$T_7 = T_{13} - 65$$



The relations for heat input and expansion processes are

$$q_{\text{in}} = c_p(T_8 - T_7) \longrightarrow T_8 = T_7 + \frac{q_{\text{in}}}{c_p}$$

$$T_9 = T_8\left(\frac{1}{r_p}\right)^{(k-1)/k}$$

$$T_{10} = T_9 + \frac{q_{\text{in}}}{c_p}, \quad T_{11} = T_{10}\left(\frac{1}{r_p}\right)^{(k-1)/k}$$

$$T_{12} = T_{11} + \frac{q_{\text{in}}}{c_p}, \quad T_{13} = T_{12}\left(\frac{1}{r_p}\right)^{(k-1)/k}$$

The simultaneous solution of above equations using EES software gives the following results

$$T_7 = 520.7\text{ K}, \quad T_8 = 819.2\text{ K}, \quad T_9 = 551.3\text{ K}$$
$$T_{10} = 849.8\text{ K}, \quad T_{11} = 571.9\text{ K}, \quad T_{12} = 870.4\text{ K}, \quad T_{13} = 585.7\text{ K}$$

From an energy balance on the regenerator,

$$T_7 - T_6 = T_{13} - T_{14}$$
$$(T_{13} - 65) - T_6 = T_{13} - T_{14} \longrightarrow T_{14} = T_6 + 65 = 430.9 + 65 = 495.9\text{ K}$$

The heat input is

$$q_{\text{in}} = 300 + 300 + 300 = 900\text{ kJ/kg}$$

The heat rejected is

$$\begin{aligned} q_{\text{out}} &= c_p(T_{14} - T_1) + c_p(T_2 - T_3) + c_p(T_4 - T_5) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(495.9 - 290 + 430.9 - 290 + 430.9 - 290)\text{ R} \\ &= 490.1\text{ kJ/kg} \end{aligned}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{490.1}{900} = 0.455 = \mathbf{45.5\%}$$

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Jet-Propulsion Cycles

9-136C The power developed from the thrust of the engine is called the propulsive power. It is equal to thrust times the aircraft velocity.

9-137C The ratio of the propulsive power developed and the rate of heat input is called the propulsive efficiency. It is determined by calculating these two quantities separately, and taking their ratio.

9-138C It reduces the exit velocity, and thus the thrust.

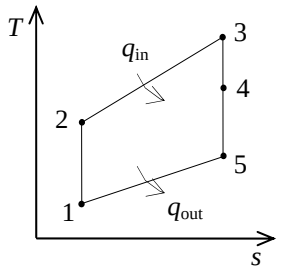
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9-139E A turboprop engine operating on an ideal cycle is considered. The thrust force generated is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm·R (Table A-1E), c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis Working across the two isentropic processes of the cycle yields



$$T_2 = T_1 r_p^{(k-1)/k} = (450\text{ R})(10)^{0.4/1.4} = 868.8\text{ R}$$

$$T_5 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1400\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 725.1\text{ R}$$

Since the work produced by expansion 3-4 equals that used by compression 1-2, an energy balance gives

$$T_4 = T_3 - (T_2 - T_1) = 1400 - (868.8 - 450) = 981.2\text{ R}$$

The excess enthalpy generated by expansion 4-5 is used to increase the kinetic energy of the flow through the propeller,

$$\dot{m}_e c_p (T_4 - T_5) = \dot{m}_p \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2}$$

which when solved for the velocity at which the air leaves the propeller gives

$$V_{\text{exit}} = \left[2\frac{\dot{m}_e}{\dot{m}_p}c_p(T_4 - T_5) + V_{\text{inlet}}^2\right]^{1/2}$$

$$= \left[2\frac{1}{20}(0.24\text{ Btu/lbm}\cdot\text{R})(981.2 - 725.1)\text{R}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right) + (600\text{ ft/s})^2\right]^{1/2}$$

$$= 716.9\text{ ft/s}$$

The mass flow rate through the propeller is

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3)(450\text{ R})}{8\text{ psia}} = 20.84\text{ ft}^3/\text{lbm}$$

$$\dot{m}_p = \frac{AV_1}{\nu_1} = \frac{\pi D^2}{4}\frac{V_1}{\nu_1} = \frac{\pi(10\text{ ft})^2}{4}\frac{600\text{ ft/s}}{20.84\text{ ft}^3/\text{lbm}} = 2261\text{ lbm/s}$$

The thrust force generated by this propeller is then

$$F = \dot{m}_p(V_{\text{exit}} - V_{\text{inlet}}) = (2261\text{ lbm/s})(716.9 - 600)\text{ft/s}\left(\frac{1\text{ lbf}}{32.174\text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{8215\text{ lbf}}$$

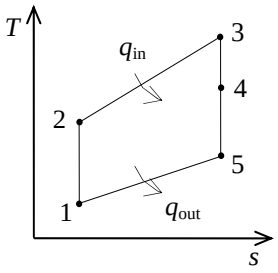
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9-140E A turboprop engine operating on an ideal cycle is considered. The thrust force generated is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm·R (Table A-1E), c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis Working across the two isentropic processes of the cycle yields



$$T_2 = T_1 r_p^{(k-1)/k} = (450\text{ R})(10)^{0.4/1.4} = 868.8\text{ R}$$

$$T_5 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1400\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 725.1\text{ R}$$

Since the work produced by expansion 3-4 equals that used by compression 1-2, an energy balance gives

$$T_4 = T_3 - (T_2 - T_1) = 1400 - (868.8 - 450) = 981.2\text{ R}$$

The mass flow rate through the propeller is

$$\nu_1 = \frac{RT}{P} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3)(450\text{ R})}{8\text{ psia}} = 20.84\text{ ft}^3/\text{lbm}$$

$$\dot{m}_p = \frac{AV_1}{\nu_1} = \frac{\pi D^2}{4}\frac{V_1}{\nu_1} = \frac{\pi(8\text{ ft})^2}{4}\frac{600\text{ ft/s}}{20.84\text{ ft}^3/\text{lbm}} = 1447\text{ lbm/s}$$

According to the previous problem,

$$\dot{m}_e = \frac{\dot{m}_p}{20} = \frac{2261\text{ lbm/s}}{20} = 113.1\text{ lbm/s}$$

The excess enthalpy generated by expansion 4-5 is used to increase the kinetic energy of the flow through the propeller,

$$\dot{m}_e c_p (T_4 - T_5) = \dot{m}_p \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2}$$

which when solved for the velocity at which the air leaves the propeller gives

$$V_{\text{exit}} = \left[2\frac{\dot{m}_e}{\dot{m}_p}c_p(T_4 - T_5) + V_{\text{inlet}}^2\right]^{1/2}$$

$$= \left[2\frac{113.1\text{ lbm/s}}{1447\text{ lbm/s}}(0.24\text{ Btu/lbm}\cdot\text{R})(981.2 - 725.1)\text{R}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right) + (600\text{ ft/s})^2\right]^{1/2}$$

$$= 775.0\text{ ft/s}$$

The thrust force generated by this propeller is then

$$F = \dot{m}_p(V_{\text{exit}} - V_{\text{inlet}}) = (1447\text{ lbm/s})(775 - 600)\text{ft/s}\left(\frac{1\text{ lbf}}{32.174\text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{7870\text{ lbf}}$$

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9-141 A turbofan engine operating on an ideal cycle produces 50,000 N of thrust. The air temperature at the fan outlet needed to produce this thrust is to be determined.

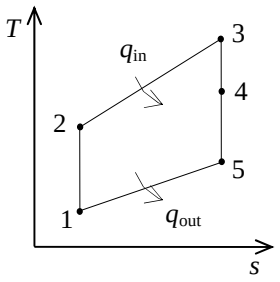
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis The total mass flow rate is

$$\nu_1 = \frac{RT}{P} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3)(253\ \text{K})}{50\ \text{kPa}} = 1.452\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{AV_1}{\nu_1} = \frac{\pi D^2}{4}\frac{V_1}{\nu_1} = \frac{\pi(2.5\ \text{m})^2}{4}\frac{200\ \text{m/s}}{1.452\ \text{m}^3/\text{kg}} = 676.1\ \text{kg/s}$$



Now,

$$\dot{m}_e = \frac{\dot{m}}{8} = \frac{676.1\ \text{kg/s}}{8} = 84.51\ \text{kg/s}$$

The mass flow rate through the fan is

$$\dot{m}_f = \dot{m} - \dot{m}_e = 676.1 - 84.51 = 591.6\ \text{kg/s}$$

In order to produce the specified thrust force, the velocity at the fan exit will be

$$F = \dot{m}_f (V_{\text{exit}} - V_{\text{inlet}})$$

$$V_{\text{exit}} = V_{\text{inlet}} + \frac{F}{\dot{m}_f} = (200\ \text{m/s}) + \frac{50{,}000\ \text{N}}{591.6\ \text{kg/s}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 284.5\ \text{m/s}$$

An energy balance on the stream passing through the fan gives

$$c_p(T_4 - T_5) = \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2}$$

$$T_5 = T_4 - \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2c_p}$$

$$= 253\ \text{K} - \frac{(284.5\ \text{m/s})^2 - (200\ \text{m/s})^2}{2(1.005\ \text{kJ/kg}\cdot\text{K})}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$= \mathbf{232.6\ K}$$

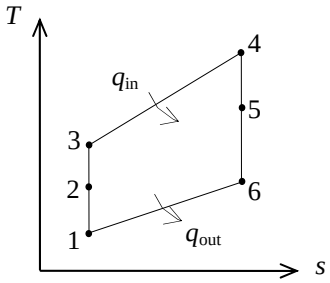
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9-142 A pure jet engine operating on an ideal cycle is considered. The velocity at the nozzle exit and the thrust produced are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are $R = 0.287$ kPa·m^3/kg·K, $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).

Analysis (*a*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1 = 240$ m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\,\nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 260\text{ K} + \frac{(240\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 288.7\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (45\text{ kPa})\left(\frac{288.7\text{ K}}{260\text{ K}}\right)^{1.4/0.4} = 64.88\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (13)(64.88\text{ kPa}) = 843.5\text{ kPa}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (288.7\text{ K})(13)^{0.4/1.4} = 600.7\text{ K}$$

Turbine:

$$w_{comp,in} = w_{turb,out} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or $$T_5 = T_4 - T_3 + T_2 = 830 - 600.7 + 288.7 = 518.0\text{ K}$$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (830\text{ K})\left(\frac{45\text{ kPa}}{843.5\text{ kPa}}\right)^{0.4/1.4} = 359.3\text{ K}$$

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\,\nearrow 0}}{2} \longrightarrow 0 = c_p(T_6 - T_5) + V_6^2/2$$

or $$V_6 = V_{exit} = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(518.0 - 359.3)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{564.8\text{ m/s}}$$

The mass flow rate through the engine is

$$\nu_1 = \frac{RT}{P} = \frac{(0.287\text{ kPa}\cdot\text{m}^3)(260\text{ K})}{45\text{ kPa}} = 1.658\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{AV_1}{\nu_1} = \frac{\pi D^2}{4}\frac{V_1}{\nu_1} = \frac{\pi(1.6\text{ m})^2}{4}\frac{240\text{ m/s}}{1.658\text{ m}^3/\text{kg}} = 291.0\text{ kg/s}$$

The thrust force generated is then

$$F = \dot{m}(V_{exit} - V_{inlet}) = (291.0\text{ kg/s})(564.8 - 240)\text{m/s}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{94{,}520\text{ N}}$$

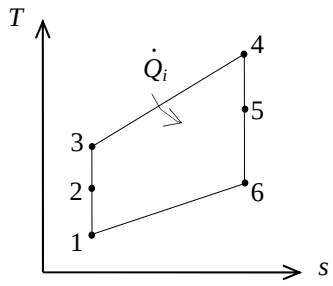
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9-143 A turbojet aircraft flying at an altitude of 9150 m is operating on the ideal jet propulsion cycle. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit. **5** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg.K and $k = 1.4$ (Table A-2a).

Analysis (*a*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1 = 320$ m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\,\nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 241\text{ K} + \frac{(320\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 291.9\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (32\text{ kPa})\left(\frac{291.9\text{ K}}{241\text{ K}}\right)^{1.4/0.4} = 62.6\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (12)(62.6\text{ kPa}) = 751.2\text{ kPa}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (291.9\text{ K})(12)^{0.4/1.4} = 593.7\text{ K}$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or $$T_5 = T_4 - T_3 + T_2 = 1400 - 593.7 + 291.9 = 1098.2\text{K}$$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{32\text{ kPa}}{751.2\text{ kPa}}\right)^{0.4/1.4} = 568.2\text{ K}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^{2\,\nearrow 0} - V_5^2}{2} \longrightarrow 0 = c_p(T_6 - T_5) + V_6^2/2$$

or $$V_6 = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(1098.2 - 568.2)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{1032\text{ m/s}}$$

(*b*) $$\dot{W}_p = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}} = (60\text{ kg/s})(1032 - 320)\text{m/s}(320\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{13{,}670\text{ kW}}$$

(*c*) $$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 593.7)\text{K} = 48{,}620\text{ kJ/s}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}_{\text{in}}}{\text{HV}} = \frac{48{,}620\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.14\text{ kg/s}}$$

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9-144 A turbojet aircraft is flying at an altitude of 9150 m. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a).

Analysis (*a*) For convenience, we assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 320 m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).

Diffuser:

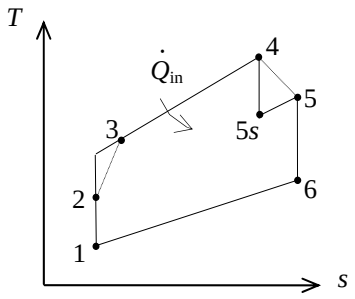
$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

$$0 = h_2 - h_1 + \frac{V_2^{2\,\nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$



$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 241\text{ K} + \frac{(320\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 291.9\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (32\text{ kPa})\left(\frac{291.9\text{ K}}{241\text{ K}}\right)^{1.4/0.4} = 62.6\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (12)(62.6\text{ kPa}) = 751.2\text{ kPa}$$

$$T_{3s} = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (291.9\text{ K})(12)^{0.4/1.4} = 593.7\text{ K}$$

$$\eta_C = \frac{h_{3s} - h_2}{h_3 - h_2} = \frac{c_p(T_{3s} - T_2)}{c_p(T_3 - T_2)}$$

$$T_3 = T_2 + (T_{3s} - T_2)/\eta_C = 291.9 + (593.7 - 291.9)/(0.80) = 669.2\text{ K}$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or,

$$T_5 = T_4 - T_3 + T_2 = 1400 - 669.2 + 291.9 = 1022.7\text{ K}$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} = \frac{c_p(T_4 - T_5)}{c_p(T_4 - T_{5s})}$$

$$T_{5s} = T_4 - (T_4 - T_5)/\eta_T = 1400 - (1400 - 1022.7)/0.85 = 956.1\text{ K}$$

$$P_5 = P_4\left(\frac{T_{5s}}{T_4}\right)^{k/(k-1)} = (751.2\text{ kPa})\left(\frac{956.1\text{ K}}{1400\text{ K}}\right)^{1.4/0.4} = 197.7\text{ kPa}$$

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Nozzle:

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (1022.7\text{ K})\left(\frac{32\text{ kPa}}{197.7\text{ kPa}}\right)^{0.4/1.4} = 607.8\text{ K}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\,\nearrow 0}}{2}$$

$$0 = c_p(T_6 - T_5) + V_6^2/2$$

or,

$$V_6 = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(1022.7 - 607.8)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{913.2\ m/s}$$

(*b*) $$\dot{W}_p = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$

$$= (60\text{ kg/s})(913.2 - 320)\text{m/s}(320\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$= \mathbf{11{,}390\ kW}$$

(*c*) $$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 669.2)\text{K} = 44{,}067\text{ kJ/s}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}_{\text{in}}}{\text{HV}} = \frac{44{,}067\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.03\ kg/s}$$

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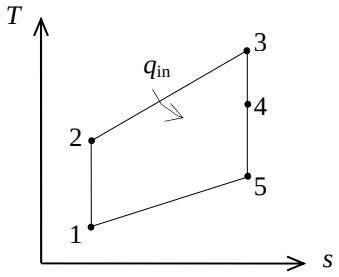
9-145 A turbojet aircraft that has a pressure rate of 9 is stationary on the ground. The force that must be applied on the brakes to hold the plane stationary is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the nozzle exit.

Properties The properties of air are given in Table A-17.

Analysis (*a*) Using variable specific heats for air,

Compressor:



$$T_1 = 290\ \text{K} \longrightarrow h_1 = 290.16\ \text{kJ/kg}$$
$$P_{r_1} = 1.2311$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (9)(1.2311) = 11.08 \longrightarrow h_2 = 544.07\ \text{kJ/kg}$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{fuel}} \times \text{HV} = (0.5\ \text{kg/s})(42{,}700\ \text{kJ/kg}) = 21{,}350\ \text{kJ/s}$$

$$q_{\text{in}} = \frac{\dot{Q}_{\text{in}}}{\dot{m}} = \frac{21{,}350\ \text{kJ/s}}{20\ \text{kg/s}} = 1067.5\ \text{kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 \longrightarrow h_3 = h_2 + q_{in} = 544.07 + 1067.5 = 1611.6\ \text{kJ/kg} \longrightarrow P_{r_3} = 568.5$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_2 - h_1 = h_3 - h_4$$

or

$$h_4 = h_3 - h_2 + h_1 = 1611.6 - 544.07 + 290.16 = 1357.7\ \text{kJ/kg}$$

Nozzle:

$$P_{r_5} = P_{r_3}\left(\frac{P_5}{P_3}\right) = (568.5)\left(\frac{1}{9}\right) = 63.17 \longrightarrow h_5 = 888.56\ \text{kJ/kg}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$h_4 + V_4^2/2 = h_5 + V_5^2/2$$
$$0 = h_5 - h_4 + \frac{V_5^2 - V_4^{2\nearrow 0}}{2}$$

or

$$V_5 = \sqrt{2(h_4 - h_5)} = \sqrt{(2)(1357.7 - 888.56)\text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 968.6\ \text{m/s}$$

$$\text{Brake force} = \text{Thrust} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}}) = (20\ \text{kg/s})(968.6 - 0)\text{m/s}\left(\frac{1\ \text{N}}{1\ \text{kg} \cdot \text{m/s}^2}\right) = \mathbf{19{,}370\ N}$$

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9-146 Problem 9-145 is reconsidered. The effect of compressor inlet temperature on the force that must be applied to the brakes to hold the plane stationary is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
P_ratio =9
T_1 = 7 [C]
T[1] = T_1+273 "[K]"
P[1]= 95 [kPa]
P[5]=P[1]
Vel[1]=0 [m/s]
V_dot[1] = 18.1 [m^3/s]
HV_fuel = 42700 [kJ/kg]
m_dot_fuel = 0.5 [kg/s]
Eta_c = 1.0
Eta_t = 1.0
Eta_N = 1.0

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])
v[1]=volume(Air,T=T[1],P=P[1])
m_dot = V_dot[1]/v[1]
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c = W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"

"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
Q_dot_in = m_dot_fuel*HV_fuel
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
{P_ratio= P[3] /P[4]}
T_s[4]=TEMPERATURE(Air,h=h_s[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
{h_s[4]=ENTHALPY(Air,T=T_s[4])}  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency, Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"
T[4]=TEMPERATURE(Air,h=h[4])
P[4]=pressure(Air,s=s_s[4],h=h_s[4])
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
W_dot_net = 0 [kW]

"Exit nozzle analysis:"
s[4]=entropy('air',T=T[4],P=P[4])
s_s[5]=s[4] "For the ideal case the entropies are constant across the nozzle"

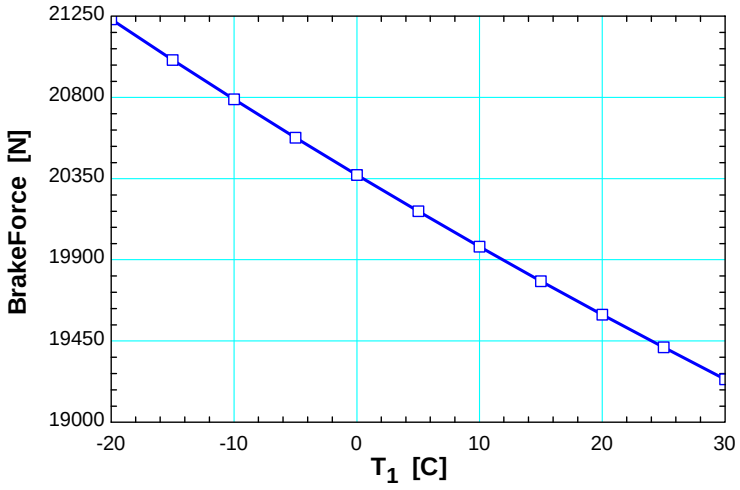
T_s[5]=TEMPERATURE(Air,s=s_s[5], P=P[5]) "T_s[5] is the isentropic value of T[5] at nozzle exit"
h_s[5]=ENTHALPY(Air,T=T_s[5])
Eta_N=(h[4]-h[5])/(h[4]-h_s[5])
m_dot*h[4] =  m_dot*(h_s[5] + Vel_s[5]^2/2*convert(m^2/s^2,kJ/kg))
```

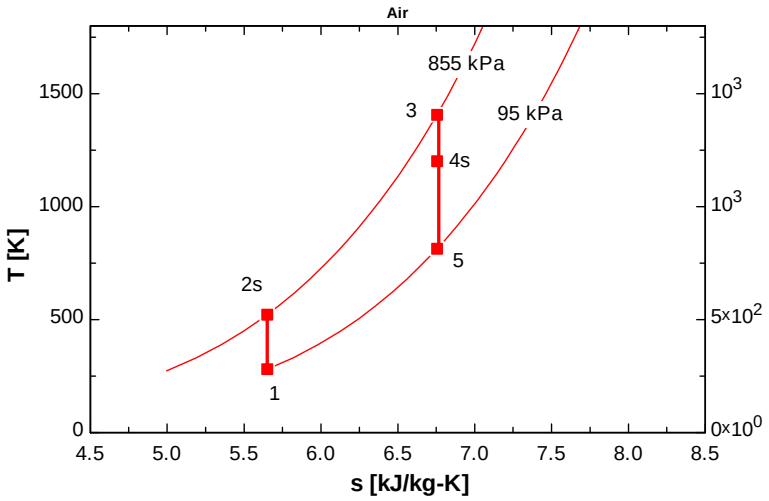
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```
m_dot*h[4] =  m_dot*(h[5] + Vel[5]^2/2*convert(m^2/s^2,kJ/kg))
T[5]=TEMPERATURE(Air,h=h[5])
s[5]=entropy('air',T=T[5],P=P[5])

"Brake Force to hold the aircraft:"
Thrust = m_dot*(Vel[5] - Vel[1]) "[N]"
BrakeForce = Thrust "[N]"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
s[2]=entropy('air',T=T[2],P=P[2])
```

Brake Force [N]	m [kg/s]	T_3 [K]	T_1 [C]
21232	23.68	1284	-20
21007	23.22	1307	-15
20788	22.78	1330	-10
20576	22.35	1352	-5
20369	21.94	1375	0
20168	21.55	1398	5
19972	21.17	1420	10
19782	20.8	1443	15
19596	20.45	1466	20
19415	20.1	1488	25
19238	19.77	1510	30





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9-147 Air enters a turbojet engine. The thrust produced by this turbojet engine is to be determined.

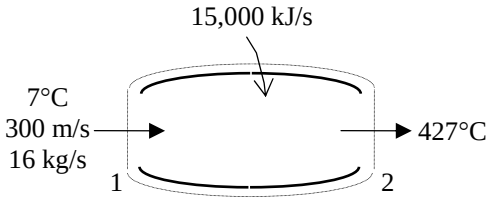
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit.

Properties The properties of air are given in Table A-17.

Analysis We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 300 m/s. Taking the entire engine as our control volume and writing the steady-flow energy balance yield

$$T_1 = 280 \text{ K} \quad \longrightarrow \quad h_1 = 280.13 \text{ kJ/kg}$$

$$T_2 = 700 \text{ K} \quad \longrightarrow \quad h_2 = 713.27 \text{ kJ/kg}$$



$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$\dot{Q}_{\text{in}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)$$

$$15{,}000 \text{ kJ/s} = (16 \text{ kg/s})\left[713.27 - 280.13 + \frac{V_2^2 - (300 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right]$$

It gives

$$V_2 = 1048 \text{ m/s}$$

Thus,

$$F_p = \dot{m}(V_2 - V_1) = (16 \text{ kg/s})(1048 - 300)\text{m/s} = \mathbf{11{,}968\ N}$$

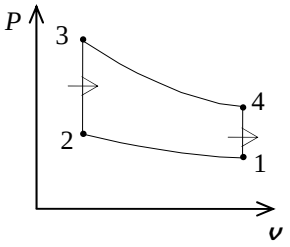
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Second-Law Analysis of Gas Power Cycles

9-148 The process with the highest exergy destruction for an ideal Otto cycle described in Prob. 9-36 is to be determined.

Analysis From Prob. 9-36, q_{in} = 582.5 kJ/kg, q_{out} = 253.6 kJ/kg, T_1 = 288 K, T_2 = 661.7 K, T_3 = 1473 K, and T_4 = 641.2 K. The exergy destruction during a process of the cycle is

$$x_{dest} = T_0 s_{gen} = T_0\left(\Delta s - \frac{q_{in}}{T_{source}} + \frac{q_{out}}{T_{sink}}\right)$$

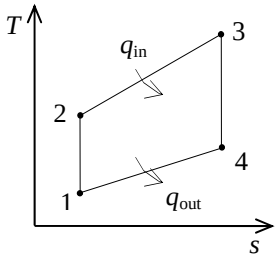


Application of this equation for each process of the cycle gives

$$x_{dest,1\text{-}2} = \mathbf{0} \quad \text{(isentropic process)}$$

$$s_3 - s_2 = s_4 - s_1 = c_v \ln\frac{T_3}{T_2} + R\ln\frac{v_3}{v_2}$$

$$= (0.718\ \text{kJ/kg}\cdot\text{K})\ln\frac{1473\ \text{K}}{661.7\ \text{K}} + 0 = 0.5746\ \text{kJ/kg}\cdot\text{K}$$



$$x_{dest,2\text{-}3} = T_0\left(s_3 - s_2 - \frac{q_{in}}{T_{source}}\right)$$

$$= (288\ \text{K})\left(0.5746\ \text{kJ/kg}\cdot\text{K} - \frac{582.5\ \text{kJ/kg}}{1473\ \text{K}}\right)$$

$$= \mathbf{51.59\ kJ/kg}$$

$$x_{dest,3\text{-}4} = \mathbf{0} \quad \text{(isentropic process)}$$

$$x_{dest,4\text{-}1} = T_0\left(s_1 - s_4 + \frac{q_{out}}{T_{sink}}\right)$$

$$= (288\ \text{K})\left(-0.5746\ \text{kJ/kg}\cdot\text{K} + \frac{253.6\ \text{kJ/kg}}{288\ \text{K}}\right)$$

$$= \mathbf{88.12\ kJ/kg}$$

The largest exergy destruction in the cycle occurs during the heat-rejection process.

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9-149E The exergy destruction associated with the heat rejection process of the Diesel cycle described in Prob. 9-55E and the exergy at the end of the expansion stroke are to be determined.

Analysis From Prob. 9-55E, q_{out} = 158.9 Btu/lbm, T_1 = 540 R, P_1 = 14.7 psia, T_4 = 1420.6 R, P_4 = 38.62 psia and $v_4 = v_1$.

The entropy change during process 4-1 is

$$\begin{aligned} s_1 - s_4 &= s^{\text{o}}_{1\ @540\text{R}} - s^{\text{o}}_{4\ @1420.6\text{ R}} - R\ln(P_1 / P_4) \\ &= 0.60078 - 0.83984 - (0.06855)\ln(14.7/38.62) \\ &= -0.1728\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

Thus,

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (540\text{R})\left(-0.1728\ \text{Btu/lbm}\cdot\text{R} + \frac{158.9\ \text{Btu/lbm}}{540\ \text{R}}\right) = \mathbf{65.6\ \ Btu/lbm}$$

Noting that state 4 is identical to the state of the surroundings, the exergy at the end of the power stroke (state 4) is determined from

$$\phi_4 = (u_4 - u_0) - T_0(s_4 - s_0) + P_0(v_4 - v_0)$$

where

$$\begin{aligned} u_4 - u_0 &= u_4 - u_1 = q_{\text{out}} = 158.9\ \text{Btu/lbm}\cdot\text{R} \\ v_4 - v_0 &= v_4 - v_1 = 0 \\ s_4 - s_0 &= s_4 - s_1 = 0.1741\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

Thus,

$$\phi_4 = (158.9\ \text{Btu/lbm}) - (540\text{R})(0.1728\ \text{Btu/lbm}\cdot\text{R}) + 0 = \mathbf{65.6\ Btu/lbm}$$

Discussion Note that the exergy at state 4 is identical to the exergy destruction for the process 4-1 since state 1 is identical to the dead state, and the entire exergy at state 4 is wasted during process 4-1.

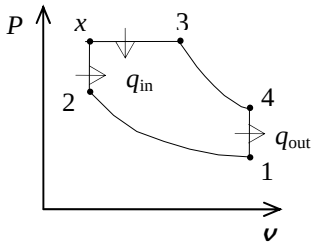
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9-150 The exergy loss of each process for an ideal dual cycle described in Prob. 9-63 is to be determined.

Analysis From Prob. 9-63, $q_{\text{in},x\text{-}3}$ = 114.6 kJ/kg, T_1 = 291 K, T_2 = 1037 K, T_x = 1141 K, T_3 = 1255 K, and T_4 = 494.8 K. Also,

$$q_{\text{in},2-x} = c_v(T_x - T_2) = (0.718\ \text{kJ/kg}\cdot\text{K})(1141-1037)\text{K} = 74.67\ \text{kJ/kg}$$

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.718\ \text{kJ/kg}\cdot\text{K})(494.8-291)\text{K} = 146.3\ \text{kJ/kg}$$



The exergy destruction during a process of the cycle is

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(\Delta s - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.005\ \text{kJ/kg}\cdot\text{K})\ln\frac{1037\ \text{K}}{291\ \text{K}} - (0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{5148\ \text{kPa}}{90\ \text{kPa}} \\ &= 0.1158\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$x_{\text{dest, 1-2}} = T_0(s_2 - s_1) = (291\ \text{K})(0.1158\ \text{kJ/kg}\cdot\text{K}) = \mathbf{33.7\ kJ/kg}$$

$$\begin{aligned} s_x - s_2 &= c_v \ln\frac{T_x}{T_2} + R\ln\frac{v_x}{v_2} \\ &= (0.718\ \text{kJ/kg}\cdot\text{K})\ln\frac{1141\ \text{K}}{1037\ \text{K}} + 0 = 0.06862\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$x_{\text{dest, 2-}x} = T_0\left(s_x - s_2 - \frac{q_{\text{in, 2-}x}}{T_{\text{source}}}\right) = (291\ \text{K})\left(0.06862\ \text{kJ/kg}\cdot\text{K} - \frac{74.67\ \text{kJ/kg}}{1255\ \text{K}}\right) = \mathbf{2.65\ kJ/kg}$$

$$s_3 - s_x = c_p \ln\frac{T_3}{T_x} - R\ln\frac{P_3}{P_x} = (1.005\ \text{kJ/kg}\cdot\text{K})\ln\frac{1255\ \text{K}}{1141\ \text{K}} - 0 = 0.09571\ \text{kJ/kg}\cdot\text{K}$$

$$x_{\text{dest, }x\text{-3}} = T_0\left(s_3 - s_x - \frac{q_{\text{in, }x\text{-3}}}{T_{\text{source}}}\right) = (291\ \text{K})\left(0.09571\ \text{kJ/kg}\cdot\text{K} - \frac{114.6\ \text{kJ/kg}}{1255\ \text{K}}\right) = \mathbf{1.28\ kJ/kg}$$

$$\begin{aligned} s_4 - s_3 &= c_v \ln\frac{T_4}{T_3} + R\ln\frac{v_4}{v_3} = c_v \ln\frac{T_4}{T_3} + R\ln\frac{r}{r_c} \\ &= (0.718\ \text{kJ/kg}\cdot\text{K})\ln\frac{494.8\ \text{K}}{1255\ \text{K}} + (0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{18}{1.1} = 0.1339\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$x_{\text{dest, 3-4}} = T_0(s_4 - s_3) = (291\ \text{K})(0.1339\ \text{kJ/kg}\cdot\text{K}) = \mathbf{39.0\ kJ/kg}$$

$$s_1 - s_4 = c_v \ln\frac{T_1}{T_4} + R\ln\frac{v_1}{v_4} = (0.718\ \text{kJ/kg}\cdot\text{K})\ln\frac{291\ \text{K}}{494.8\ \text{K}} + 0 = -0.3811\ \text{kJ/kg}\cdot\text{K}$$

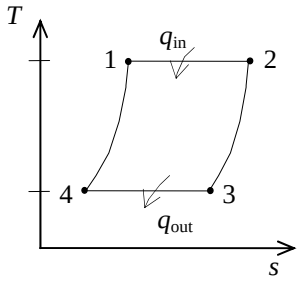
$$x_{\text{dest,4-1}} = T_0\left(s_1 - s_4 + \frac{q_{\text{out}}}{T_{\text{sink}}}\right) = (291\ \text{K})\left(-0.3811\ \text{kJ/kg}\cdot\text{K} + \frac{146.3\ \text{kJ/kg}}{291\ \text{K}}\right) = \mathbf{35.4\ kJ/kg}$$

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9-151 The exergy loss of each process for an air-standard Stirling cycle described in Prob. 9-81 is to be determined.

Analysis From Prob. 9-81, q_{in} = 1275 kJ/kg, q_{out} = 212.5 kJ/kg, $T_1 = T_2$ = 1788 K, $T_3 = T_4$ = 298 K. The exergy destruction during a process of the cycle is

$$x_{dest} = T_0 s_{gen} = T_0\left(\Delta s - \frac{q_{in}}{T_{source}} + \frac{q_{out}}{T_{sink}}\right)$$



Application of this equation for each process of the cycle gives

$$s_2 - s_1 = c_v \ln\frac{T_2}{T_1} + R\ln\frac{v_2}{v_1}$$
$$= 0 + (0.287\ \text{kJ/kg}\cdot\text{K})\ln(12) = 0.7132\ \text{kJ/kg}\cdot\text{K}$$

$$x_{dest,1\text{-}2} = T_0\left(s_2 - s_1 - \frac{q_{in}}{T_{source}}\right)$$
$$= (298\ \text{K})\left(0.7132\ \text{kJ/kg}\cdot\text{K} - \frac{1275\ \text{kJ/kg}}{1788\ \text{K}}\right) = \mathbf{0.034\ kJ/kg \approx 0}$$

$$s_4 - s_3 = c_v \ln\frac{T_4}{T_3} + R\ln\frac{v_4}{v_3}$$
$$= 0 + (0.287\ \text{kJ/kg}\cdot\text{K})\ln\left(\frac{1}{12}\right) = -0.7132\ \text{kJ/kg}\cdot\text{K}$$

$$x_{dest,3\text{-}4} = T_0\left(s_4 - s_3 + \frac{q_{out}}{T_{sink}}\right)$$
$$= (298\ \text{K})\left(-0.7132\ \text{kJ/kg}\cdot\text{K} + \frac{212.5\ \text{kJ/kg}}{298\ \text{K}}\right) = \mathbf{-0.034\ kJ/kg \approx 0}$$

These results are not surprising since Stirling cycle is totally reversible. Exergy destructions are not calculated for processes 2-3 and 4-1 because there is no interaction with the surroundings during these processes to alter the exergy destruction.

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9-152 The exergy destruction associated with each of the processes of the Brayton cycle described in Prob. 9-89 is to be determined.

Analysis From Prob. 9-89, q_{in} = 698.3 kJ/kg, q_{out} = 487.9 kJ/kg, and

$$T_1 = 295\ \text{K} \longrightarrow s_1^\circ = 1.68515\ \text{kJ/kg}\cdot\text{K}$$
$$h_2 = 626.60\ \text{kJ/kg} \longrightarrow s_2^\circ = 2.44117\ \text{kJ/kg}\cdot\text{K}$$
$$T_3 = 1240\ \text{K} \longrightarrow s_3^\circ = 3.21751\ \text{kJ/kg}\cdot\text{K}$$
$$h_4 = 783.04\ \text{kJ/kg} \longrightarrow s_4^\circ = 2.66807\ \text{kJ/kg}\cdot\text{K}$$

Thus,

$$x_{\text{destroyed},12} = T_0 s_{\text{gen},12} = T_0(s_2 - s_1) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right) =$$
$$= (310\ \text{K})(2.44117 - 1.68515 - (0.287\ \text{kJ/kg}\cdot\text{K})\ln(10)) = \mathbf{29.51\ kJ/kg}$$
$$x_{\text{destroyed},23} = T_0 s_{\text{gen},23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = T_0\left(s_3^\circ - s_2^\circ - R\ln\frac{P_3}{P_2}^{\nearrow 0} + \frac{-q_{\text{in}}}{T_H}\right)$$
$$= (310\ \text{K})\left(3.21751 - 2.44117 - \frac{698.3\ \text{kJ/kg}}{1600\ \text{K}}\right) = \mathbf{105.4\ kJ/kg}$$
$$x_{\text{destroyed},34} = T_0 s_{\text{gen},34} = T_0(s_4 - s_3) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right) =$$
$$= (310\ \text{K})(2.66807 - 3.21751 - (0.287\ \text{kJ/kg}\cdot\text{K})\ln(1/10)) = \mathbf{34.53\ kJ/kg}$$
$$x_{\text{destroyed},41} = T_0 s_{\text{gen},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = T_0\left(s_1^\circ - s_4^\circ - R\ln\frac{P_1}{P_4}^{\nearrow 0} + \frac{q_{\text{out}}}{T_L}\right)$$
$$= (310\ \text{K})\left(1.68515 - 2.66807 + \frac{487.9\ \text{kJ/kg}}{310\ \text{K}}\right) = \mathbf{183.2\ kJ/kg}$$

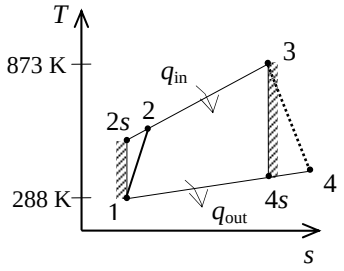
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9-153 Exergy analysis is to be used to answer the question in Prob. 9-94.

Analysis From Prob. 9-94, $T_1 = 288$ K, $T_{2s} = 585.8$ K, $T_2 = 618.9$ K, $T_3 = 873$ K, $T_{4s} = 429.2$ K, $T_4 = 473.6$ K, $r_p = 12$. The exergy change of a flow stream between an inlet and exit state is given by

$$\Delta\psi = h_e - h_i - T_0(s_e - s_i)$$

This is also the expression for reversible work. Application of this equation for isentropic and actual compression processes gives



$$s_{2s} - s_1 = c_p \ln\frac{T_{2s}}{T_1} - R\ln\frac{P_2}{P_1}$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{585.8\text{ K}}{288\text{ K}} - (0.287\text{ kJ/kg}\cdot\text{K})\ln(12)$$
$$= 0.0003998\text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},1-2s} = c_p(T_{2s} - T_1) - T_0(s_{2s} - s_1)$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})(585.8 - 288)\text{K} - (288\text{ K})(0.0003998\text{ kJ/kg}\cdot\text{K}) = 299.2\text{ kJ/kg}$$

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{618.9\text{ K}}{288\text{ K}} - (0.287\text{ kJ/kg}\cdot\text{K})\ln(12) = 0.05564\text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},1-2} = c_p(T_2 - T_1) - T_0(s_2 - s_1)$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})(618.9 - 288)\text{K} - (288\text{ K})(0.05564\text{ kJ/kg}\cdot\text{K}) = 316.5\text{ kJ/kg}$$

The irreversibilities therefore increase the minimum work that must be supplied to the compressor by

$$\Delta w_{\text{rev,C}} = w_{\text{rev},1-2} - w_{\text{rev},1-2s} = 316.5 - 299.2 = \mathbf{17.3\text{ kJ/kg}}$$

Repeating the calculations for the turbine,

$$s_3 - s_{4s} = c_p \ln\frac{T_3}{T_{4s}} - R\ln\frac{P_3}{P_4}$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{873\text{ K}}{429.2\text{K}} - (0.287\text{ kJ/kg}\cdot\text{K})\ln(12) = 0.0003944\text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},3-4s} = c_p(T_3 - T_{4s}) - T_0(s_3 - s_{4s})$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})(873 - 429.2)\text{K} - (288\text{ K})(0.0003944\text{ kJ/kg}\cdot\text{K}) = 445.9\text{ kJ/kg}$$

$$s_3 - s_4 = c_p \ln\frac{T_3}{T_4} - R\ln\frac{P_3}{P_4}$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{873\text{ K}}{473.6\text{K}} - (0.287\text{ kJ/kg}\cdot\text{K})\ln(12) = -0.09854\text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},3-4} = c_p(T_3 - T_{4s}) - T_0(s_3 - s_{4s})$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})(873 - 473.6)\text{K} - (288\text{ K})(-0.09854\text{ kJ/kg}\cdot\text{K}) = 429.8\text{ kJ/kg}$$

$$\Delta w_{\text{rev,T}} = w_{\text{rev},3-4s} - w_{\text{rev},3-4} = 445.9 - 429.8 = \mathbf{16.1\text{ kJ/kg}}$$

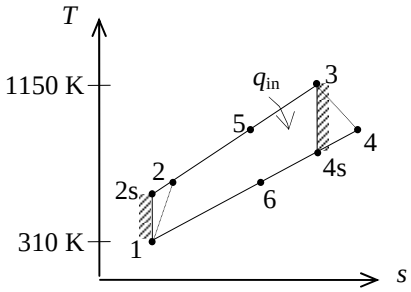
Hence, it is clear that the compressor is a little more sensitive to the irreversibilities than the turbine.

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9-154 The total exergy destruction associated with the Brayton cycle described in Prob. 9-116 and the exergy at the exhaust gases at the turbine exit are to be determined.

Properties The gas constant of air is R = 0.287 kJ/kg·K (Table A-1).

Analysis From Prob. 9-116, q_{in} = 480.82, q_{out} = 372.73 kJ/kg, and



$$T_1 = 310\ \text{K} \longrightarrow s_1^\circ = 1.73498\ \text{kJ/kg}\cdot\text{K}$$
$$h_2 = 618.26\ \text{kJ/kg} \longrightarrow s_2^\circ = 2.42763\ \text{kJ/kg}\cdot\text{K}$$
$$T_3 = 1150\ \text{K} \longrightarrow s_3^\circ = 3.12900\ \text{kJ/kg}\cdot\text{K}$$
$$h_4 = 803.14\ \text{kJ/kg} \longrightarrow s_4^\circ = 2.69407\ \text{kJ/kg}\cdot\text{K}$$
$$h_5 = 738.43\ \text{kJ/kg} \longrightarrow s_5^\circ = 2.60815\ \text{kJ/kg}\cdot\text{K}$$

and, from an energy balance on the heat exchanger,

$$h_5 - h_2 = h_4 - h_6 \longrightarrow h_6 = 803.14 - (738.43 - 618.26) = 682.97\ \text{kJ/kg}$$
$$\longrightarrow s_6^\circ = 2.52861\ \text{kJ/kg}\cdot\text{K}$$

Thus,

$$x_{\text{destroyed},12} = T_0 s_{\text{gen},12} = T_0(s_2 - s_1) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$
$$= (290\ \text{K})(2.42763 - 1.73498 - (0.287\ \text{kJ/kg}\cdot\text{K})\ln(7)) = \mathbf{38.91\ kJ/kg}$$
$$x_{\text{destroyed},34} = T_0 s_{\text{gen},34} = T_0(s_4 - s_3) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right)$$
$$= (290\ \text{K})(2.69407 - 3.12900 - (0.287\text{kJ/kg}\cdot\text{K})\ln(1/7)) = \mathbf{35.83\ kJ/kg}$$
$$x_{\text{destroyed, regen}} = T_0 s_{\text{gen,regen}} = T_0[(s_5 - s_2) + (s_6 - s_4)] = T_0[(s_5^\circ - s_2^\circ) + (s_6^\circ - s_4^\circ)]$$
$$= (290\ \text{K})(2.60815 - 2.42763 + 2.52861 - 2.69407) = \mathbf{4.37\ kJ/kg}$$
$$x_{\text{destroyed},53} = T_0 s_{\text{gen},53} = T_0\left(s_3 - s_5 - \frac{q_{R,53}}{T_R}\right) = T_0\left(s_3^\circ - s_5^\circ - R\ln\frac{P_3}{P_5}^{\nearrow 0} - \frac{q_{in}}{T_H}\right)$$
$$= (290\ \text{K})\left(3.12900 - 2.60815 - \frac{480.82\ \text{kJ/kg}}{1500\ \text{K}}\right) = \mathbf{58.09\ kJ/kg}$$
$$x_{\text{destroyed},61} = T_0 s_{\text{gen},61} = T_0\left(s_1 - s_6 + \frac{q_{R,61}}{T_R}\right) = T_0\left(s_1^\circ - s_6^\circ - R\ln\frac{P_1}{P_6}^{\nearrow 0} + \frac{q_{\text{out}}}{T_L}\right)$$
$$= (290\ \text{K})\left(1.73498 - 2.52861 + \frac{372.73\ \text{kJ/kg}}{290\ \text{K}}\right) = \mathbf{142.6\ kJ/kg}$$

Noting that $h_0 = h_{@\ 290\ \text{K}}$ = 290.16 kJ/kg and $T_0 = 290\ \text{K} \longrightarrow s_1^\circ = 1.66802\ \text{kJ/kg}\cdot\text{K}$, the stream exergy at the exit of the regenerator (state 6) is determined from

$$\phi_6 = (h_6 - h_0) - T_0(s_6 - s_0) + \frac{V_6^{2\ \nearrow 0}}{2} + gz_6^{\nearrow 0}$$

where

$$s_6 - s_0 = s_6 - s_1 = s_6^\circ - s_1^\circ - R\ln\frac{P_6}{P_1}^{\nearrow 0} = 2.52861 - 1.66802 = 0.86059\ \text{kJ/kg}\cdot\text{K}$$

Thus,

$$\phi_6 = 682.97 - 290.16 - (290\ \text{K})(0.86059\ \text{kJ/kg}\cdot\text{K}) = \mathbf{143.2\ kJ/kg}$$

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9-155 Prob. 9-154 is reconsidered. The effect of the cycle pressure on the total irreversibility for the cycle and the exergy of the exhaust gas leaving the regenerator is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
"Given"
T[1]=310 [K]
P[1]=100 [kPa]
Ratio_P=7
P[2]=Ratio_P*P[1]
T[3]=1150 [K]
eta_C=0.75
eta_T=0.82
epsilon=0.65
T_H=1500 [K]
T0=290 [K]
P0=100 [kPa]

"Analysis for Problem 9-154"
q_in=h[3]-h[5]
q_out=h[6]-h[1]
h[5]-h[2]=h[4]-h[6]
s[2]=entropy(Fluid$, P=P[2], h=h[2])
s[4]=entropy(Fluid$, h=h[4], P=P[4])
s[5]=entropy(Fluid$, h=h[5], P=P[5])
P[5]=P[2]
s[6]=entropy(Fluid$, h=h[6], P=P[6])
P[6]=P[1]
h[0]=enthalpy(Fluid$, T=T0)
s[0]=entropy(Fluid$, T=T0, P=P0)
x_destroyed_12=T0*(s[2]-s[1])
x_destroyed_34=T0*(s[4]-s[3])
x_destroyed_regen=T0*(s[5]-s[2]+s[6]-s[4])
x_destroyed_53=T0*(s[3]-s[5]-q_in/T_H)
x_destroyed_61=T0*(s[1]-s[6]+q_out/T0)
x_total=x_destroyed_12+x_destroyed_34+x_destroyed_regen+x_destroyed_53+x_destroyed_61
x6=h[6]-h[0]-T0*(s[6]-s[0]) "since state 0 and state 1 are identical"

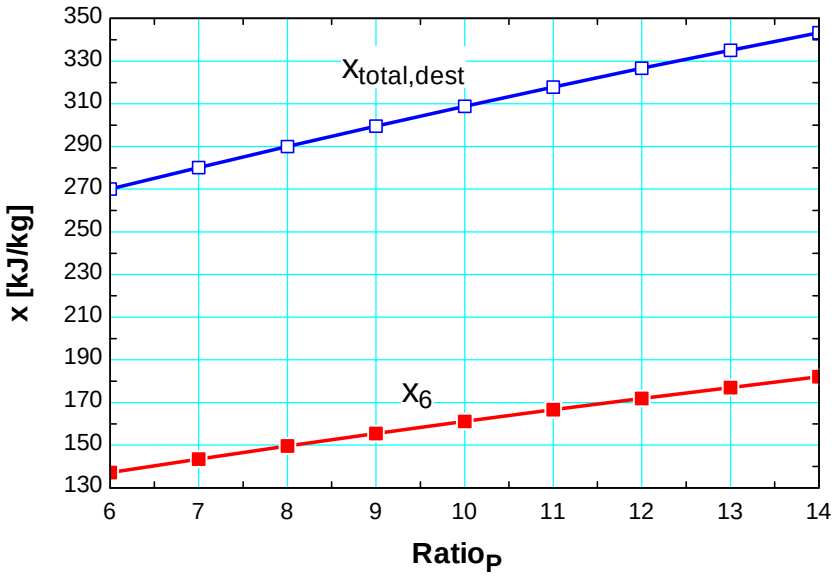
"Analysis for Problem 9-116"
Fluid$='air'
"(a)"
h[1]=enthalpy(Fluid$, T=T[1])
s[1]=entropy(Fluid$, T=T[1], P=P[1])
s_s[2]=s[1] "isentropic compression"
h_s[2]=enthalpy(Fluid$, P=P[2], s=s_s[2])
eta_C=(h_s[2]-h[1])/(h[2]-h[1])
h[3]=enthalpy(Fluid$, T=T[3])
s[3]=entropy(Fluid$, T=T[3], P=P[3])
P[3]=P[2]
s_s[4]=s[3] "isentropic expansion"
h_s[4]=enthalpy(Fluid$, P=P[4], s=s_s[4])
P[4]=P[1]
eta_T=(h[3]-h[4])/(h[3]-h_s[4])
q_regen=epsilon*(h[4]-h[2])

"(b)"
w_C_in=(h[2]-h[1])
w_T_out=h[3]-h[4]
```

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w_net_out=w_T_out-w_C_in
q_in=(h[3]-h[2])-q_regen
eta_th=w_net_out/q_in

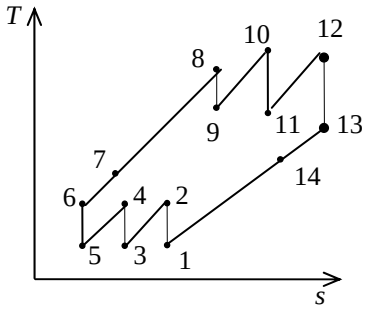
Ratio_P	x_{total} [kJ/kg]	x6 [kJ/kg]
6	270.1	137.2
7	280	143.5
8	289.9	149.6
9	299.5	155.5
10	308.8	161.1
11	317.8	166.6
12	326.6	171.9
13	335.1	177.1
14	343.3	182.1



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9-156 The exergy loss of each process for a regenerative Brayton cycle with three stages of reheating and intercooling described in Prob. 9-135 is to be determined.

Analysis From Prob. 9-135,



$r_p = 4$, $q_{\text{in,7-8}} = q_{\text{in,9-10}} = q_{\text{in,11-12}} = 300$ kJ/kg,

$q_{\text{out,14-1}} = 206.9$ kJ/kg, $q_{\text{out,2-3}} = q_{\text{out,4-5}} = 141.6$ kJ/kg,

$T_1 = T_3 = T_5 = 290$ K , $T_2 = T_4 = T_6 = 430.9$ K

$T_7 = 520.7$ K, $T_8 = 819.2$ K, $T_9 = 551.3$ K

$T_{10} = 849.8$ K, $T_{11} = 571.9$ K, $T_{12} = 870.4$ K,

$T_{13} = 585.7$ K, $T_{14} = 495.9$ K

The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0 \left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}} \right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{dest,1-2}} = x_{\text{dest, 3-4}} = x_{\text{dest, 5-6}} = T_0 \left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right)$$

$$= (290) \left[(1.005) \ln \frac{430.9}{290} - (0.287) \ln(4) \right] = \mathbf{0.03\ kJ/kg \approx 0}$$

$$x_{\text{dest, 7-8}} = T_0 \left(c_p \ln \frac{T_8}{T_7} - R \ln \frac{P_8}{P_7} - \frac{q_{\text{in,7-8}}}{T_{\text{source}}} \right) = (290) \left[(1.005) \ln \frac{819.2}{520.7} - 0 - \frac{300}{870.4} \right] = \mathbf{32.1\ kJ/kg}$$

$$x_{\text{dest, 9-10}} = T_0 \left(c_p \ln \frac{T_{10}}{T_9} - R \ln \frac{P_8}{P_7} - \frac{q_{\text{in,9-10}}}{T_{\text{source}}} \right) = (290) \left[(1.005) \ln \frac{849.8}{551.3} - 0 - \frac{300}{870.4} \right] = \mathbf{26.2\ kJ/kg}$$

$$x_{\text{dest,11-12}} = T_0 \left(c_p \ln \frac{T_{12}}{T_{11}} - R \ln \frac{P_{12}}{P_{11}} - \frac{q_{\text{in,11-12}}}{T_{\text{source}}} \right) = (290) \left[(1.005) \ln \frac{870.4}{571.9} - 0 - \frac{300}{870.4} \right] = \mathbf{22.5\ kJ/kg}$$

$$x_{\text{dest, 8-9}} = T_0 \left(c_p \ln \frac{T_9}{T_8} - R \ln \frac{P_9}{P_8} \right) = (290) \left[(1.005) \ln \frac{551.3}{819.2} - (0.287) \ln \left(\frac{1}{4} \right) \right] = -\mathbf{0.05\ kJ/kg \approx 0}$$

$$x_{\text{dest,10-11}} = T_0 \left(c_p \ln \frac{T_{11}}{T_{10}} - R \ln \frac{P_{11}}{P_{10}} \right) = (290) \left[(1.005) \ln \frac{571.9}{849.8} - (0.287) \ln \left(\frac{1}{4} \right) \right] = -\mathbf{0.04\ kJ/kg \approx 0}$$

$$x_{\text{dest,12-13}} = T_0 \left(c_p \ln \frac{T_{13}}{T_{12}} - R \ln \frac{P_{13}}{P_{12}} \right) = (290) \left[(1.005) \ln \frac{585.7}{870.4} - (0.287) \ln \left(\frac{1}{4} \right) \right] = -\mathbf{0.08\ kJ/kg \approx 0}$$

$$x_{\text{dest,14-1}} = T_0 \left(c_p \ln \frac{T_1}{T_{14}} - R \ln \frac{P_1}{P_{14}} + \frac{q_{\text{out,14-1}}}{T_{\text{sink}}} \right) = (290) \left[(1.005) \ln \frac{290}{495.9} - 0 + \frac{206.9}{290} \right] = \mathbf{50.6\ kJ/kg}$$

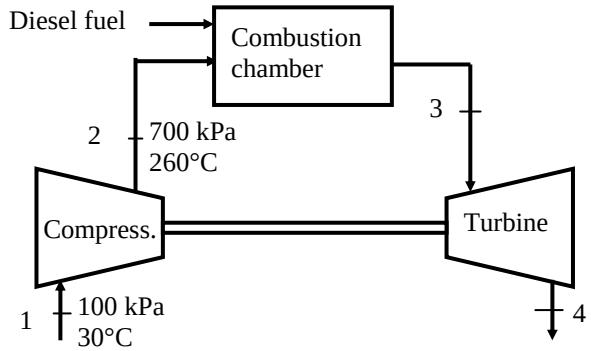
$$x_{\text{dest, 2-3}} = x_{\text{dest, 4-5}} = T_0 \left(c_p \ln \frac{T_3}{T_2} - R \ln \frac{P_3}{P_2} + \frac{q_{\text{out,2-3}}}{T_{\text{sink}}} \right) = (290) \left[(1.005) \ln \frac{290}{430.9} - 0 + \frac{141.6}{290} \right] = \mathbf{26.2\ kJ/kg}$$

$$x_{\text{dest,regen}} = T_0 (\Delta s_{6-7} + \Delta s_{13-14}) = T_0 \left(c_p \ln \frac{T_7}{T_6} + c_p \ln \frac{T_{14}}{T_{13}} \right)$$

$$= (290) \left[(1.005) \ln \frac{520.7}{430.9} + (1.005) \ln \frac{495.9}{585.7} \right] = \mathbf{6.66\ kJ/kg}$$

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9-157 A gas-turbine plant uses diesel fuel and operates on simple Brayton cycle. The isentropic efficiency of the compressor, the net power output, the back work ratio, the thermal efficiency, and the second-law efficiency are to be determined.



Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at 500ºC = 773 K are c_p = 1.093 kJ/kg·K, c_v = 0.806 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.357 (Table A-2b).

Analysis (*a*) The isentropic efficiency of the compressor may be determined if we first calculate the exit temperature for the isentropic case

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (303\text{ K})\left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)^{(1.357-1)/1.357} = 505.6\text{ K}$$

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{(505.6 - 303)\text{K}}{(533 - 303)\text{K}} = \mathbf{0.881}$$

(*b*) The total mass flowing through the turbine and the rate of heat input are

$$\dot{m}_t = \dot{m}_a + \dot{m}_f = \dot{m}_a + \frac{\dot{m}_a}{\text{AF}} = 12.6\text{ kg/s} + \frac{12.6\text{ kg/s}}{60} = 12.6\text{ kg/s} + 0.21\text{ kg/s} = 12.81\text{ kg/s}$$

$$\dot{Q}_{\text{in}} = \dot{m}_f q_{\text{HV}} \eta_c = (0.21\text{ kg/s})(42{,}000\text{ kJ/kg})(0.97) = 8555\text{ kW}$$

The temperature at the exit of combustion chamber is

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_3 - T_2) \longrightarrow 8555\text{ kJ/s} = (12.81\text{ kg/s})(1.093\text{ kJ/kg.K})(T_3 - 533)\text{K} \longrightarrow T_3 = 1144\text{ K}$$

The temperature at the turbine exit is determined using isentropic efficiency relation

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1144\text{ K})\left(\frac{100\text{ kPa}}{700\text{ kPa}}\right)^{(1.357-1)/1.357} = 685.7\text{ K}$$

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow 0.85 = \frac{(1144 - T_4)\text{K}}{(1144 - 685.7)\text{K}} \longrightarrow T_4 = 754.4\text{ K}$$

The net power and the back work ratio are

$$\dot{W}_{\text{C,in}} = \dot{m}_a c_p(T_2 - T_1) = (12.6\text{ kg/s})(1.093\text{ kJ/kg.K})(533 - 303)\text{K} = 3168\text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}c_p(T_3 - T_4) = (12.81\text{ kg/s})(1.093\text{ kJ/kg.K})(1144 - 754.4)\text{K} = 5455\text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{C,in}} = 5455 - 3168 = \mathbf{2287\text{ kW}}$$

$$r_{\text{bw}} = \frac{\dot{W}_{\text{C,in}}}{\dot{W}_{\text{T,out}}} = \frac{3168\text{ kW}}{5455\text{ kW}} = \mathbf{0.581}$$

(*c*) The thermal efficiency is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{2287\text{ kW}}{8555\text{ kW}} = \mathbf{0.267}$$

The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{\max} = 1 - \frac{T_1}{T_3} = 1 - \frac{303\text{ K}}{1144\text{ K}} = 0.735$$

and

$$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\max}} = \frac{0.267}{0.735} = \mathbf{0.364}$$

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9-158 A modern compression ignition engine operates on the ideal dual cycle. The maximum temperature in the cycle, the net work output, the thermal efficiency, the mean effective pressure, the net power output, the second-law efficiency of the cycle, and the rate of exergy of the exhaust gases are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at 1000 K are c_p = 1.142 kJ/kg·K, c_v = 0.855 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.336 (Table A-2b).

Analysis (*a*) The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

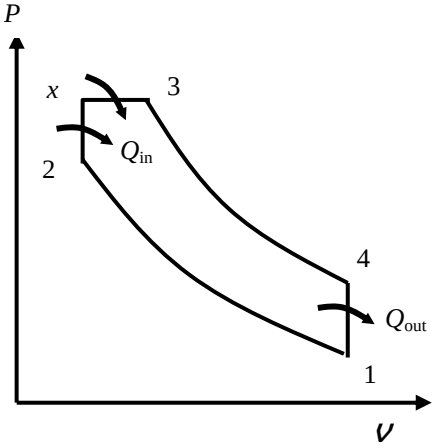
$$r = \frac{V_c + V_d}{V_c} \longrightarrow 16 = \frac{V_c + 0.0018\,\text{m}^3}{V_c} \longrightarrow V_c = 0.00012\,\text{m}^3 = V_2 = V_x$$

$$V_1 = V_c + V_d = 0.00012 + 0.0018 = 0.00192\,\text{m}^3 = V_4$$

Process 1-2: Isentropic compression

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (343\text{ K})(16)^{1.336\text{-}1} = 870.7\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = (95\text{ kPa})(16)^{1.336} = 3859\text{ kPa}$$



Process 2-x and x-3: Constant-volume and constant pressure heat addition processes:

$$T_x = T_2\frac{P_x}{P_2} = (870.7\text{ K})\frac{7500\text{ kPa}}{3859\text{ kPa}} = 1692\text{ K}$$

$$q_{2\text{-}x} = c_v(T_x - T_2) = (0.855\text{ kJ/kg.K})(1692 - 870.7)\text{K} = 702.6\text{ kJ/kg}$$

$$q_{2-x} = q_{x\text{-}3} = c_p(T_3 - T_x) \longrightarrow 702.6\text{ kJ/kg} = (0.855\text{ kJ/kg.K})(T_3 - 1692)\text{K} \longrightarrow T_3 = \mathbf{2308\text{ K}}$$

(*b*) $$q_{\text{in}} = q_{2-x} + q_{x\text{-}3} = 702.6 + 702.6 = 1405\text{ kJ/kg}$$

$$V_3 = V_x\frac{T_3}{T_x} = (0.00012\,\text{m}^3)\frac{2308\text{ K}}{1692\text{ K}} = 0.0001636\,\text{m}^3$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = (2308\text{ K})\left(\frac{0.0001636\,\text{m}^3}{0.00192\,\text{m}^3}\right)^{1.336\text{-}1} = 1009\text{ K}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = (7500\text{ kPa})\left(\frac{0.0001636\,\text{m}^3}{0.00192\,\text{m}^3}\right)^{1.336} = 279.4\text{ kPa}$$

Process 4-1: constant voume heat rejection.

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.855\text{ kJ/kg}\cdot\text{K})(1009 - 343)\text{K} = 569.3\text{ kJ/kg}$$

The net work output and the thermal efficiency are

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 1405 - 569.3 = \mathbf{835.8\text{ kJ/kg}}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{835.8\text{ kJ/kg}}{1405\text{ kJ/kg}} = 0.5948 = \mathbf{59.5\%}$$

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(*c*) The mean effective pressure is determined to be

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\ \text{kPa})(0.00192\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343\ \text{K})} = 0.001853\ \text{kg}$$

$$\text{MEP} = \frac{m w_{\text{net,out}}}{V_1 - V_2} = \frac{(0.001853\ \text{kg})(835.8\text{kJ/kg})}{(0.00192 - 0.00012)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{860.4\ kPa}$$

(*d*) The power for engine speed of 3500 rpm is

$$\dot{W}_{\text{net}} = m w_{\text{net}} \frac{\dot{n}}{2} = (0.001853\ \text{kg})(835.8\ \text{kJ/kg})\frac{2200\ (\text{rev/min})}{(2\ \text{rev/cycle})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = \mathbf{28.39\ kW}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). We take the dead state temperature and pressure to be 25ºC and 100 kPa.

$$\eta_{\max} = 1 - \frac{T_0}{T_3} = 1 - \frac{(25 + 273)\ \text{K}}{2308\ \text{K}} = 0.8709$$

and

$$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\max}} = \frac{0.5948}{0.8709} = 0.683 = \mathbf{68.3\%}$$

The rate of exergy of the exhaust gases is determined as follows

$$x_4 = u_4 - u_0 - T_0(s_4 - s_0) = c_v(T_4 - T_0) - T_0\left[c_p \ln\frac{T_4}{T_0} - R\ln\frac{P_4}{P_0}\right]$$

$$= (0.855)(1009 - 298) - (298)\left[(1.142\ \text{kJ/kg.K})\ln\frac{1009}{298} - (0.287\ \text{kJ/kg.K})\ln\frac{279.4}{100}\right] = 285.0\ \text{kJ/kg}$$

$$\dot{X}_4 = m x_4 \frac{\dot{n}}{2} = (0.001853\ \text{kg})(285.0\ \text{kJ/kg})\frac{2200\ (\text{rev/min})}{(2\ \text{rev/cycle})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = \mathbf{9.683\ kW}$$

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Review Problems

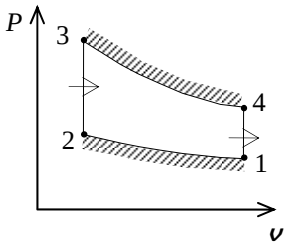
9-159 An Otto cycle with a compression ratio of 7 is considered. The thermal efficiency is to be determined using constant and variable specific heats.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are $R = 0.287$ kPa·m^3/kg·K, $c_p = 1.005$ kJ/kg·K, $c_v = 0.718$ kJ/kg·K, and $k = 1.4$ (Table A-2a).

Analysis (*a*) Constant specific heats:

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{7^{1.4-1}} = 0.5408 = \mathbf{54.1\%}$$



(b) Variable specific heats: (using air properties from Table A-17)

Process 1-2: isentropic compression.

$$T_1 = 288\,\text{K} \longrightarrow \begin{matrix} u_1 = 205.48\,\text{kJ/kg} \\ v_{r1} = 688.1 \end{matrix}$$

$$v_{r2} = \frac{v_2}{v_1} v_{r2} = \frac{1}{r} v_{r2} = \frac{1}{7}(688.1) = 98.3 \longrightarrow u_2 = 447.62\,\text{kJ/kg}$$

Process 2-3: v = constant heat addition.

$$T_3 = 1273\,\text{K} \longrightarrow \begin{matrix} u_3 = 998.51\,\text{kJ/kg} \\ v_{r3} = 12.045 \end{matrix}$$

$$q_{in} = u_3 - u_2 = 998.51 - 447.62 = 550.89\,\text{kJ/kg}$$

Process 3-4: isentropic expansion.

$$v_{r4} = \frac{v_4}{v_3} v_{r3} = r v_{r3} = (7)(12.045) = 84.32 \longrightarrow u_4 = 475.54\,\text{kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 475.54 - 205.48 = 270.06\,\text{kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{270.06\,\text{kJ/kg}}{550.89\,\text{kJ/kg}} = 0.5098 = \mathbf{51.0\%}$$

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9-160E An ideal diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency is to be determined using constant and variable specific heats.

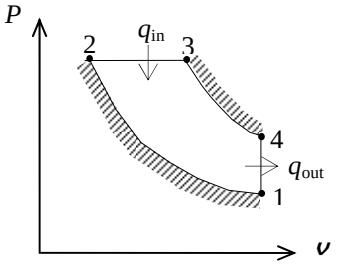
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, R = 0.06855 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis (*a*) Constant specific heats:

Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (505\text{ R})(20)^{0.4} = 1673.8\text{ R}$$



Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2260\text{ R}}{1673.8\text{ R}} = 1.350$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1.350 v_2}{v_4}\right)^{k-1} = T_3\left(\frac{1.350}{r}\right)^{k-1} = (2260\text{ R})\left(\frac{1.350}{20}\right)^{0.4} = 768.8\text{ R}$$

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) = (0.240\text{ Btu/lbm}\cdot\text{R})(2260 - 1673.8)\text{R} = 140.7\text{ Btu/lbm}$$

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.171\text{ Btu/lbm}\cdot\text{R})(768.8 - 505)\text{R} = 45.11\text{ Btu/lbm}$$

$$w_{net,out} = q_{in} - q_{out} = 140.7 - 45.11 = 95.59\text{ Btu/lbm}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{95.59\text{ Btu/lbm}}{140.7\text{ Btu/lbm}} = 0.6794 = \mathbf{67.9\%}$$

(*b*) Variable specific heats: (using air properties from Table A-17)

Process 1-2: isentropic compression.

$$T_1 = 505\text{ R} \longrightarrow \begin{matrix} u_1 = 86.06\text{ Btu/lbm} \\ v_{r1} = 170.82 \end{matrix}$$

$$v_{r2} = \frac{v_2}{v_1} v_{r1} = \frac{1}{r} v_{r1} = \frac{1}{20}(170.82) = 8.541 \longrightarrow \begin{matrix} T_2 = 1582.3\text{ R} \\ h_2 = 391.01\text{ Btu/lbm} \end{matrix}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2260\text{ R}}{1582.3\text{ R}} = 1.428$$

$$T_3 = 2260\text{ R} \longrightarrow \begin{matrix} h_3 = 577.52\text{ Btu/lbm} \\ v_{r3} = 2.922 \end{matrix}$$

$$q_{in} = h_3 - h_2 = 577.52 - 391.01 = 186.51\text{ Btu/lbm}$$

Process 3-4: isentropic expansion.

$$v_{r4} = \frac{v_4}{v_3} v_{r3} = \frac{v_4}{1.428 v_2} v_{r3} = \frac{r}{1.428} v_{r3} = \frac{20}{1.428}(2.922) = 40.92 \longrightarrow u_4 = 152.65\text{ Btu/lbm}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = 152.65 - 86.06 = 66.59\text{ Btu/lbm}$$

Then $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{66.59\text{ Btu/lbm}}{186.51\text{ Btu/lbm}} = 0.6430 = \mathbf{64.3\%}$$

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9-161E A simple ideal Brayton cycle with air as the working fluid operates between the specified temperature limits. The net work is to be determined using constant and variable specific heats.

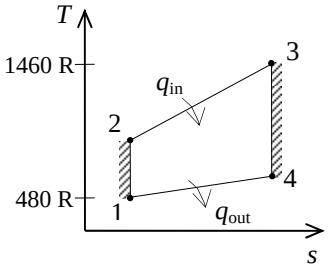
Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis (*a*) Constant specific heats:

$$T_2 = T_1 r_p^{(k-1)/k} = (480\ \text{R})(12)^{0.4/1.4} = 976.3\ \text{R}$$

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1460\ \text{R})\left(\frac{1}{12}\right)^{0.4/1.4} = 717.8\ \text{R}$$



$$\begin{aligned} w_{net} &= w_{turb} - w_{comp} \\ &= c_p(T_3 - T_4) - c_p(T_2 - T_1) \\ &= c_p(T_3 - T_4 + T_1 - T_2) \\ &= (0.240\ \text{Btu/lbm}\cdot\text{R})(1460 - 717.8 + 480 - 976.3)\text{R} \\ &= \mathbf{59.0\ Btu/lbm} \end{aligned}$$

(*b*) Variable specific heats: (using air properties from Table A-17E)

$$T_1 = 480\ \text{R} \longrightarrow \begin{matrix} h_1 = 114.69\ \text{Btu/lbm} \\ P_{r1} = 0.9182 \end{matrix}$$

$$P_{r2} = \frac{P_2}{P_1} P_{r1} = (12)(0.9182) = 11.02 \longrightarrow h_2 = 233.63\ \text{Btu/lbm}$$

$$T_3 = 1460\ \text{R} \longrightarrow \begin{matrix} h_3 = 358.63\ \text{Btu/lbm} \\ P_{r3} = 50.40 \end{matrix}$$

$$P_{r4} = \frac{P_4}{P_3} P_{r3} = \left(\frac{1}{12}\right)(50.40) = 4.12 \longrightarrow h_4 = 176.32\ \text{Btu/lbm}$$

$$\begin{aligned} w_{net} &= w_{turb} - w_{comp} \\ &= (h_3 - h_4) - (h_2 - h_1) \\ &= (358.63 - 176.32) - (233.63 - 114.69) \\ &= \mathbf{63.4\ Btu/lbm} \end{aligned}$$

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9-162 A turbocharged four-stroke V-16 diesel engine produces 3500 hp at 1200 rpm. The amount of power produced per cylinder per mechanical and per thermodynamic cycle is to be determined.

Analysis Noting that there are 16 cylinders and each thermodynamic cycle corresponds to 2 mechanical cycles (revolutions), we have

(*a*)

$$w_{\text{mechanical}} = \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of mechanical cycles})}$$

$$= \frac{3500\ \text{hp}}{(16\ \text{cylinders})(1200\ \text{rev/min})}\left(\frac{42.41\ \text{Btu/min}}{1\ \text{hp}}\right)$$

$$= \mathbf{7.73\ Btu/cyl \cdot mech\ cycle} \quad (= 8.16\ \text{kJ/cyl} \cdot \text{mech cycle})$$

(*b*)

$$w_{\text{thermodynamic}} = \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of thermodynamic cycles})}$$

$$= \frac{3500\ \text{hp}}{(16\ \text{cylinders})(1200/2\ \text{rev/min})}\left(\frac{42.41\ \text{Btu/min}}{1\ \text{hp}}\right)$$

$$= \mathbf{15.46\ Btu/cyl \cdot therm\ cycle} \quad (= 16.31\ \text{kJ/cyl} \cdot \text{therm cycle})$$

9-163 A simple ideal Brayton cycle operating between the specified temperature limits is considered. The pressure ratio for which the compressor and the turbine exit temperature of air are equal is to be determined.

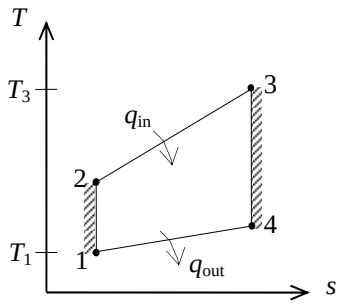
Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is k =1.4 (Table A-2).

Analysis We treat air as an ideal gas with constant specific heats. Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1(r_p)^{(k-1)/k}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k}$$



Setting $T_2 = T_4$ and solving for r_p gives

$$r_p = \left(\frac{T_3}{T_1}\right)^{k/2(k-1)} = \left(\frac{1500\ \text{K}}{300\ \text{K}}\right)^{1.4/0.8} = \mathbf{16.7}$$

Therefore, the compressor and turbine exit temperatures will be equal when the compression ratio is 16.7.

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9-164 A four-cylinder spark-ignition engine with a compression ratio of 8 is considered. The amount of heat supplied per cylinder, the thermal efficiency, and the rpm for a net power output of 60 kW are to be determined.

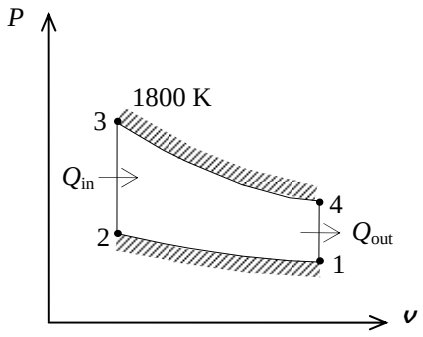
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-1). The properties of air are given in Table A-17.

Analysis (*a*) Process 1-2: isentropic compression.

$$T_1 = 310 \text{ K} \longrightarrow u_1 = 221.25 \text{ kJ/kg}$$
$$v_{r_1} = 572.3$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{10.5}(572.3) = 54.50$$
$$\longrightarrow u_2 = 564.29 \text{ kJ/kg}$$



Process 2-3: v = constant heat addition.

$$T_3 = 2100 \text{ K} \longrightarrow u_3 = 1775.3 \text{ kJ/kg}$$
$$v_{r_3} = 2.356$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{(98 \text{ kPa})(0.0004 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(310 \text{ K})} = 4.406 \times 10^{-4} \text{ kg}$$

$$Q_{\text{in}} = m(u_3 - u_2) = (4.406 \times 10^{-4} \text{ kg})(1775.3 - 564.29) \text{kJ/kg} = \mathbf{0.5336 \text{ kJ}}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (10.5)(2.356) = 24.74 \longrightarrow u_4 = 764.05 \text{ kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$Q_{\text{out}} = m(u_4 - u_1) = (4.406 \times 10^{-4} \text{ kg})(764.05 - 221.25) \text{kJ/kg} = 0.2392 \text{ kJ}$$

$$W_{\text{net}} = Q_{\text{in}} - Q_{\text{out}} = 0.5336 - 0.2392 = 0.2944 \text{ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{0.2944 \text{ kJ}}{0.5336 \text{ kJ}} = 0.5517 = \mathbf{55.2\%}$$

(*c*) $$\dot{n} = 2 \frac{\dot{W}_{\text{net}}}{n_{\text{cyl}} W_{\text{net,cyl}}} = (2 \text{ rev/cycle}) \frac{45 \text{ kJ/s}}{4 \times (0.2944 \text{ kJ/cycle})} \left(\frac{60 \text{ s}}{1 \text{ min}} \right) = \mathbf{4586 \text{ rpm}}$$

Note that for four-stroke cycles, there are two revolutions per cycle.

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9-165 Problem 9-164 is reconsidered. The effect of the compression ratio net work done and the efficiency of the cycle is to be investigated. Also, the *T-s* and *P-v* diagrams for the cycle are to be plotted.

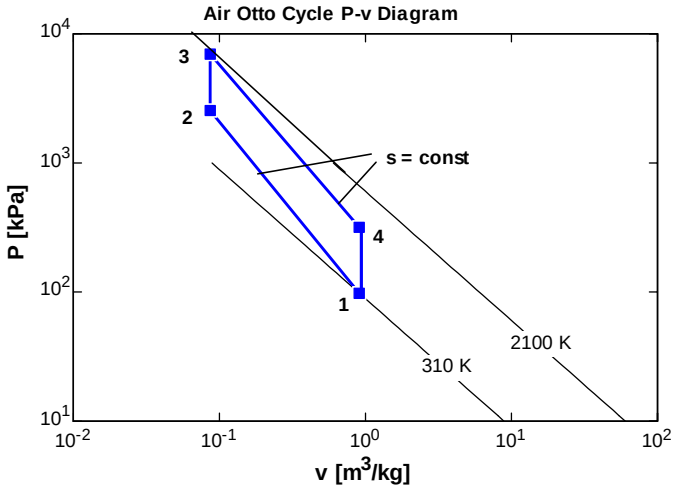
Analysis Using EES, the problem is solved as follows:

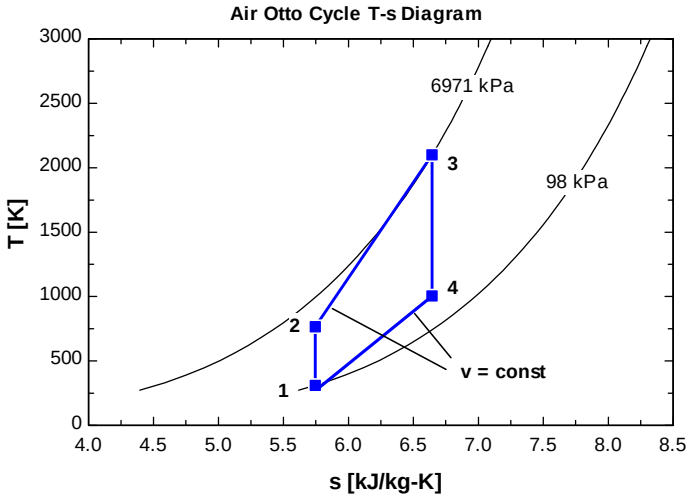
```
"Input Data"
T[1]=(37+273) [K]
P[1]=98 [kPa]
T[3]= 2100 [K]
V_cyl=0.4 [L]*Convert(L, m^3)
r_v=10.5 "Compression ratio"
W_dot_net = 45 [kW]
N_cyl=4 "number of cyclinders"
v[1]/v[2]=r_v

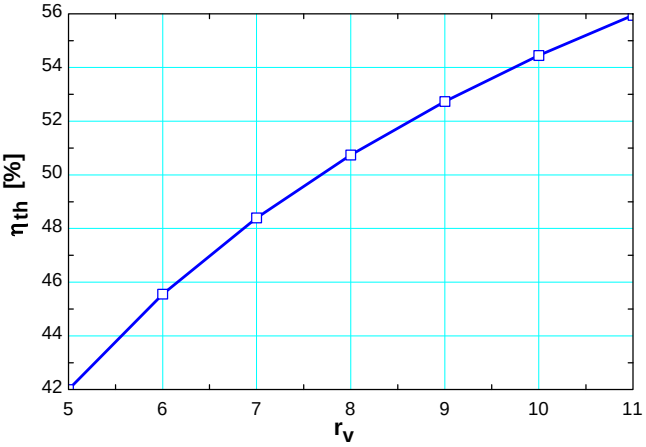
"The first part of the solution is done per unit mass."
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
s[2]=entropy(air, T=T[2], v=v[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
"Conservation of energy for process 1 to 2: no heat transfer (s=const.) with work input"
 w_in = DELTAu_12
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=R*T[3]
v[3]=v[2]
"Conservation of energy for process 2 to 3: the work is zero for v=const, heat is added"
q_in  = DELTAu_23
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=entropy(air,T=T[4],P=P[4])
s[4]=s[3]
P[4]*v[4]/T[4]=P[3]*v[3]/T[3]
{P[4]*v[4]=R*T[4]}
"Conservation of energy for process 3 to 4: no heat transfer (s=const) with work output"
 - w_out = DELTAu_34
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
v[4]=v[1]
"Conservation of energy for process 2 to 3: the work is zero for v=const; heat is rejected"
- q_out = DELTAu_41
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
w_net = w_out - w_in
Eta_th=w_net/q_in*Convert(, %) "Thermal efficiency, in percent"
"The mass contained in each cylinder is found from the volume of the cylinder:"
V_cyl=m*v[1]
"The net work done per cycle is:"
W_dot_net=m*w_net"kJ/cyl"*N_cyl*N_dot"mechanical cycles/min"*1"min"/60"s"*1"thermal cycle"/2"mechanical cycles"
```

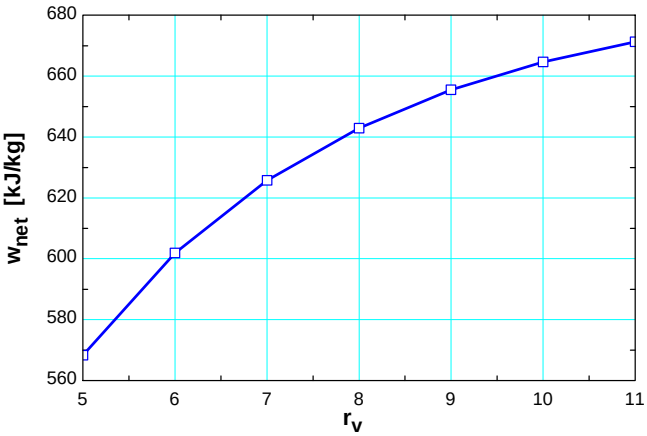
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r_v	η_{th} [%]	w_{net} [kJ/kg]
5	42	568.3
6	45.55	601.9
7	48.39	625.7
8	50.74	642.9
9	52.73	655.5
10	54.44	664.6
11	55.94	671.2









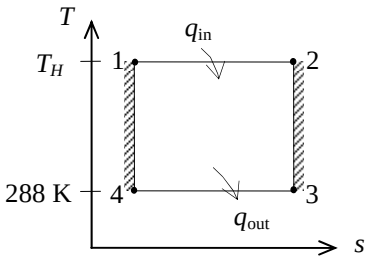
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9-166 An ideal gas Carnot cycle with helium as the working fluid is considered. The pressure ratio, compression ratio, and minimum temperature of the energy source are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is $k = 1.667$ (Table A-2a).

Analysis From the definition of the thermal efficiency of a Carnot heat engine,



$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} \longrightarrow T_H = \frac{T_L}{1-\eta_{\text{th,Carnot}}} = \frac{(15+273)\,\text{K}}{1-0.50} = \mathbf{576\ K}$$

An isentropic process for an ideal gas is one in which $P\nu^k$ remains constant. Then, the pressure ratio is

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{k/(k-1)} = \left(\frac{576\,\text{K}}{288\,\text{K}}\right)^{1.667/(1.667-1)} = \mathbf{5.65}$$

Based on the process equation, the compression ratio is

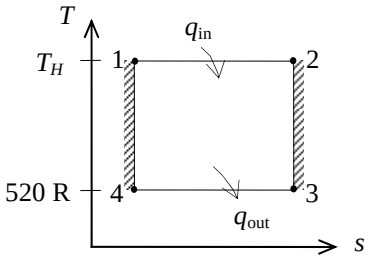
$$\frac{\nu_1}{\nu_2} = \left(\frac{P_2}{P_1}\right)^{1/k} = (5.65)^{1/1.667} = \mathbf{2.83}$$

9-167E An ideal gas Carnot cycle with helium as the working fluid is considered. The pressure ratio, compression ratio, and minimum temperature of the energy-source reservoir are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is $k = 1.667$ (Table A-2Ea).

Analysis From the definition of the thermal efficiency of a Carnot heat engine,



$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} \longrightarrow T_H = \frac{T_L}{1-\eta_{\text{th,Carnot}}} = \frac{(60+460)\,\text{R}}{1-0.60} = \mathbf{1300\ R}$$

An isentropic process for an ideal gas is one in which $P\nu^k$ remains constant. Then, the pressure ratio is

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{k/(k-1)} = \left(\frac{1300\,\text{R}}{520\,\text{R}}\right)^{1.667/(1.667-1)} = \mathbf{9.88}$$

Based on the process equation, the compression ratio is

$$\frac{\nu_1}{\nu_2} = \left(\frac{P_2}{P_1}\right)^{1/k} = (9.88)^{1/1.667} = \mathbf{3.95}$$

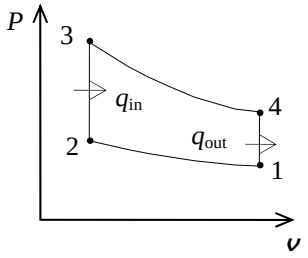
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9-168 The compression ratio required for an ideal Otto cycle to produce certain amount of work when consuming a given amount of fuel is to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats. **4** The combustion efficiency is 100 percent.

Properties The properties of air at room temperature are $k = 1.4$ (Table A-2).

Analysis The heat input to the cycle for 0.043 grams of fuel consumption is



$$Q_{in} = m_{fuel} q_{HV} = (0.035 \times 10^{-3}\ \text{kg})(43{,}000\ \text{kJ/kg}) = 1.505\ \text{kJ}$$

The thermal efficiency is then

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{1\ \text{kJ}}{1.505\ \text{kJ}} = 0.6645$$

From the definition of thermal efficiency, we obtain the required compression ratio to be

$$\eta_{th} = 1 - \frac{1}{r^{k-1}} \longrightarrow r = \frac{1}{(1-\eta_{th})^{1/(k-1)}} = \frac{1}{(1-0.6645)^{1/(1.4-1)}} = \mathbf{15.3}$$

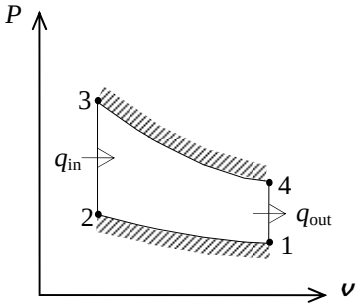
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9-169 An ideal Otto cycle with air as the working fluid with a compression ratio of 9.2 is considered. The amount of heat transferred to the air, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-1). The properties of air are given in Table A-17.

Analysis (*a*) Process 1-2: isentropic compression.



$$T_1 = 300 \text{ K} \longrightarrow u_1 = 214.07 \text{ kJ/kg}$$
$$v_{r_1} = 621.2$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{9.2}(621.2) = 67.52 \longrightarrow T_2 = 708.3 \text{ K}$$
$$u_2 = 518.9 \text{ kJ/kg}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1} P_1 = (9.2)\left(\frac{708.3 \text{ K}}{300 \text{ K}}\right)(98 \text{ kPa}) = 2129 \text{ kPa}$$

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2} T_2 = 2T_2 = (2)(708.3) = 1416.6 \text{ K} \longrightarrow u_3 = 1128.7 \text{ kJ/kg}$$
$$v_{r_3} = 8.593$$

$$q_{in} = u_3 - u_2 = 1128.7 - 518.9 = \mathbf{609.8 \text{ kJ/kg}}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (9.2)(8.593) = 79.06 \longrightarrow u_4 = 487.75 \text{ kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 487.75 - 214.07 = 273.7 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 609.8 - 273.7 = \mathbf{336.1 \text{ kJ/kg}}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{336.1 \text{ kJ/kg}}{609.8 \text{ kJ/kg}} = \mathbf{55.1\%}$$

(*d*) $$v_{\max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})}{98 \text{ kPa}} = 0.879 \text{ m}^3/\text{kg}$$

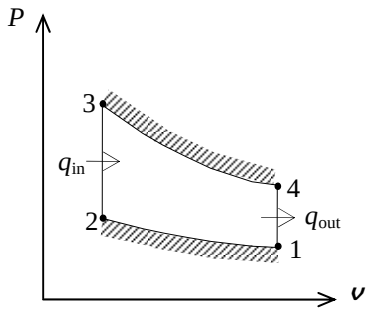
$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{w_{\text{net}}}{v_1(1 - 1/r)} = \frac{336.1 \text{ kJ/kg}}{(0.879 \text{ m}^3/\text{kg})(1 - 1/9.2)}\left(\frac{1 \text{ kPa} \cdot \text{m}^3}{1 \text{ kJ}}\right) = \mathbf{429 \text{ kPa}}$$

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9-170 An ideal Otto cycle with air as the working fluid with a compression ratio of 9.2 is considered. The amount of heat transferred to the air, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.



Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis (a) Process 1-2 is isentropic compression:

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{ K})(9.2)^{0.4} = 728.8\text{ K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.2)\left(\frac{728.8\text{ K}}{300\text{ K}}\right)(98\text{ kPa}) = 2190\text{ kPa}$$

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2}T_2 = 2T_2 = (2)(728.8) = 1457.6\text{ K}$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2) = (0.718\text{ kJ/kg}\cdot\text{K})(1457.6 - 728.8)\text{K} = \mathbf{523.3\ kJ/kg}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1457.6\text{ K})\left(\frac{1}{9.2}\right)^{0.4} = 600.0\text{ K}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(600 - 300)\text{K} = 215.4\text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 523.3 - 215.4 = \mathbf{307.9\ kJ/kg}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{307.9\text{ kJ/kg}}{523.3\text{ kJ/kg}} = \mathbf{58.8\%}$$

(*d*) $$v_{\max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{98\text{ kPa}} = 0.879\text{ m}^3/\text{kg}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{w_{\text{net}}}{v_1(1 - 1/r)} = \frac{307.9\text{ kJ/kg}}{(0.879\text{ m}^3/\text{kg})(1 - 1/9.2)}\left(\frac{1\text{ kPa}\cdot\text{m}^3}{1\text{ kJ}}\right) = \mathbf{393\ kPa}$$

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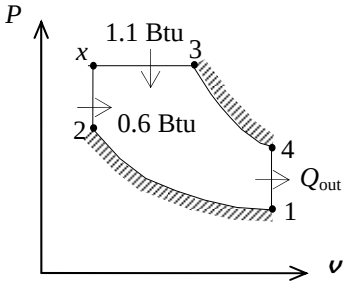
9-171E An ideal dual cycle with air as the working fluid with a compression ratio of 12 is considered. The thermal efficiency of the cycle is to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm.R, c_v = 0.171 Btu/lbm.R, and k = 1.4 (Table A-2E).

Analysis The mass of air is

$$m=\frac{P_1V_1}{RT_1}=\frac{(14.7\text{ psia})(98/1728\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(580\text{ R})}=0.003881\text{ lbm}$$



Process 1-2: isentropic compression.

$$T_2=T_1\left(\frac{V_1}{V_2}\right)^{k-1}=(580\text{ R})(14)^{0.4}=1667\text{ R}$$

Process 2-x: *v* = *constant* heat addition,

$$Q_{2-x,\text{in}}=m(u_x-u_2)=mc_v(T_x-T_2)$$
$$0.6\text{ Btu}=(0.003881\text{ lbm})(0.171\text{ Btu/lbm}\cdot\text{R})(T_x-1667)\text{R}\longrightarrow T_x=2571\text{ R}$$

Process x-3: P = *constant* heat addition.

$$Q_{x-3,\text{in}}=m(h_3-h_x)=mc_p(T_3-T_x)$$
$$1.1\text{ Btu}=(0.003881\text{ lbm})(0.240\text{ Btu/lbm}\cdot\text{R})(T_3-2571)\text{R}\longrightarrow T_3=3752\text{ R}$$

$$\frac{P_3V_3}{T_3}=\frac{P_xV_x}{T_x}\longrightarrow r_c=\frac{V_3}{V_x}=\frac{T_3}{T_x}=\frac{3752\text{ R}}{2571\text{ R}}=1.459$$

Process 3-4: isentropic expansion.

$$T_4=T_3\left(\frac{V_3}{V_4}\right)^{k-1}=T_3\left(\frac{1.459V_1}{V_4}\right)^{k-1}=T_3\left(\frac{1.459}{r}\right)^{k-1}=(3752\text{ R})\left(\frac{1.459}{14}\right)^{0.4}=1519\text{ R}$$

Process 4-1: v = constant heat rejection.

$$Q_{\text{out}}=m(u_4-u_1)=mc_v(T_4-T_1)$$
$$=(0.003881\text{ lbm})(0.171\text{ Btu/lbm}\cdot\text{R})(1519-580)\text{R}=0.6229\text{ Btu}$$

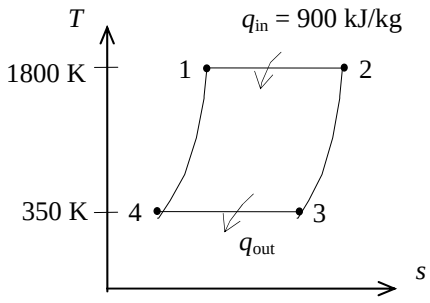
$$\eta_{\text{th}}=1-\frac{Q_{\text{out}}}{Q_{\text{in}}}=1-\frac{0.6229\text{ Btu}}{1.7\text{ Btu}}=0.6336=\mathbf{63.4\%}$$

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9-172 An ideal Stirling cycle with air as the working fluid is considered. The maximum pressure in the cycle and the net work output are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $R = 0.287$ kJ/kg.K, $c_p = 1.005$ kJ/kg.K, $c_v = 0.718$ kJ/kg·K, and $k = 1.4$ (Table A-2).



Analysis (*a*) The entropy change during process 1-2 is

$$s_2 - s_1 = \frac{q_{12}}{T_H} = \frac{900 \text{ kJ/kg}}{1800 \text{ K}} = 0.5 \text{ kJ/kg} \cdot \text{K}$$

and

$$s_2 - s_1 = c_v \ln \frac{T_2}{T_1}^{\nearrow 0} + R \ln \frac{v_2}{v_1} \longrightarrow 0.5 \text{ kJ/kg} \cdot \text{K} = (0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{v_2}{v_1} \longrightarrow \frac{v_2}{v_1} = 5.710$$

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow P_1 = P_3 \frac{v_3}{v_1} \frac{T_1}{T_3} = P_3 \frac{v_2}{v_1} \frac{T_1}{T_3} = (200 \text{ kPa})(5.710)\left(\frac{1800 \text{ K}}{350 \text{ K}}\right) = \mathbf{5873 \text{ kPa}}$$

(*b*) The net work output is

$$w_{net} = \eta_{th} q_{in} = \left(1 - \frac{T_L}{T_H}\right) q_{in} = \left(1 - \frac{350 \text{ K}}{1800 \text{ K}}\right)(900 \text{ kJ/kg}) = \mathbf{725 \text{ kJ/kg}}$$

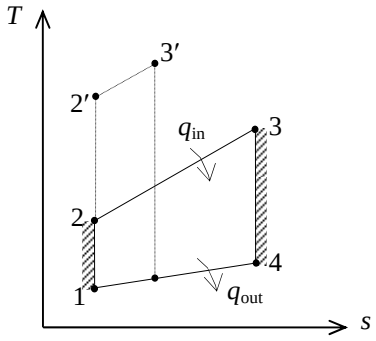
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9-173 A simple ideal Brayton cycle with air as the working fluid is considered. The changes in the net work output per unit mass and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-17.

Analysis The properties at various states are



$$T_1 = 300\ \text{K} \longrightarrow h_1 = 300.19\ \text{kJ/kg},\quad P_{r_1} = 1.386$$

$$T_3 = 1300\ \text{K} \longrightarrow h_3 = 1395.97\ \text{kJ/kg},\quad P_{r_3} = 330.9$$

For $r_p = 6$,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (6)(1.386) = 8.316 \longrightarrow h_2 = 501.40\ \text{kJ/kg}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{6}\right)(330.9) = 55.15 \longrightarrow h_4 = 855.3\ \text{kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = 1395.97 - 501.40 = 894.57\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 855.3 - 300.19 = 555.11\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 894.57 - 555.11 = 339.46\ \text{kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{339.46\ \text{kJ/kg}}{894.57\ \text{kJ/kg}} = 37.9\%$$

For $r_p = 12$,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (12)(1.386) = 16.63 \longrightarrow h_2 = 610.6\ \text{kJ/kg}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{12}\right)(330.9) = 27.58 \longrightarrow h_4 = 704.6\ \text{kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = 1395.97 - 610.60 = 785.37\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 704.6 - 300.19 = 404.41\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 785.37 - 404.41 = 380.96\ \text{kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{380.96\ \text{kJ/kg}}{785.37\ \text{kJ/kg}} = 48.5\%$$

Thus,

(*a*) $\Delta w_{\text{net}} = 380.96 - 339.46 = \mathbf{41.5\ kJ/kg}$ (increase)

(*b*) $\Delta \eta_{\text{th}} = 48.5\% - 37.9\% = \mathbf{10.6\%}$ (increase)

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9-174 A simple ideal Brayton cycle with air as the working fluid is considered. The changes in the net work output per unit mass and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.287 kJ/kg.K, c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis Processes 1-2 and 3-4 are isentropic. Therefore, For r_p = 6,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(6)^{0.4/1.4} = 500.6\text{ K}$$

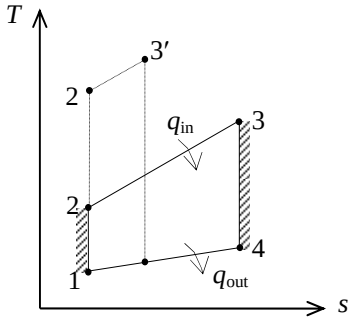
$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{ K})\left(\frac{1}{6}\right)^{0.4/1.4} = 779.1\text{ K}$$

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = c_p(T_3 - T_2) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(1300 - 500.6)\text{K} = 803.4\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{\text{out}} &= h_4 - h_1 = c_p(T_4 - T_1) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(779.1 - 300)\text{K} = 481.5\text{ kJ/kg} \end{aligned}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 803.4 - 481.5 = 321.9\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{321.9\text{ kJ/kg}}{803.4\text{ kJ/kg}} = 40.1\%$$



For r_p = 12,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(12)^{0.4/1.4} = 610.2\text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{ K})\left(\frac{1}{12}\right)^{0.4/1.4} = 639.2\text{ K}$$

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = c_p(T_3 - T_2) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(1300 - 610.2)\text{K} = 693.2\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{\text{out}} &= h_4 - h_1 = c_p(T_4 - T_1) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(639.2 - 300)\text{K} = 340.9\text{ kJ/kg} \end{aligned}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 693.2 - 340.9 = 352.3\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{352.3\text{ kJ/kg}}{693.2\text{ kJ/kg}} = 50.8\%$$

Thus,

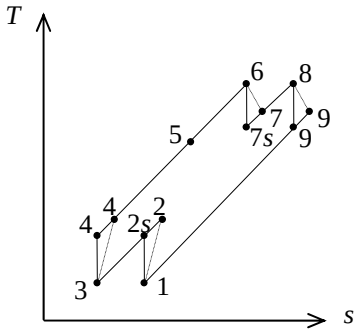
(*a*) $\Delta w_{\text{net}} = 352.3 - 321.9 = \mathbf{30.4\ kJ/kg}$ (increase)

(*b*) $\Delta\eta_{\text{th}} = 50.8\% - 40.1\% = \mathbf{10.7\%}$ (increase)

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9-175 A regenerative gas-turbine engine operating with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.



Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.

$$T_{4s} = T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\ \text{K})(4)^{0.4/1.4} = 445.8\ \text{K}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{c_p(T_{2s} - T_1)}{c_p(T_2 - T_1)} \longrightarrow T_4 = T_2 = T_1 + (T_{2s} - T_1)/\eta_C$$
$$= 300 + (445.8 - 300)/(0.78)$$
$$= 486.9\ \text{K}$$

$$T_{9s} = T_{7s} = T_6\left(\frac{P_7}{P_6}\right)^{(k-1)/k} = (1400\ \text{K})\left(\frac{1}{4}\right)^{0.4/1.4} = 942.1\ \text{K}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} = \frac{c_p(T_6 - T_7)}{c_p(T_6 - T_{7s})} \longrightarrow T_9 = T_7 = T_6 - \eta_T(T_6 - T_{7s})$$
$$= 1400 - (0.86)(1400 - 942.1)$$
$$= 1006\ \text{K}$$

$$\varepsilon = \frac{h_5 - h_4}{h_9 - h_4} = \frac{c_p(T_5 - T_4)}{c_p(T_9 - T_4)} \longrightarrow T_5 = T_4 + \varepsilon(T_9 - T_4)$$
$$= 486.9 + (0.75)(1006 - 486.9)$$
$$= 876.4\ \text{K}$$

$$w_{\text{C,in}} = 2(h_2 - h_1) = 2c_p(T_2 - T_1) = 2(1.005\ \text{kJ/kg}\cdot\text{K})(486.9 - 300)\text{K} = 375.7\ \text{kJ/kg}$$

$$w_{\text{T,out}} = 2(h_6 - h_7) = 2c_p(T_6 - T_7) = 2(1.005\ \text{kJ/kg}\cdot\text{K})(1400 - 1006)\text{K} = 791.5\ \text{kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{375.7\ \text{kJ/kg}}{791.5\ \text{kJ/kg}} = \mathbf{0.475}$$

$$q_{\text{in}} = (h_6 - h_5) + (h_8 - h_7) = c_p[(T_6 - T_5) + (T_8 - T_7)]$$
$$= (1.005\ \text{kJ/kg}\cdot\text{K})[(1400 - 876.4) + (1400 - 1006)]\text{K} = 922.0\ \text{kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 791.5 - 375.7 = 415.8\ \text{kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{415.8\ \text{kJ/kg}}{922.0\ \text{kJ/kg}} = 0.451 = \mathbf{45.1\%}$$

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9-176 Problem 9-175 is reconsidered. The effect of the isentropic efficiencies for the compressor and turbine and regenerator effectiveness on net work done and the heat supplied to the cycle is to be investigated. Also, the *T-s* diagram for the cycle is to be plotted.

Analysis Using EES, the problem is solved as follows:

```
"Input data"
T[6] = 1400 [K]
T[8] = T[6]
Pratio = 4
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = T[1]
Eta_reg = 0.75 "Regenerator effectiveness"
Eta_c =0.78 "Compressor isentorpic efficiency"
Eta_t =0.86 "Turbine isentropic efficiency"

"LP Compressor:"
"Isentropic Compressor anaysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen_LP/w_comp_LP
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the LP compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[1] + w_compisen_LP = h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])

"Actual compressor analysis:"
h[1] + w_comp_LP = h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"HP Compressor:"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the HP compressor"
P[4] = Pratio*P[3]
P[3] = P[2]
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])
"T_s[4] is the isentropic value of T[4] at compressor exit"
Eta_c = w_compisen_HP/w_comp_HP
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[3] + w_compisen_HP = h_s[4]
h[3]=ENTHALPY(Air,T=T[3])
h_s[4]=ENTHALPY(Air,T=T_s[4])

"Actual compressor analysis:"
h[3] + w_comp_HP = h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])
```

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```
"Intercooling heat loss:"
h[2] = q_out_intercool + h[3]

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
h[4] + q_in_noreg = h[6]

h[6]=ENTHALPY(Air,T=T[6])
P[6]=P[4]"process 4-6 is SSSF constant pressure"

"HP Turbine analysis"

s[6]=ENTROPY(Air,T=T[6],P=P[6])
s_s[7]=s[6] "For the ideal case the entropies are constant across the turbine"
P[7] = P[6] /Pratio
s_s[7]=ENTROPY(Air,T=T_s[7],P=P[7])"T_s[7] is the isentropic value of T[7] at HP turbine exit"
Eta_t = w_turb_HP /w_turbisen_HP "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
h[6] = w_turbisen_HP + h_s[7]
h_s[7]=ENTHALPY(Air,T=T_s[7])
"Actual Turbine analysis:"
h[6] = w_turb_HP + h[7]
h[7]=ENTHALPY(Air,T=T[7])
s[7]=ENTROPY(Air,T=T[7], P=P[7])

"Reheat Q_in:"
h[7] + q_in_reheat = h[8]
h[8]=ENTHALPY(Air,T=T[8])

"HL Turbine analysis"

P[8]=P[7]
s[8]=ENTROPY(Air,T=T[8],P=P[8])
s_s[9]=s[8] "For the ideal case the entropies are constant across the turbine"
P[9] = P[8] /Pratio
s_s[9]=ENTROPY(Air,T=T_s[9],P=P[9])"T_s[9] is the isentropic value of T[9] at LP turbine exit"
Eta_t = w_turb_LP /w_turbisen_LP "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
h[8] = w_turbisen_LP + h_s[9]
h_s[9]=ENTHALPY(Air,T=T_s[9])
"Actual Turbine analysis:"
h[8] = w_turb_LP + h[9]
h[9]=ENTHALPY(Air,T=T[9])
s[9]=ENTROPY(Air,T=T[9], P=P[9])

"Cycle analysis"
w_net=w_turb_HP+w_turb_LP - w_comp_HP - w_comp_LP
q_in_total_noreg=q_in_noreg+q_in_reheat
Eta_th_noreg=w_net/(q_in_total_noreg)*Convert(, %) "[%]"    "Cycle thermal efficiency"
Bwr=(w_comp_HP + w_comp_LP)/(w_turb_HP+w_turb_LP)"Back work ratio"

"With the regenerator, the heat added in the external heat exchanger is"
h[5] + q_in_withreg = h[6]
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[4]
```

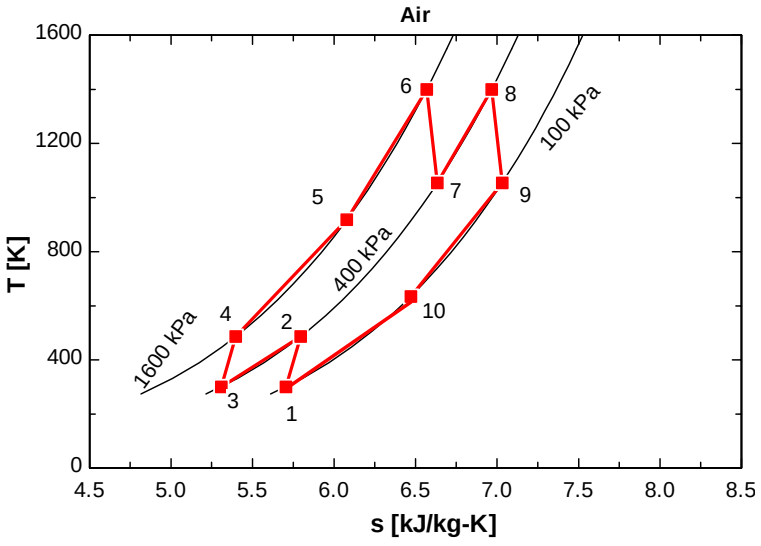
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"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[4])/(h[9]-h[4])
"Energy balance on regenerator gives h[10] and thus T[10] as:"
h[4] + h[9]=h[5] + h[10]
h[10]=ENTHALPY(Air, T=T[10])
s[10]=ENTROPY(Air,T=T[10], P=P[10])
P[10]=P[9]

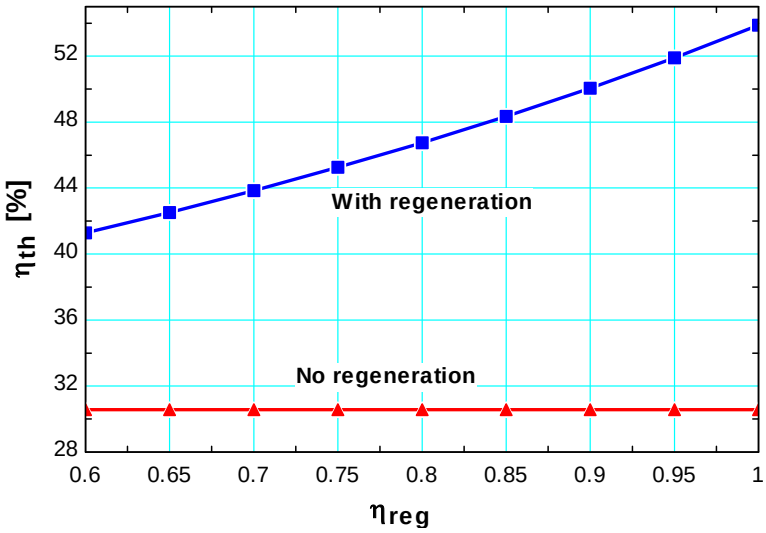
"Cycle thermal efficiency with regenerator"
q_in_total_withreg=q_in_withreg+q_in_reheat
Eta_th_withreg=w_net/(q_in_total_withreg)*Convert(, %) "[%]"

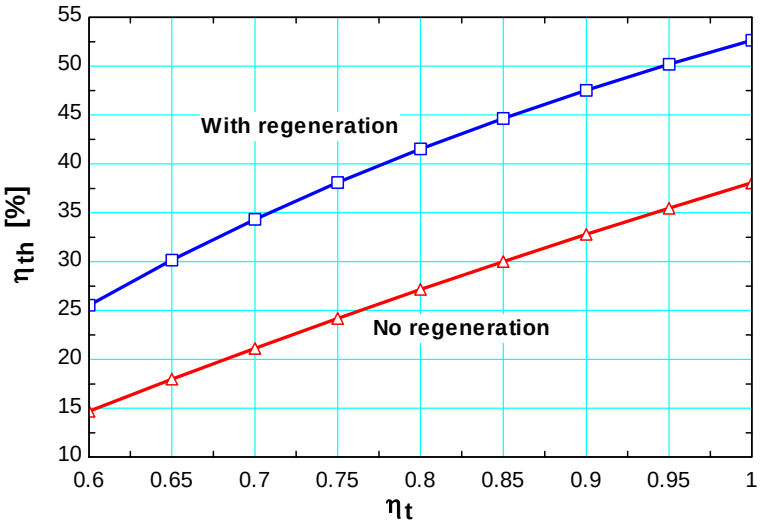
"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]
s_s[8]=s[8]
T_s[8]=T[8]
s_s[10]=s[10]
T_s[10]=T[10]

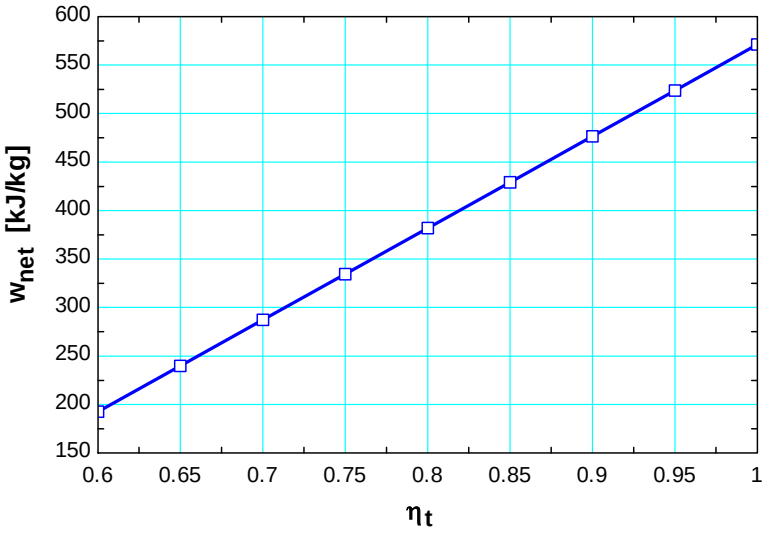
η_{reg}	η_C	η_t	$\eta_{th,noreg}$ [%]	$\eta_{th,withreg}$ [%]	$q_{in,total,noreg}$ [kJ/kg]	$q_{in,total,withreg}$ [kJ/kg]	w_{net} [kJ/kg]
0.6	0.78	0.86	30.57	41.29	1434	1062	438.5
0.65	0.78	0.86	30.57	42.53	1434	1031	438.5
0.7	0.78	0.86	30.57	43.85	1434	1000	438.5
0.75	0.78	0.86	30.57	45.25	1434	969.1	438.5
0.8	0.78	0.86	30.57	46.75	1434	938.1	438.5
0.85	0.78	0.86	30.57	48.34	1434	907.1	438.5
0.9	0.78	0.86	30.57	50.06	1434	876.1	438.5
0.95	0.78	0.86	30.57	51.89	1434	845.1	438.5
1	0.78	0.86	30.57	53.87	1434	814.1	438.5



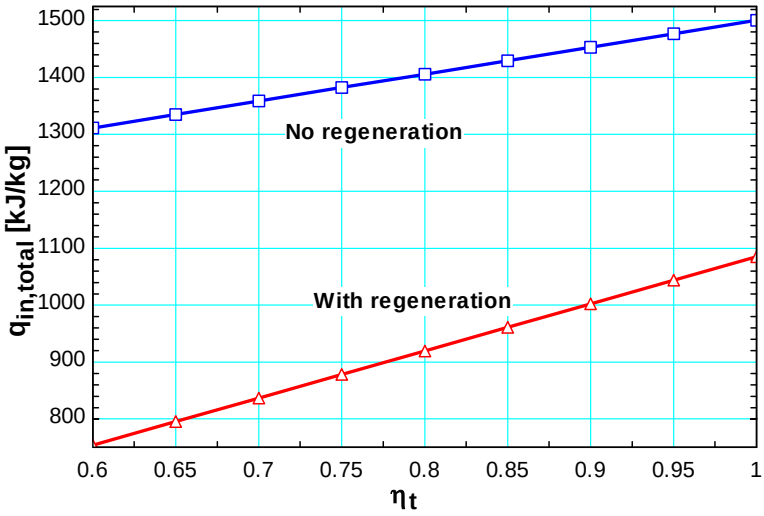
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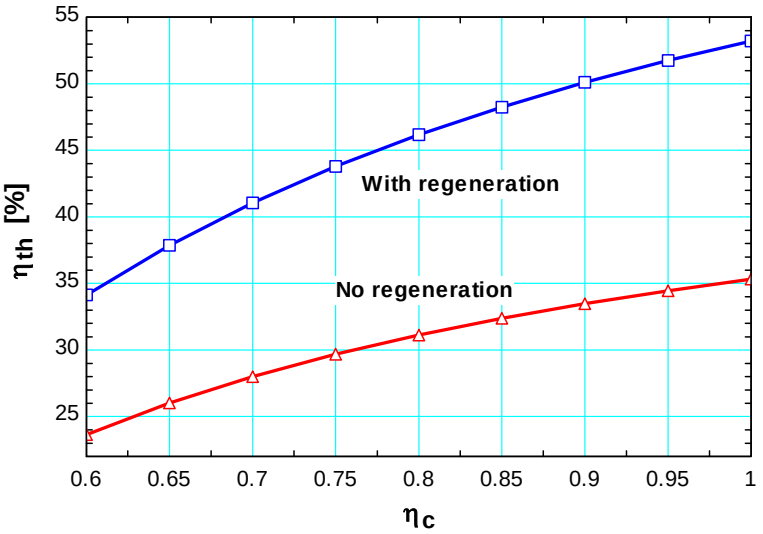






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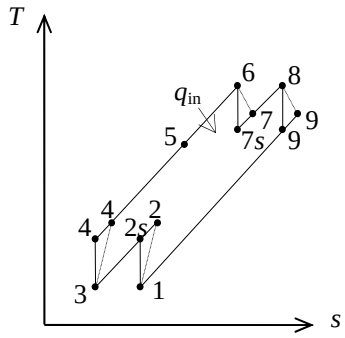




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9-177 A regenerative gas-turbine engine operating with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.



Properties The properties of helium at room temperature are c_p = 5.1926 kJ/kg.K and k = 1.667 (Table A-2).

Analysis The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.

$$T_{4s} = T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(4)^{0.667/1.667} = 522.4\text{ K}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{c_p(T_{2s} - T_1)}{c_p(T_2 - T_1)} \longrightarrow T_4 = T_2 = T_1 + (T_{2s} - T_1)/\eta_C$$
$$= 300 + (522.4 - 300)/(0.78)$$
$$= 585.2\text{ K}$$

$$T_{9s} = T_{7s} = T_6\left(\frac{P_7}{P_6}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{1}{4}\right)^{0.667/1.667} = 804.0\text{ K}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} = \frac{c_p(T_6 - T_7)}{c_p(T_6 - T_{7s})} \longrightarrow T_9 = T_7 = T_6 - \eta_T(T_6 - T_{7s})$$
$$= 1400 - (0.86)(1400 - 804.0)$$
$$= 887.4\text{ K}$$

$$\varepsilon = \frac{h_5 - h_4}{h_9 - h_4} = \frac{c_p(T_5 - T_4)}{c_p(T_9 - T_4)} \longrightarrow T_5 = T_4 + \varepsilon(T_9 - T_4)$$
$$= 585.2 + (0.75)(887.4 - 585.2)$$
$$= 811.8\text{ K}$$

$$w_{C,in} = 2(h_2 - h_1) = 2c_p(T_2 - T_1) = 2(5.1926\text{ kJ/kg}\cdot\text{K})(585.2 - 300)\text{K} = 2961\text{ kJ/kg}$$

$$w_{T,out} = 2(h_6 - h_7) = 2c_p(T_6 - T_7) = 2(5.1926\text{ kJ/kg}\cdot\text{K})(1400 - 887.4)\text{K} = 5323\text{ kJ/kg}$$

Thus,

$$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{2961\text{ kJ/kg}}{5323\text{ kJ/kg}} = \mathbf{0.556}$$

$$q_{in} = (h_6 - h_5) + (h_8 - h_7) = c_p[(T_6 - T_5) + (T_8 - T_7)]$$
$$= (5.1926\text{ kJ/kg}\cdot\text{K})[(1400 - 811.8) + (1400 - 887.4)]\text{K} = 5716\text{ kJ/kg}$$

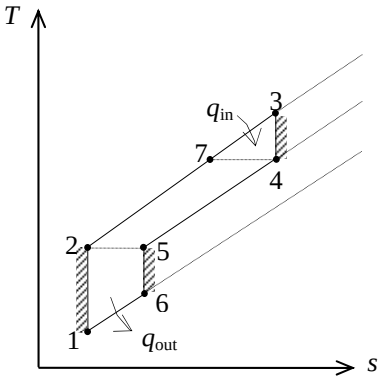
$$w_{net} = w_{T,out} - w_{C,in} = 5323 - 2961 = 2362\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{2362\text{ kJ/kg}}{5716\text{ kJ/kg}} = 0.4133 = \mathbf{41.3\%}$$

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9-178 An ideal gas-turbine cycle with one stage of compression and two stages of expansion and regeneration is considered. The thermal efficiency of the cycle as a function of the compressor pressure ratio and the high-pressure turbine to compressor inlet temperature ratio is to be determined, and to be compared with the efficiency of the standard regenerative cycle.

Analysis The *T*-*s* diagram of the cycle is as shown in the figure. If the overall pressure ratio of the cycle is r_p, which is the pressure ratio across the compressor, then the pressure ratio across each turbine stage in the ideal case becomes $\sqrt{r_p}$. Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as



$$T_5 = T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1\left(r_p\right)^{(k-1)/k}$$

$$T_7 = T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{\sqrt{r_p}}\right)^{(k-1)/k} = T_3 r_p^{(1-k)/2k}$$

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = T_5\left(\frac{1}{\sqrt{r_p}}\right)^{(k-1)/k} = T_2 r_p^{(1-k)/2k} = T_1 r_p^{(k-1)/k} r_p^{(1-k)/2k} = T_1 r_p^{(k-1)/2k}$$

Then,

$$q_{\text{in}} = h_3 - h_7 = c_p\left(T_3 - T_7\right) = c_p T_3\left(1 - r_p^{(1-k)/2k}\right)$$

$$q_{\text{out}} = h_6 - h_1 = c_p\left(T_6 - T_1\right) = c_p T_1\left(r_p^{(k-1)/2k} - 1\right)$$

and thus

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p T_1\left(r_p^{(k-1)/2k} - 1\right)}{c_p T_3\left(1 - r_p^{(1-k)/2k}\right)}$$

which simplifies to

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_3} r_p^{(k-1)/2k}$$

The thermal efficiency of the single stage ideal regenerative cycle is given as

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_3} r_p^{(k-1)/k}$$

Therefore, the regenerative cycle with two stages of expansion has a higher thermal efficiency than the standard regenerative cycle with a single stage of expansion for any given value of the pressure ratio r_p.

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9-179 A gas-turbine plant operates on the regenerative Brayton cycle with reheating and intercooling. The back work ratio, the net work output, the thermal efficiency, the second-law efficiency, and the exergies at the exits of the combustion chamber and the regenerator are to be determined.

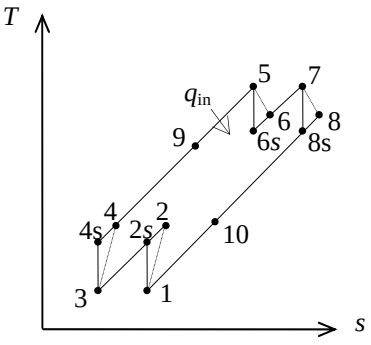
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K.

Analysis (*a*) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.

Optimum intercooling and reheating pressure is

$$P_2 = \sqrt{P_1 P_4} = \sqrt{(100)(1200)} = 346.4 \text{ kPa}$$



Process 1-2, 3-4: Compression

$$\left.\begin{matrix} T_1 = 300 \text{ K} \longrightarrow h_1 = 300.43 \text{ kJ/kg} \\ T_1 = 300 \text{ K} \\ P_1 = 100 \text{ kPa} \end{matrix}\right\} s_1 = 5.7054 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{matrix} P_2 = 346.4 \text{ kPa} \\ s_2 = s_1 = 5.7054 \text{ kJ/kg.K} \end{matrix}\right\} h_{2s} = 428.79 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.80 = \frac{428.79 - 300.43}{h_2 - 300.43} \longrightarrow h_2 = 460.88 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_3 = 350 \text{ K} \longrightarrow h_3 = 350.78 \text{ kJ/kg} \\ T_3 = 350 \text{ K} \\ P_3 = 346.4 \text{ kPa} \end{matrix}\right\} s_3 = 5.5040 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{matrix} P_4 = 1200 \text{ kPa} \\ s_4 = s_3 = 5.5040 \text{ kJ/kg.K} \end{matrix}\right\} h_{4s} = 500.42 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{4s} - h_3}{h_4 - h_3} \longrightarrow 0.80 = \frac{500.42 - 350.78}{h_4 - 350.78} \longrightarrow h_4 = 537.83 \text{ kJ/kg}$$

Process 6-7, 8-9: Expansion

$$\left.\begin{matrix} T_6 = 1400 \text{ K} \longrightarrow h_6 = 1514.9 \text{ kJ/kg} \\ T_6 = 1400 \text{ K} \\ P_6 = 1200 \text{ kPa} \end{matrix}\right\} s_6 = 6.6514 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{matrix} P_7 = 346.4 \text{ kPa} \\ s_7 = s_6 = 6.6514 \text{ kJ/kg.K} \end{matrix}\right\} h_{7s} = 1083.9 \text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} \longrightarrow 0.80 = \frac{1514.9 - h_7}{1514.9 - 1083.9} \longrightarrow h_7 = 1170.1 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_8 = 1300 \text{ K} \longrightarrow h_8 = 1395.6 \text{ kJ/kg} \\ T_8 = 1300 \text{ K} \\ P_8 = 346.4 \text{ kPa} \end{matrix}\right\} s_8 = 6.9196 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{matrix} P_9 = 100 \text{ kPa} \\ s_9 = s_8 = 6.9196 \text{ kJ/kg.K} \end{matrix}\right\} h_{9s} = 996.00 \text{ kJ/kg}$$

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$$\eta_T = \frac{h_8 - h_9}{h_8 - h_{9s}} \longrightarrow 0.80 = \frac{1395.6 - h_9}{1395.6 - 996.00} \longrightarrow h_9 = 1075.9 \text{ kJ/kg}$$

Cycle analysis:

$$w_{C,in} = h_2 - h_1 + h_4 - h_3 = 460.88 - 300.43 + 537.83 - 350.78 = 347.50 \text{ kJ/kg}$$

$$w_{T,out} = h_6 - h_7 + h_8 - h_9 = 1514.9 - 1170.1 + 1395.6 - 1075.9 = 664.50 \text{ kJ/kg}$$

$$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{347.50}{664.50} = \mathbf{0.523}$$

$$w_{net} = w_{T,out} - w_{C,in} = 664.50 - 347.50 = \mathbf{317.0 \text{ kJ/kg}}$$

Regenerator analysis:

$$\varepsilon_{regen} = \frac{h_9 - h_{10}}{h_9 - h_4} \longrightarrow 0.75 = \frac{1075.9 - h_{10}}{1075.9 - 537.83} \longrightarrow h_{10} = 672.36 \text{ kJ/kg}$$

$$\left.\begin{array}{l} h_{10} = 672.36 \text{ K} \\ P_{10} = 100 \text{ kPa} \end{array}\right\} s_{10} = 6.5157 \text{ kJ/kg} \cdot \text{K}$$

$$q_{regen} = h_9 - h_{10} = h_5 - h_4 \longrightarrow 1075.9 - 672.36 = h_5 - 537.83 \longrightarrow h_5 = 941.40 \text{ kJ/kg}$$

(*b*) $$q_{in} = h_6 - h_5 = 1514.9 - 941.40 = 573.54 \text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{317.0}{573.54} = \mathbf{0.553}$$

(*c*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{max} = 1 - \frac{T_1}{T_6} = 1 - \frac{300 \text{ K}}{1400 \text{ K}} = 0.786$$

and

$$\eta_{II} = \frac{\eta_{th}}{\eta_{max}} = \frac{0.553}{0.786} = \mathbf{0.704}$$

(*d*) The exergies at the combustion chamber exit and the regenerator exit are

$$\begin{aligned} x_6 &= h_6 - h_0 - T_0(s_6 - s_0) \\ &= (1514.9 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.6514 - 5.7054)\text{kJ/kg.K} \\ &= \mathbf{930.7 \text{ kJ/kg}} \end{aligned}$$

$$\begin{aligned} x_{10} &= h_{10} - h_0 - T_0(s_{10} - s_0) \\ &= (672.36 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.5157 - 5.7054)\text{kJ/kg.K} \\ &= \mathbf{128.8 \text{ kJ/kg}} \end{aligned}$$

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9-180 The thermal efficiency of a two-stage gas turbine with regeneration, reheating and intercooling to that of a three-stage gas turbine is to be compared.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

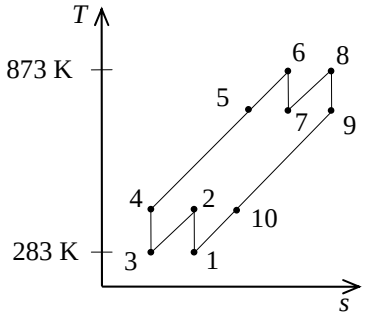
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis

Two Stages:

The pressure ratio across each stage is

$$r_p = \sqrt{16} = 4$$



The temperatures at the end of compression and expansion are

$$T_c = T_{\min} r_p^{(k-1)/k} = (283\,\text{K})(4)^{0.4/1.4} = 420.5\,\text{K}$$

$$T_e = T_{\max}\left(\frac{1}{r_p}\right)^{(k-1)/k} = (873\,\text{K})\left(\frac{1}{4}\right)^{0.4/1.4} = 587.5\,\text{K}$$

The heat input and heat output are

$$q_{\text{in}} = 2c_p(T_{\max} - T_e) = 2(1.005\,\text{kJ/kg}\cdot\text{K})(873 - 587.5)\,\text{K} = 573.9\,\text{kJ/kg}$$

$$q_{\text{out}} = 2c_p(T_c - T_{\min}) = 2(1.005\,\text{kJ/kg}\cdot\text{K})(420.5 - 283)\,\text{K} = 276.4\,\text{kJ/kg}$$

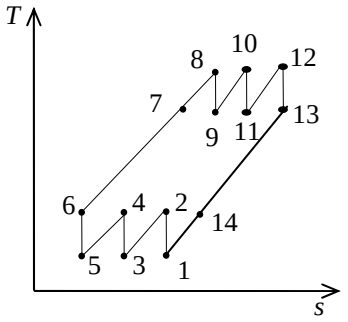
The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{276.4}{573.9} = \mathbf{0.518}$$

Three Stages:

The pressure ratio across each stage is

$$r_p = 16^{1/3} = 2.520$$



The temperatures at the end of compression and expansion are

$$T_c = T_{\min} r_p^{(k-1)/k} = (283\,\text{K})(2.520)^{0.4/1.4} = 368.5\,\text{K}$$

$$T_e = T_{\max}\left(\frac{1}{r_p}\right)^{(k-1)/k} = (873\,\text{K})\left(\frac{1}{2.520}\right)^{0.4/1.4} = 670.4\,\text{K}$$

The heat input and heat output are

$$q_{\text{in}} = 3c_p(T_{\max} - T_e) = 3(1.005\,\text{kJ/kg}\cdot\text{K})(873 - 670.4)\,\text{K} = 610.8\,\text{kJ/kg}$$

$$q_{\text{out}} = 3c_p(T_c - T_{\min}) = 3(1.005\,\text{kJ/kg}\cdot\text{K})(368.5 - 283)\,\text{K} = 257.8\,\text{kJ/kg}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{257.8}{610.8} = \mathbf{0.578}$$

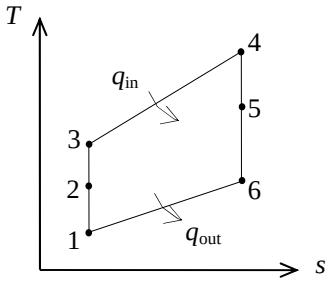
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9-181E A pure jet engine operating on an ideal cycle is considered. The thrust force produced per unit mass flow rate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are $R = 0.3704$ psia·ft^3/lbm·R (Table A-1E), $c_p = 0.24$ Btu/lbm·R and $k = 1.4$ (Table A-2Ea).

Analysis (*a*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1 = 1200$ ft/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\,\nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 490\text{ R} + \frac{(1200\text{ ft/s})^2}{(2)(0.24\text{ BtuJlbm}\cdot\text{R})}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 609.8\text{ R}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (10\text{ psia})\left(\frac{609.8\text{ R}}{490\text{ R}}\right)^{1.4/0.4} = 21.5\text{ psia}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (9)(21.5\text{ psia}) = 193.5\text{ psia}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (609.8\text{ R})(9)^{0.4/1.4} = 1142.4\text{ R}$$

Turbine:

$$w_{comp,in} = w_{turb,out} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or $$T_5 = T_4 - T_3 + T_2 = 1160 - 1142.4 + 609.8 = 627.4\text{ R}$$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (1160\text{ R})\left(\frac{10\text{ psia}}{193.5\text{ psia}}\right)^{0.4/1.4} = 497.5\text{ R}$$

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\,\nearrow 0}}{2} \longrightarrow 0 = c_p(T_6 - T_5) + V_6^2/2$$

or, $$V_6 = V_{exit} = \sqrt{(2)(0.24\text{ Btu/lbm}\cdot\text{R})(627.4 - 497.5)\text{R}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)} = 1249\text{ ft/s}$$

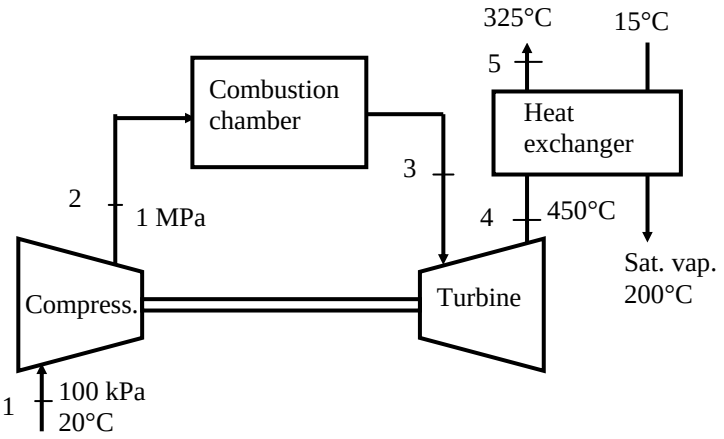
The specific impulse is then

$$\frac{F}{\dot{m}} = V_{exit} - V_{inlet} = 1249 - 1200 = \mathbf{49\ m/s}$$

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9-182 The electricity and the process heat requirements of a manufacturing facility are to be met by a cogeneration plant consisting of a gas-turbine and a heat exchanger for steam production. The mass flow rate of the air in the cycle, the back work ratio, the thermal efficiency, the rate at which steam is produced in the heat exchanger, and the utilization efficiency of the cogeneration plant are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.



Analysis (*a*) For this problem, we use the properties of air from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.

Process 1-2: Compression

$$T_1 = 20°\text{C} \longrightarrow h_1 = 293.5\ \text{kJ/kg}$$

$$\left.\begin{matrix} T_1 = 20°\text{C} \\ P_1 = 100\ \text{kPa} \end{matrix}\right\} s_1 = 5.682\ \text{kJ/kg} \cdot \text{K}$$

$$\left.\begin{matrix} P_2 = 1000\ \text{kPa} \\ s_2 = s_1 = 5.682\ \text{kJ/kg.K} \end{matrix}\right\} h_{2s} = 567.2\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.86 = \frac{567.2 - 293.5}{h_2 - 293.5} \longrightarrow h_2 = 611.8\ \text{kJ/kg}$$

Process 3-4: Expansion

$$T_4 = 450°\text{C} \longrightarrow h_4 = 738.5\ \text{kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.88 = \frac{h_3 - 738.5}{h_3 - h_{4s}}$$

We cannot find the enthalpy at state 3 directly. However, using the following lines in EES together with the isentropic efficiency relation, we find h_3 = 1262 kJ/kg, T_3 = 913.2ºC, s_3 = 6.507 kJ/kg.K. The solution by hand would require a trial-error approach.

```
h_3=enthalpy(Air, T=T_3)
s_3=entropy(Air, T=T_3, P=P_2)
h_4s=enthalpy(Air, P=P_1, s=s_3)
```

Also,

$$T_5 = 325°\text{C} \longrightarrow h_5 = 605.4\ \text{kJ/kg}$$

The inlet water is compressed liquid at 15ºC and at the saturation pressure of steam at 200ºC (1555 kPa). This is not available in the tables but we can obtain it in EES. The alternative is to use saturated liquid enthalpy at the given temperature.

$$\left.\begin{matrix} T_{w1} = 15°\text{C} \\ P_1 = 1555\ \text{kPa} \end{matrix}\right\} h_{w1} = 64.47\ \text{kJ/kg}$$

$$\left.\begin{matrix} T_{w2} = 200°\text{C} \\ x_2 = 1 \end{matrix}\right\} h_{w2} = 2792\ \text{kJ/kg}$$

The net work output is

$$w_{C,in} = h_2 - h_1 = 611.8 - 293.5 = 318.2\ \text{kJ/kg}$$

$$w_{T,out} = h_3 - h_4 = 1262 - 738.5 = 523.4\ \text{kJ/kg}$$

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$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 523.4 - 318.2 = 205.2 \text{ kJ/kg}$$

The mass flow rate of air is

$$\dot{m}_a = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{1500 \text{ kJ/s}}{205.2 \text{ kJ/kg}} = \mathbf{7.311 kg/s}$$

(*b*) The back work ratio is

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{318.2}{523.4} = \mathbf{0.608}$$

The rate of heat input and the thermal efficiency are

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_3 - h_2) = (7.311 \text{ kg/s})(1262 - 611.8)\text{kJ/kg} = 4753 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1500 \text{ kW}}{4753 \text{ kW}} = 0.3156 = \mathbf{31.6\%}$$

(*c*) An energy balance on the heat exchanger gives

$$\dot{m}_a (h_4 - h_5) = \dot{m}_w (h_{w2} - h_{w1})$$

$$(7.311 \text{ kg/s})(738.5 - 605.4)\text{kJ/kg} = \dot{m}_w (2792 - 64.47)\text{kJ/kg} \longrightarrow \dot{m}_w = \mathbf{0.3569 kg/s}$$

(*d*) The heat supplied to the water in the heat exchanger (process heat) and the utilization efficiency are

$$\dot{Q}_{\text{p}} = \dot{m}_w (h_{w2} - h_{w1}) = (0.3569 \text{ kg/s})(2792 - 64.47)\text{kJ/kg} = 973.5 \text{ kW}$$

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{p}}}{\dot{Q}_{\text{in}}} = \frac{1500 + 973.5}{4753 \text{ kW}} = 0.5204 = \mathbf{52.0\%}$$

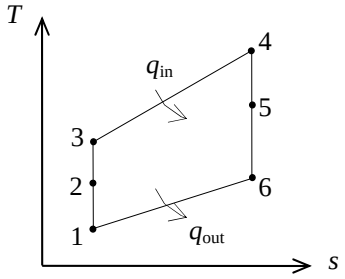
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9-183 A turbojet aircraft flying is considered. The pressure of the gases at the turbine exit, the mass flow rate of the air through the compressor, the velocity of the gases at the nozzle exit, the propulsive power, and the propulsive efficiency of the cycle are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-1).

Analysis (*a*) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.



Diffuser, Process 1-2:

$$T_1 = -35°\text{C} \longrightarrow h_1 = 238.23\ \text{kJ/kg}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$(238.23\ \text{kJ/kg}) + \frac{(900/3.6\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = h_2 + \frac{(15\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) \longrightarrow h_2 = 269.37\ \text{kJ/kg}$$

$$\left.\begin{matrix} h_2 = 269.37\ \text{kJ/kg} \\ P_2 = 50\ \text{kPa} \end{matrix}\right\} s_2 = 5.7951\ \text{kJ/kg}\cdot\text{K}$$

Compressor, Process 2-3:

$$\left.\begin{matrix} P_3 = 450\ \text{kPa} \\ s_3 = s_2 = 5.7951\ \text{kJ/kg.K} \end{matrix}\right\} h_{3s} = 505.19\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{3s} - h_2}{h_3 - h_2} \longrightarrow 0.83 = \frac{505.19 - 269.37}{h_3 - 269.37} \longrightarrow h_3 = 553.50\ \text{kJ/kg}$$

Turbine, Process 3-4:

$$T_4 = 950°\text{C} \longrightarrow h_4 = 1304.8\ \text{kJ/kg}$$

$$h_3 - h_2 = h_4 - h_5 \longrightarrow 553.50 - 269.37 = 1304.8 - h_5 \longrightarrow h_5 = 1020.6\ \text{kJ/kg}$$

where the mass flow rates through the compressor and the turbine are assumed equal.

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow 0.83 = \frac{1304.8 - 1020.6}{1304.8 - h_{5s}} \longrightarrow h_{5s} = 962.45\ \text{kJ/kg}$$

$$\left.\begin{matrix} T_4 = 950°\text{C} \\ P_4 = 450\ \text{kPa} \end{matrix}\right\} s_4 = 6.7725\ \text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{matrix} h_{5s} = 962.45\ \text{kJ/kg} \\ s_5 = s_4 = 6.7725\ \text{kJ/kg}\cdot\text{K} \end{matrix}\right\} P_5 = \mathbf{147.4\ kPa}$$

(*b*) The mass flow rate of the air through the compressor is

$$\dot{m} = \frac{\dot{W}_C}{h_3 - h_2} = \frac{500\ \text{kJ/s}}{(553.50 - 269.37)\ \text{kJ/kg}} = \mathbf{1.760\ kg/s}$$

(*c*) *Nozzle, Process 5-6:*

$$\left.\begin{matrix} h_5 = 1020.6\ \text{kJ/kg} \\ P_5 = 147.4\ \text{kPa} \end{matrix}\right\} s_5 = 6.8336\ \text{kJ/kg}\cdot\text{K}$$

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$$\left.\begin{array}{l} P_6 = 40\ \text{kPa} \\ s_6 = s_5 = 6.8336\ \text{kJ/kg.K} \end{array}\right\} h_{6s} = 709.66\ \text{kJ/kg}$$

$$\eta_N = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow 0.83 = \frac{1020.6 - h_6}{1020.6 - 709.66} \longrightarrow h_6 = 762.52\ \text{kJ/kg}$$

$$h_5 + \frac{V_5^2}{2} = h_6 + \frac{V_6^2}{2}$$

$$(1020.6\ \text{kJ/kg}) + 0 = 762.52\ \text{kJ/kg} + \frac{V_6^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) \longrightarrow V_6 = \mathbf{718.5\ m/s}$$

where the velocity at nozzle inlet is assumed zero.

(*d*) The propulsive power and the propulsive efficiency are

$$\dot{W}_p = \dot{m}(V_6 - V_1)V_1 = (1.76\ \text{kg/s})(718.5\ \text{m/s} - 250\ \text{m/s})(250\ \text{m/s})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{206.1\ kW}$$

$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = (1.76\ \text{kg/s})(1304.8 - 553.50)\text{kJ/kg} = 1322\ \text{kW}$$

$$\eta_p = \frac{\dot{W}_p}{\dot{Q}_{\text{in}}} = \frac{206.1\ \text{kW}}{1322\ \text{kW}} = \mathbf{0.156}$$

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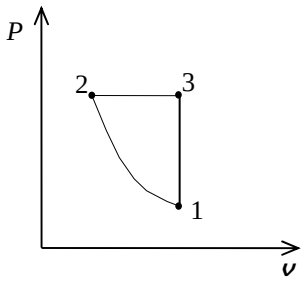
9-184 The three processes of an air standard cycle are described. The cycle is to be shown on the *P*-ν and *T*-*s* diagrams, and the expressions for back work ratio and the thermal efficiency are to be obtained.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis (a) The *P*-ν and *T*-*s* diagrams for this cycle are as shown.

(b) The work of compression is found by the first law for process 1-2:

$$q_{1-2} - w_{1-2} = \Delta u_{1-2}$$
$$q_{1-2} = 0 (isentropic\ \ process)$$
$$w_{1-2} = -\Delta u_{1-2} = -C_v\left(T_2 - T_1\right)$$
$$w_{comp} = -w_{1-2} = C_v\left(T_2 - T_1\right)$$

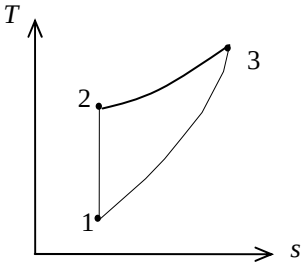


The expansion work is found by

$$w_{\text{exp}} = w_{2-3} = \int_2^3 P dv = P\left(v_3 - v_2\right) = R\left(T_3 - T_2\right)$$

The back work ratio is

$$\frac{w_{comp}}{w_{\text{exp}}} = \frac{C_v\left(T_3 - T_1\right)}{R\left(T_3 - T_2\right)} = \frac{C_v}{R}\frac{T_1}{T_2}\frac{\left(T_3/T_1 - 1\right)}{\left(T_3/T_2 - 1\right)}$$



Process 1-2 is isentropic; therefore,

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \frac{1}{r^{k-1}} \text{ and } \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k = r^k$$

Process 2-3 is constant pressure; therefore,

$$\frac{P_3 V_3}{T_3} = \frac{P_2 V_2}{T_2} \Rightarrow \frac{T_3}{T_2} = \frac{V_3}{V_2} = \frac{V_1}{V_2} = r$$

Process 3-1 is constant volume; therefore,

$$\frac{P_3 V_3}{T_3} = \frac{P_1 V_1}{T_1} \Rightarrow \frac{T_3}{T_1} = \frac{P_3}{P_1} = \frac{P_2}{P_1} = r^k$$

The back work ratio becomes (C_v=R/(k-1))

$$\frac{w_{comp}}{w_{\text{exp}}} = \frac{1}{k-1}\frac{1}{r^{k-1}}\frac{r^{k-1}-1}{r-1}$$

(c) Apply first law to the closed system for processes 2-3 and 3-1 to show:

$$q_{in} = C_p\left(T_3 - T_2\right)$$
$$q_{out} = C_v\left(T_3 - T_1\right)$$

The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_v\left(T_3 - T_1\right)}{C_p\left(T_3 - T_2\right)} = 1 - \frac{1}{k}\frac{T_1\left(T_3/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right)}$$

The efficiency becomes

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$$\eta_{th} = 1 - \frac{1}{k}\frac{1}{r^{k-1}}\frac{r^k - 1}{r - 1}$$

(d) Determine the value of the back work ratio and efficiency as r goes to unity.

$$\frac{w_{comp}}{w_{\exp}} = \frac{1}{k-1}\frac{1}{r^{k-1}}\frac{r^{k-1} - 1}{r - 1}$$

$$\lim_{r\to 1}\frac{w_{comp}}{w_{\exp}} = \frac{1}{k-1}\left\{\lim_{r\to 1}\frac{1}{r^{k-1}}\frac{r^{k-1} - 1}{r - 1}\right\} = \frac{1}{k-1}\left\{\lim_{r\to 1}\frac{r^{k-1} - 1}{r^k - r^{k-1}}\right\} = \frac{1}{k-1}\left\{\lim_{r\to 1}\frac{(k-1)r^{k-2}}{kr^{k-1} - (k-1)r^{k-2}}\right\}$$

$$\lim_{r\to 1}\frac{w_{comp}}{w_{\exp}} = \frac{1}{k-1}\left\{\frac{k-1}{k-k+1}\right\} = \frac{1}{k-1}\left\{\frac{k-1}{1}\right\} = 1$$

$$\eta_{th} = 1 - \frac{1}{k}\frac{1}{r^{k-1}}\frac{r^k - 1}{r - 1}$$

$$\lim_{r\to 1}\eta_{th} = 1 - \frac{1}{k}\left\{\lim_{r\to 1}\frac{1}{r^{k-1}}\frac{r^k - 1}{r - 1}\right\} = 1 - \frac{1}{k}\left\{\lim_{r\to 1}\frac{r^k - 1}{r^k - r^{k-1}}\right\} = 1 - \frac{1}{k}\left\{\lim_{r\to 1}\frac{kr^{k-1}}{kr^{k-1} - (k-1)r^{k-2}}\right\}$$

$$\lim_{r\to 1}\eta_{th} = 1 - \frac{1}{k}\left\{\frac{k}{k-k+1}\right\} = 1 - \frac{1}{k}\left\{\frac{k}{1}\right\} = 0$$

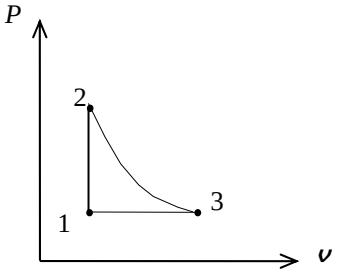
These results show that if there is no compression (i.e. $r = 1$), there can be no expansion and no net work will be done even though heat may be added to the system.

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9-185 The three processes of an air standard cycle are described. The cycle is to be shown on the *P*-ν and *T*-*s* diagrams, and the expressions for back work ratio and the thermal efficiency are to be obtained.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis (a) The *P*-ν and *T*-*s* diagrams for this cycle are as shown.

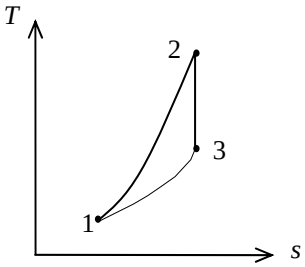


(b) The work of expansion is found by the first law for process 2-3:

$$q_{2-3} - w_{2-3} = \Delta u_{2-3}$$
$$q_{2-3} = 0 (isentropic\ \ process)$$
$$w_{2-3} = -\Delta u_{2-3} = -C_v\left(T_3 - T_2\right)$$
$$w_{\exp} = w_{2-3} = C_v\left(T_2 - T_3\right)$$

The compression work is found by

$$w_{comp} = -w_{3-1} = -\int_3^1 P dv = -P\left(v_1 - v_3\right) = R\left(T_3 - T_1\right)$$



The back work ratio is

$$\frac{w_{comp}}{w_{\exp}} = \frac{C_v\left(T_3 - T_1\right)}{R\left(T_2 - T_3\right)} = \frac{C_v}{R}\frac{T_3}{T_3}\frac{\left(1 - T_1/T_3\right)}{\left(T_2/T_3 - 1\right)} = \frac{C_v}{R}\frac{\left(1 - T_1/T_3\right)}{\left(T_2/T_3 - 1\right)}$$

Process 3-1 is constant pressure; therefore,

$$\frac{P_3 V_3}{T_3} = \frac{P_1 V_1}{T_1} \Rightarrow \frac{T_1}{T_3} = \frac{V_1}{V_3} = \frac{V_2}{V_3} = \frac{1}{r}$$

Process 2-3 is isentropic; therefore,

$$\frac{T_2}{T_3} = \left(\frac{V_2}{V_2}\right)^{k-1} = r^{k-1} \text{ and } \frac{P_2}{P_3} = \left(\frac{V_3}{V_2}\right)^{k} = r^{k}$$

The back work ratio becomes (C_v=R/(k-1))

$$\frac{w_{comp}}{w_{\exp}} = \left(k - 1\right)\frac{1 - \frac{1}{r}}{r^{k-1} - 1} = \frac{k-1}{r}\frac{r-1}{r^{k-1} - 1}$$

(c) Apply first law to the closed system for processes 1-2 and 3-1 to show:

$$q_{in} = C_v\left(T_2 - T_1\right)$$
$$q_{out} = C_p\left(T_3 - T_1\right)$$

The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p\left(T_3 - T_1\right)}{C_v\left(T_2 - T_1\right)} = 1 - k\frac{T_1\left(T_3/T_1 - 1\right)}{T_1\left(T_2/T_1 - 1\right)}$$

Process 1-2 is constant volume; therefore,

$$\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1} \Rightarrow \frac{T_2}{T_1} = \frac{P_2}{P_1} = \frac{P_2}{P_3} = r^{k}$$

The efficiency becomes

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$$\eta_{th} = 1 - k\frac{r-1}{r^k - 1}$$

(d) Determine the value of the back work ratio and efficiency as r goes to unity.

$$\frac{w_{comp}}{w_{\exp}} = (k-1)\frac{1-\frac{1}{r}}{r^{k-1}-1} = \frac{k-1}{r}\frac{r-1}{r^{k-1}-1}$$

$$\lim_{r\to 1}\frac{w_{comp}}{w_{\exp}} = (k-1)\left\{\lim_{r\to 1}\frac{r-1}{r^k - r}\right\} = (k-1)\left\{\lim_{r\to 1}\frac{1}{kr^{k-1}-1}\right\}$$

$$\lim_{r\to 1}\frac{w_{comp}}{w_{\exp}} = (k-1)\left\{\frac{1}{k-1}\right\} = 1$$

$$\eta_{th} = 1 - k\frac{r-1}{r^k - 1}$$

$$\lim_{r\to 1}\eta_{th} = 1 - k\left\{\lim_{r\to 1}\frac{r-1}{r^k - 1}\right\} = 1 - k\left\{\lim_{r\to 1}\frac{1}{kr^{k-1}}\right\}$$

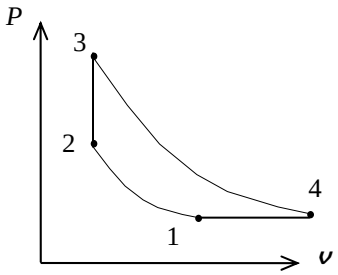
$$\lim_{r\to 1}\eta_{th} = 1 - k\left\{\frac{1}{k}\right\} = 0$$

These results show that if there is no compression (i.e. r = 1), there can be no expansion and no net work will be done even though heat may be added to the system.

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9-186 The four processes of an air-standard cycle are described. The cycle is to be shown on the *P*-ν and *T*-*s* diagrams; an expression for the cycle thermal efficiency is to be obtained; and the limit of the efficiency as the volume ratio during heat rejection approaches unity is to be evaluated.

Analysis (a) The *P*-ν and *T*-*s* diagrams of the cycle are shown in the figures.



(b) Apply first law to the closed system for processes 2-3 and 4-1 to show:

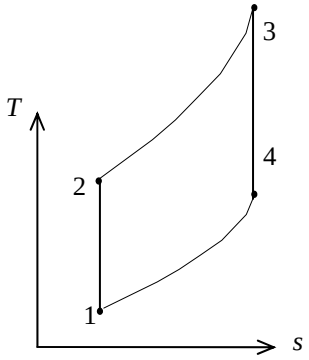
$$q_{in} = C_v\left(T_3 - T_2\right)$$
$$q_{out} = C_p\left(T_4 - T_1\right)$$

The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p\left(T_4 - T_1\right)}{C_v\left(T_3 - T_2\right)} = 1 - k\frac{T_1\left(T_4/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right)}$$

Process 1-2 is isentropic; therefore, $\dfrac{T_1}{T_2} = \left(\dfrac{V_2}{V_1}\right)^{k-1} = \dfrac{1}{r^{k-1}}$

Process 3-4 is isentropic; therefore, $\dfrac{T_3}{T_4} = \left(\dfrac{V_4}{V_3}\right)^{k-1} = r_e^{k-1}$



Process 4-1 is constant pressure; therefore,

$$\frac{P_4V_4}{T_4} = \frac{P_1V_1}{T_1} \Rightarrow \frac{T_4}{T_1} = \frac{V_4}{V_1} = r_p$$

$$\frac{T_3}{T_2} = \frac{T_3}{T_4}\frac{T_4}{T_1}\frac{T_1}{T_2} = r_e^{k-1}r_p\frac{1}{r^{k-1}} = \left(\frac{r_e}{r}\right)^{k-1} r_p$$

Since process 2-3 is constant volume and $V_3 = V_2$,

$$r_e = \frac{V_4}{V_3} = \frac{V_4}{V_2} = \frac{V_4}{V_1}\frac{V_1}{V_2} = r_p r$$

$$\frac{T_3}{T_2} = \left(\frac{r_p r}{r}\right)^{k-1} r_p = r_p^k$$

The efficiency becomes

$$\eta_{th} = 1 - k\frac{1}{r^{k-1}}\frac{r_p - 1}{r_p^k - 1}$$

(c) In the limit as r_p approaches unity, the cycle thermal efficiency becomes

$$\lim_{r_p \to 1}\eta_{th} = 1 - k\frac{1}{r^{k-1}}\left\{\lim_{r_p \to 1}\frac{r_p - 1}{r_p^k - 1}\right\} = 1 - k\frac{1}{r^{k-1}}\left\{\lim_{r_p \to 1}\frac{1}{kr_p^{k-1}}\right\}$$

$$\lim_{r_p \to 1}\eta_{th} = 1 - k\frac{1}{r^{k-1}}\left\{\frac{1}{k}\right\} = 1 - \frac{1}{r^{k-1}} = \eta_{th\ Otto}$$

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9-187 The four processes of an air-standard cycle are described. The back work ratio and its limit as r_p goes to unity are to be determined, and the result is to be compared to the expression for the Otto cycle.

Analysis The work of compression for process 1-2 is found by the first law:

$$q_{1-2} - w_{1-2} = \Delta u_{1-2}$$
$$q_{1-2} = 0 (isentropic\ \ process)$$
$$w_{1-2} = -\Delta u_{1-2} = -C_v \left(T_2 - T_1\right)$$
$$w_{comp,1-2} = -w_{1-2} = C_v \left(T_2 - T_1\right)$$

The work of compression for process 4-1 is found by

$$w_{comp,4-1} = -w_{4-1} = -\int_4^1 P dv = -P\left(v_1 - v_4\right) = R\left(T_4 - T_1\right)$$

The work of expansion for process 3-4 is found by the first law:

$$q_{3-4} - w_{3-4} = \Delta u_{3-4}$$
$$q_{3-4} = 0 (isentropic\ \ process)$$
$$w_{3-4} = -\Delta u_{3-4} = -C_v \left(T_4 - T_3\right)$$
$$w_{\exp,3-4} = -w_{3-4} = C_v \left(T_3 - T_4\right)$$

The back work ratio is

$$\frac{w_{comp}}{w_{\exp}} = \frac{R\left(T_4 - T_1\right) + C_v \left(T_2 - T_1\right)}{C_v \left(T_3 - T_4\right)} = \frac{T_1}{T_3} \frac{\dfrac{R}{C_v}\left(T_4 / T_1 - 1\right) + \left(T_2 / T_1 - 1\right)}{\left(1 - T_4 / T_3\right)}$$

Using data from the previous problem and $C_v = R/(k-1)$

$$\frac{w_{comp}}{w_{\exp}} = \frac{T_1}{T_3} \frac{(k-1)\left(r_p - 1\right) + \left(r^{k-1} - 1\right)}{\left(1 - \dfrac{1}{r_p^{k-1} r^{k-1}}\right)}$$

$$\lim_{r_p \to 1} \frac{w_{comp}}{w_{\exp}} = \frac{T_1}{T_3} \left\{ \lim_{r_p \to 1} \frac{(k-1)\left(r_p - 1\right) + \left(r^{k-1} - 1\right)}{\left(1 - \dfrac{1}{r_p^{k-1} r^{k-1}}\right)} \right\} = \frac{T_1}{T_3} \left\{ \frac{(k-1)(0) + \left(r^{k-1} - 1\right)}{1 - \dfrac{1}{r^{k-1}}} \right\}$$

$$\lim_{r_p \to 1} \frac{w_{comp}}{w_{\exp}} = \frac{T_1}{T_3} \left\{ \frac{r^{k-1} - 1}{1 - \dfrac{1}{r^{k-1}}} \right\}$$

This result is the same expression for the back work ratio for the Otto cycle.

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9-188 The effects of compression ratio on the net work output and the thermal efficiency of the Otto cycle for given operating conditions is to be investigated.

Analysis Using EES, the problem is solved as follows:

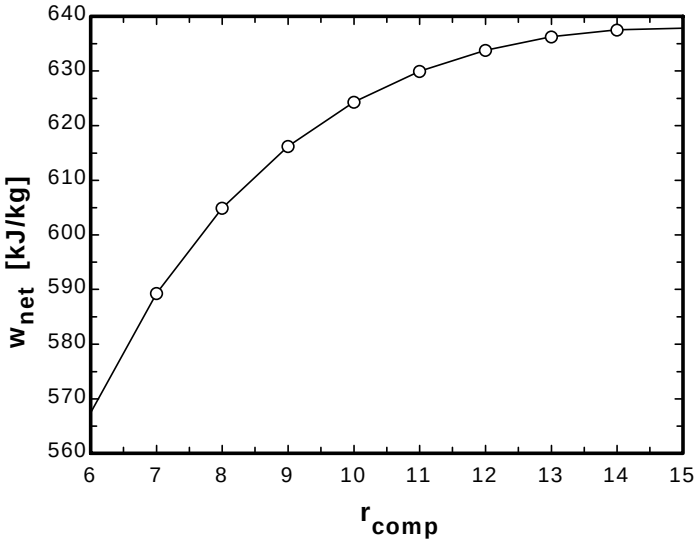
```
"Input Data"
T[1]=300 [K]
P[1]=100 [kPa]
T[3] = 2000 [K]
r_comp = 12

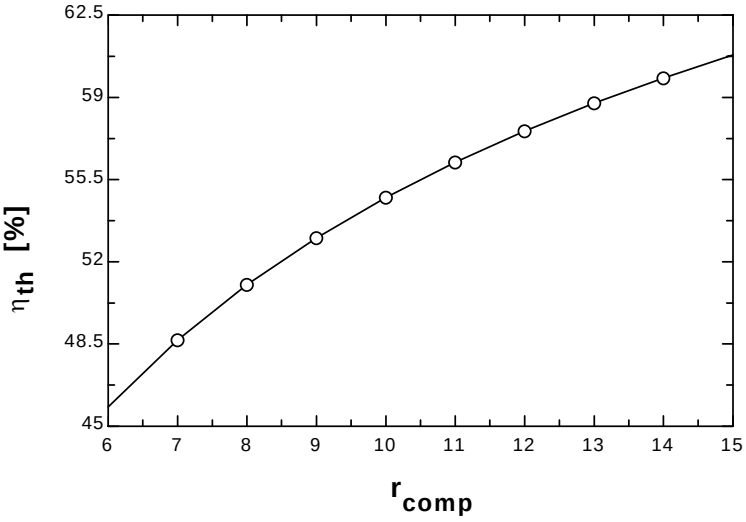
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0 "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*Convert(, %) "Thermal efficiency, in percent"
```

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η_{th} [%]	r_{comp}	w_{net} [kJ/kg]
45.83	6	567.4
48.67	7	589.3
51.03	8	604.9
53.02	9	616.2
54.74	10	624.3
56.24	11	630
57.57	12	633.8
58.75	13	636.3
59.83	14	637.5
60.8	15	637.9





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9-189 The effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle is to be investigated. The pressure ratios at which the net work output and the thermal efficiency are maximum are to be determined.

Analysis Using EES, the problem is solved as follows:

```
P_ratio = 8
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 1800 [K]
m_dot = 1 [kg/s]
Eta_c = 100/100
Eta_t = 100/100

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

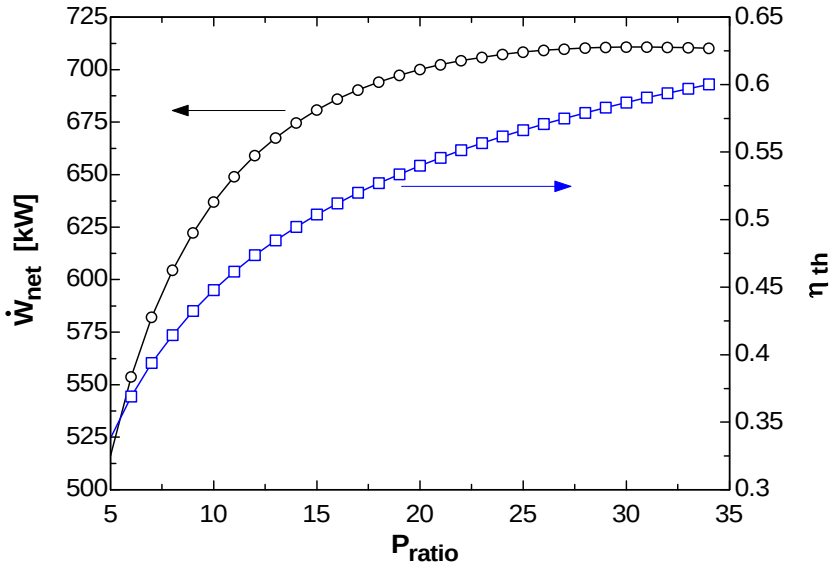
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c = W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"

"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0"
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency, Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature(air,h=h[2])
T[4]=temperature(air,h=h[4])
s[2]=entropy(air,T=T[2],P=P[2])
s[4]=entropy(air,T=T[4],P=P[4])
```

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Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.254	0.3383	5	175.8	516.3	692.1	1526
0.2665	0.3689	6	201.2	553.7	754.9	1501
0.2776	0.3938	7	223.7	582.2	805.9	1478
0.2876	0.4146	8	244.1	604.5	848.5	1458
0.2968	0.4324	9	262.6	622.4	885	1439
0.3052	0.4478	10	279.7	637	916.7	1422
0.313	0.4615	11	295.7	649	944.7	1406
0.3203	0.4736	12	310.6	659.1	969.6	1392
0.3272	0.4846	13	324.6	667.5	992.1	1378
0.3337	0.4945	14	337.8	674.7	1013	1364
0.3398	0.5036	15	350.4	680.8	1031	1352
0.3457	0.512	16	362.4	685.9	1048	1340
0.3513	0.5197	17	373.9	690.3	1064	1328
0.3567	0.5269	18	384.8	694.1	1079	1317
0.3618	0.5336	19	395.4	697.3	1093	1307
0.3668	0.5399	20	405.5	700	1106	1297
0.3716	0.5458	21	415.3	702.3	1118	1287
0.3762	0.5513	22	424.7	704.3	1129	1277
0.3806	0.5566	23	433.8	705.9	1140	1268
0.385	0.5615	24	442.7	707.2	1150	1259
0.3892	0.5663	25	451.2	708.3	1160	1251
0.3932	0.5707	26	459.6	709.2	1169	1243
0.3972	0.575	27	467.7	709.8	1177	1234
0.401	0.5791	28	475.5	710.3	1186	1227
0.4048	0.583	29	483.2	710.6	1194	1219
0.4084	0.5867	30	490.7	710.7	1201	1211
0.412	0.5903	31	498	710.8	1209	1204
0.4155	0.5937	32	505.1	710.7	1216	1197
0.4189	0.597	33	512.1	710.4	1223	1190
0.4222	0.6002	34	518.9	710.1	1229	1183



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9-190 The effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle is to be investigated assuming adiabatic efficiencies of 85 percent for both the turbine and the compressor. The pressure ratios at which the net work output and the thermal efficiency are maximum are to be determined.

Analysis Using EES, the problem is solved as follows:

```
P_ratio = 8
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 1800 [K]
m_dot = 1 [kg/s]
Eta_c = 80/100
Eta_t = 80/100

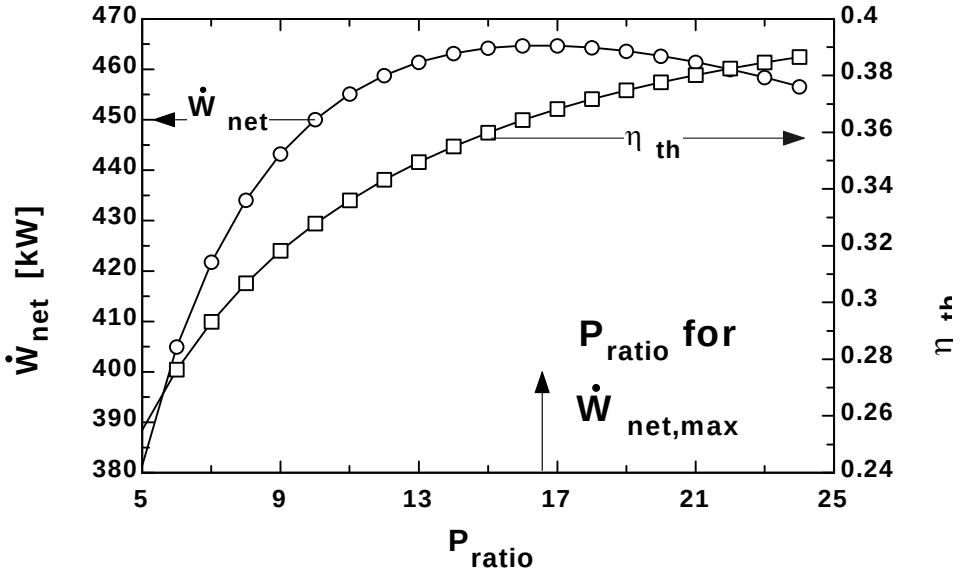
"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c = W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0"
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency, Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature(air,h=h[2])
T[4]=temperature(air,h=h[4])
s[2]=entropy(air,T=T[2],P=P[2])
s[4]=entropy(air,T=T[4],P=P[4])
```

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Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.3515	0.2551	5	206.8	381.5	588.3	1495
0.3689	0.2764	6	236.7	405	641.7	1465
0.3843	0.2931	7	263.2	421.8	685	1439
0.3981	0.3068	8	287.1	434.1	721.3	1415
0.4107	0.3182	9	309	443.3	752.2	1393
0.4224	0.3278	10	329.1	450.1	779.2	1373
0.4332	0.3361	11	347.8	455.1	803	1354
0.4433	0.3432	12	365.4	458.8	824.2	1337
0.4528	0.3495	13	381.9	461.4	843.3	1320
0.4618	0.355	14	397.5	463.2	860.6	1305
0.4704	0.3599	15	412.3	464.2	876.5	1290
0.4785	0.3643	16	426.4	464.7	891.1	1276
0.4862	0.3682	17	439.8	464.7	904.6	1262
0.4937	0.3717	18	452.7	464.4	917.1	1249
0.5008	0.3748	19	465.1	463.6	928.8	1237
0.5077	0.3777	20	477.1	462.6	939.7	1225
0.5143	0.3802	21	488.6	461.4	950	1214
0.5207	0.3825	22	499.7	460	959.6	1202
0.5268	0.3846	23	510.4	458.4	968.8	1192
0.5328	0.3865	24	520.8	456.6	977.4	1181



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9-191 The effects of pressure ratio, maximum cycle temperature, and compressor and turbine inefficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid is to be investigated. Constant specific heats at room temperature are to be used.

Analysis Using EES, the problem is solved as follows:

```
Procedure ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
"For Air:"
C_V = 0.718 [kJ/kg-K]
k = 1.4
T2 = T[1]*r_comp^(k-1)
P2 = P[1]*r_comp^k
q_in_23 = C_V*(T[3]-T2)
T4 = T[3]*(1/r_comp)^(k-1)
q_out_41 = C_V*(T4-T[1])
Eta_th_ConstProp = (1-q_out_41/q_in_23)*Convert(, %) "[%]"
"The Easy Way to calculate the constant property Otto cycle efficiency is:"
Eta_th_easy = (1 - 1/r_comp^(k-1))*Convert(, %) "[%]"
END

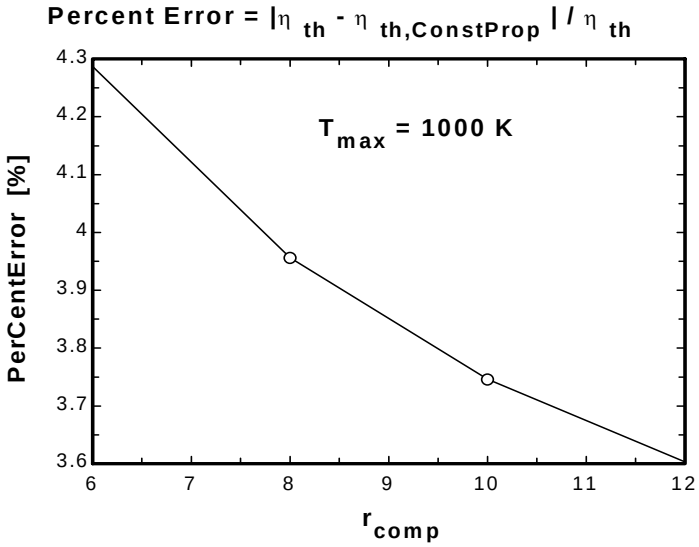
"Input Data"
T[1]=300 [K]
P[1]=100 [kPa]
{T[3] = 1000 [K]}
r_comp = 12

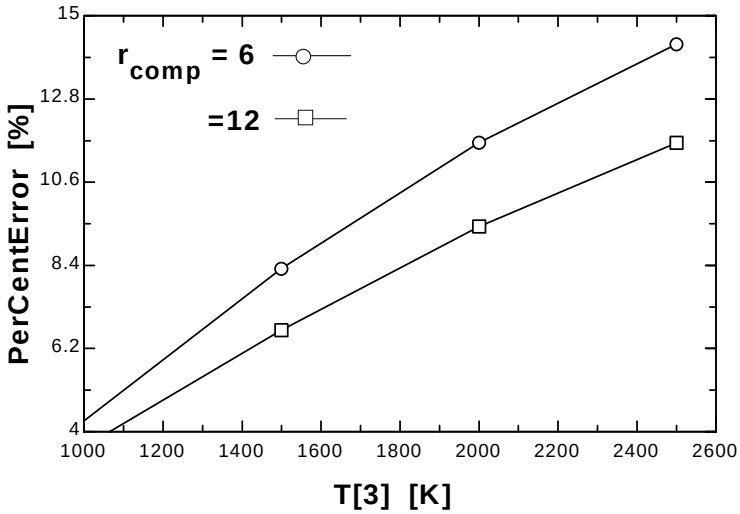
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
```

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```
w_41 =0 "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*Convert(, %) "Thermal efficiency, in percent"
Call ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
PerCentError = ABS(Eta_th - Eta_th_ConstProp)/Eta_th*Convert(, %) "[%]"
```

PerCentError [%]	r_{comp}	η_{th} [%]	$\eta_{th,ConstProp}$ [%]	$\eta_{th,easy}$ [%]	T_3 [K]
3.604	12	60.8	62.99	62.99	1000
6.681	12	59.04	62.99	62.99	1500
9.421	12	57.57	62.99	62.99	2000
11.64	12	56.42	62.99	62.99	2500





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9-192 The effects of pressure ratio, maximum cycle temperature, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid is to be investigated. Variable specific heats are to be used.

Analysis Using EES, the problem is solved as follows:

```
"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 800 [K]
m_dot = 1 [kg/s]
Eta_c = 75/100
Eta_t = 82/100}

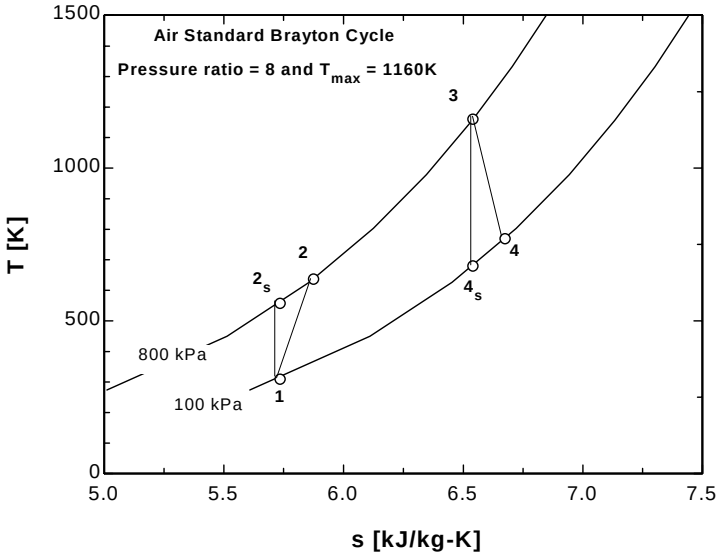
"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

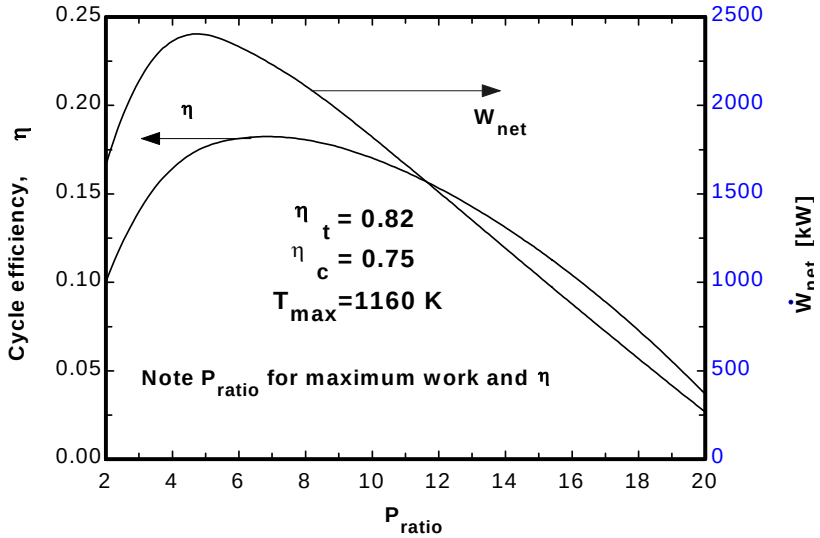
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c = W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]   "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0"
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency, Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy(air,T=T[2],P=P[2])
s[4]=entropy(air,T=T[4],P=P[4])
```

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Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.5229	0.1	2	1818	1659	3477	16587
0.6305	0.1644	4	4033	2364	6396	14373
0.7038	0.1814	6	5543	2333	7876	12862
0.7611	0.1806	8	6723	2110	8833	11682
0.8088	0.1702	10	7705	1822	9527	10700
0.85	0.1533	12	8553	1510	10063	9852
0.8864	0.131	14	9304	1192	10496	9102
0.9192	0.1041	16	9980	877.2	10857	8426
0.9491	0.07272	18	10596	567.9	11164	7809
0.9767	0.03675	20	11165	266.1	11431	7241





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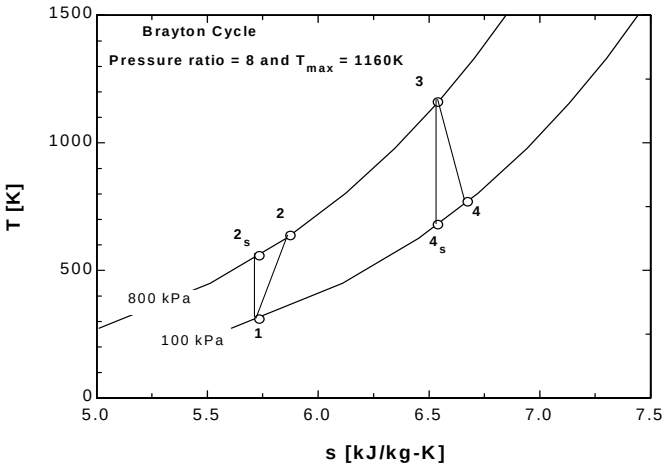
9-193 The effects of pressure ratio, maximum cycle temperature, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with helium as the working fluid is to be investigated.

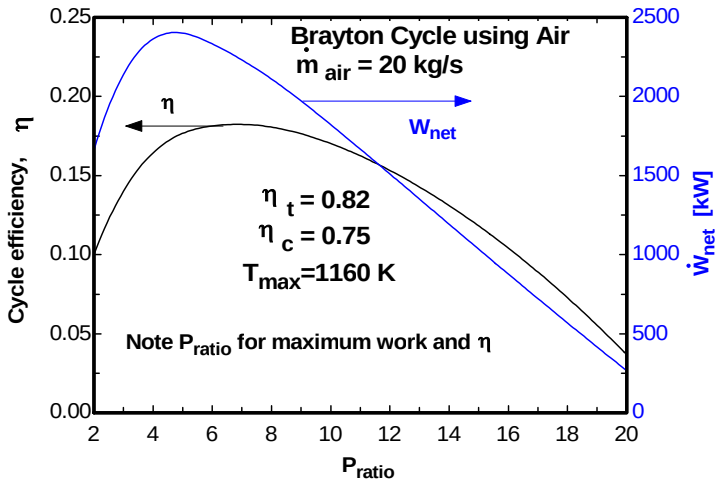
Analysis Using EES, the problem is solved as follows:

```
Function hFunc(WorkFluid$,T,P)
"The EES functions teat helium as a real gas; thus, T and P are needed for helium's enthalpy."
IF WorkFluid$ = 'Air' then hFunc:=enthalpy(Air,T=T) ELSE
        hFunc: = enthalpy(Helium,T=T,P=P)
endif
END
Procedure EtaCheck(Eta_th:EtaError$)
If Eta_th < 0 then EtaError$ = 'Why are the net work done and efficiency < 0?' Else EtaError$ = ''
END
"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 800 [K]
m_dot = 1 [kg/s]
Eta_c = 0.8
Eta_t = 0.8
WorkFluid$ = 'Helium'}
"Inlet conditions"
h[1]=hFunc(WorkFluid$,T[1],P[1])
s[1]=ENTROPY(WorkFluid$,T=T[1],P=P[1])
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(WorkFluid$,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at compressor exit"
h_s[2]=hFunc(WorkFluid$,T_s[2],P[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c = W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=hFunc(WorkFluid$,T[3],P[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0"
"Turbine analysis"
s[3]=ENTROPY(WorkFluid$,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(WorkFluid$,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=hFunc(WorkFluid$,T_s[4],P[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency, Wts_dot >
W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming: adiabatic, ke=pe=0"
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta_th=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Call EtaCheck(Eta_th:EtaError$)
Bwr=W_dot_c/W_dot_t "Back work ratio"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature(air,h=h[2])
T[4]=temperature(air,h=h[4])
s[2]=entropy(air,T=T[2],P=P[2])
s[4]=entropy(air,T=T[4],P=P[4])
```

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Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.5229	0.1	2	1818	1659	3477	16587
0.6305	0.1644	4	4033	2364	6396	14373
0.7038	0.1814	6	5543	2333	7876	12862
0.7611	0.1806	8	6723	2110	8833	11682
0.8088	0.1702	10	7705	1822	9527	10700
0.85	0.1533	12	8553	1510	10063	9852
0.8864	0.131	14	9304	1192	10496	9102
0.9192	0.1041	16	9980	877.2	10857	8426
0.9491	0.07272	18	10596	567.9	11164	7809
0.9767	0.03675	20	11165	266.1	11431	7241





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9-194 The effect of the number of compression and expansion stages on the thermal efficiency of an ideal regenerative Brayton cycle with multistage compression and expansion and air as the working fluid is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
"Input data for air"
C_P = 1.005 [kJ/kg-K]
k = 1.4
"Nstages is the number of compression and expansion stages"
Nstages = 1
T_6 = 1200 [K]
Pratio = 12
T_1 = 300 [K]
P_1= 100 [kPa]
Eta_reg = 1.0 "regenerator effectiveness"
Eta_c =1.0 "Compressor isentorpic efficiency"
Eta_t =1.0 "Turbine isentropic efficiency"
R_p = Pratio^(1/Nstages)
"Isentropic Compressor anaysis"
T_2s = T_1*R_p^((k-1)/k)
P_2 = R_p*P_1
"T_2s is the isentropic value of T_2 at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_2s-T_1)
"Actual compressor analysis:"
 w_comp = C_P*(T_2 - T_1)

"Since intercooling is assumed to occur such that T_3 = T_1 and the compressors have the same pressure ratio, the work input to each compressor is the same.  The total compressor work is:"
w_comp_total = Nstages*w_comp
"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
"The heat added in the external heat exchanger + the reheat between turbines is"
q_in_total = C_P*(T_6 - T_5) +(Nstages - 1)*C_P*(T_8 - T_7)
"Reheat is assumed to occur until:"
T_8 = T_6
"Turbine analysis"
P_7 = P_6 /R_p
"T_7s is the isentropic value of T_7 at turbine exit"
T_7s = T_6*(1/R_p)^((k-1)/k)
"Turbine adiabatic efficiency, w_turbisen > w_turb"
Eta_t = w_turb /w_turbisen
"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T_6 -  T_7s)
"Actual Turbine analysis:"
w_turb = C_P*(T_6 -  T_7)
w_turb_total = Nstages*w_turb

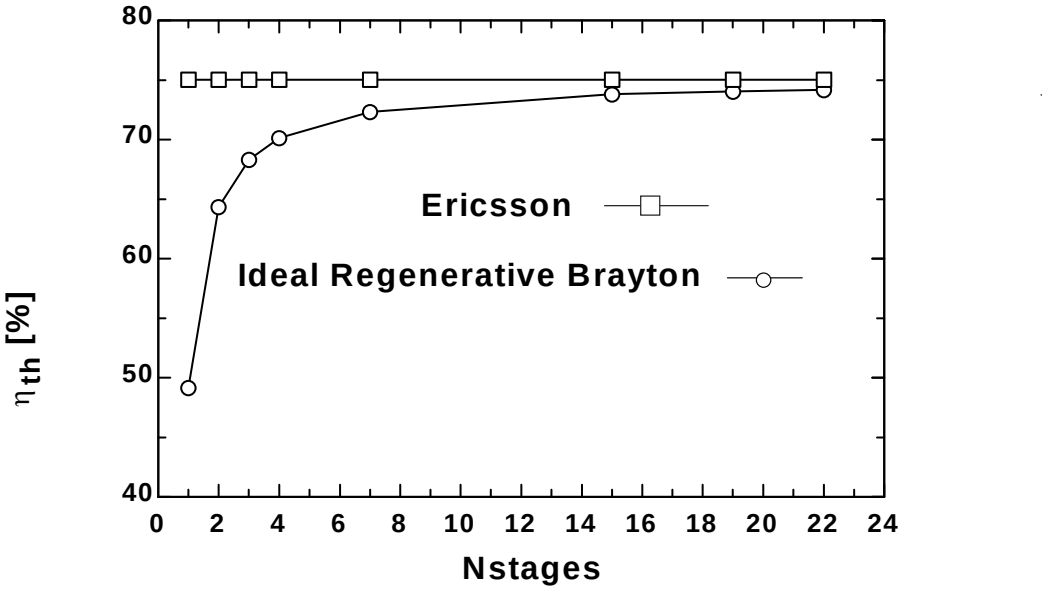
"Cycle analysis"
w_net=w_turb_total-w_comp_total "[kJ/kg]"
Bwr=w_comp/w_turb "Back work ratio"
```

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```
P_4=P_2
P_5=P_4
P_6=P_5
T_4 = T_2
"The regenerator effectiveness gives T_5 as:"
Eta_reg = (T_5 - T_4)/(T_9 - T_4)
T_9 = T_7
"Energy balance on regenerator gives T_10 as:"
T_4 + T_9=T_5  + T_10
"Cycle thermal efficiency with regenerator"
Eta_th_regenerative=w_net/q_in_total*Convert(, %) "[%]"

"The efficiency of the Ericsson cycle is the same as the Carnot cycle operating between the same max and min temperatures, T_6 and T_1 for this problem."
Eta_th_Ericsson = (1 - T_1/T_6)*Convert(, %) "[%]"
```

$\eta_{th,Ericksson}$ [%]	$\eta_{th,Regenerative}$ [%]	Nstages
75	49.15	1
75	64.35	2
75	68.32	3
75	70.14	4
75	72.33	7
75	73.79	15
75	74.05	19
75	74.18	22



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9-195 The effect of the number of compression and expansion stages on the thermal efficiency of an ideal regenerative Brayton cycle with multistage compression and expansion and helium as the working fluid is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
"Input data for Helium"
C_P = 5.1926 [kJ/kg-K]
k = 1.667
"Nstages is the number of compression and expansion stages"
{Nstages = 1}
T_6 = 1200 [K]
Pratio = 12
T_1 = 300 [K]
P_1= 100 [kPa]
Eta_reg = 1.0 "regenerator effectiveness"
Eta_c =1.0 "Compressor isentorpic efficiency"
Eta_t =1.0 "Turbine isentropic efficiency"
R_p = Pratio^(1/Nstages)
"Isentropic Compressor anaysis"

T_2s = T_1*R_p^((k-1)/k)
P_2 = R_p*P_1
"T_2s is the isentropic value of T_2 at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"
"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_2s-T_1)
"Actual compressor analysis:"
 w_comp = C_P*(T_2 - T_1)
"Since intercooling is assumed to occur such that T_3 = T_1 and the compressors have the same pressure
ratio, the work input to each compressor is the same.  The total compressor work is:"
w_comp_total = Nstages*w_comp
"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"

"The heat added in the external heat exchanger + the reheat between turbines is"
q_in_total = C_P*(T_6 - T_5) +(Nstages - 1)*C_P*(T_8 - T_7)
"Reheat is assumed to occur until:"
T_8 = T_6
"Turbine analysis"
P_7 = P_6 /R_p
"T_7s is the isentropic value of T_7 at turbine exit"
T_7s = T_6*(1/R_p)^((k-1)/k)
"Turbine adiabatic efficiency, w_turbisen > w_turb"
Eta_t = w_turb /w_turbisen
"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T_6 -  T_7s)

"Actual Turbine analysis:"
w_turb = C_P*(T_6 -  T_7)
w_turb_total = Nstages*w_turb

"Cycle analysis"
w_net=w_turb_total-w_comp_total
```

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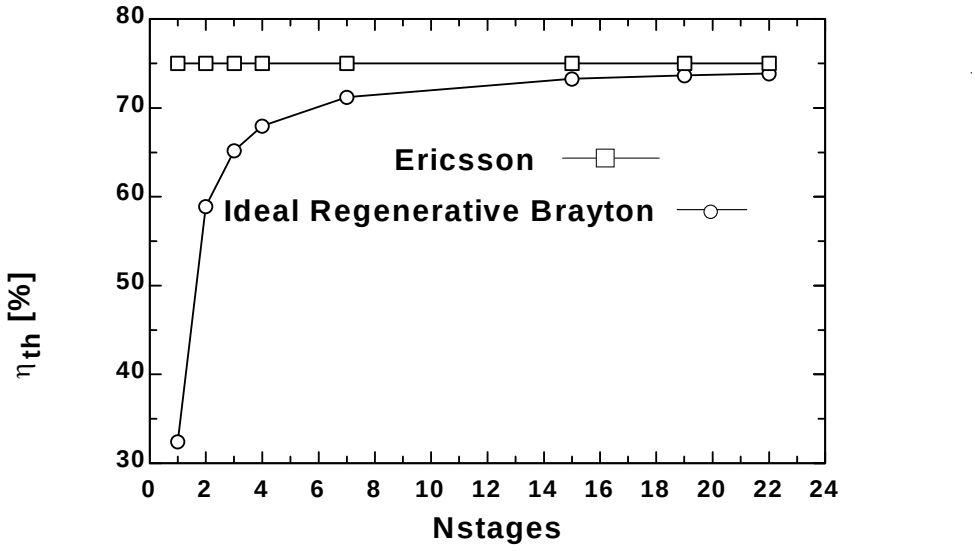
Bwr=w_comp/w_turb "Back work ratio"

P_4=P_2
P_5=P_4
P_6=P_5
T_4 = T_2
"The regenerator effectiveness gives T_5 as:"
Eta_reg = (T_5 - T_4)/(T_9 - T_4)
T_9 = T_7
"Energy balance on regenerator gives T_10 as:"
T_4 + T_9=T_5 + T_10

"Cycle thermal efficiency with regenerator"
Eta_th_regenerative=w_net/q_in_total*Convert(, %) "[%]"

"The efficiency of the Ericsson cycle is the same as the Carnot cycle operating between the same max and min temperatures, T_6 and T_1 for this problem."
Eta_th_Ericsson = (1 - T_1/T_6)*Convert(, %) "[%]"

$\eta_{th,Ericksson}$ [%]	$\eta_{th,Regenerative}$ [%]	Nstages
75	32.43	1
75	58.9	2
75	65.18	3
75	67.95	4
75	71.18	7
75	73.29	15
75	73.66	19
75	73.84	22



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Fundamentals of Engineering (FE) Exam Problems

9-196 An Otto cycle with air as the working fluid has a compression ratio of 10.4. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 10% (b) 39% (c) 61% (d) 79% (e) 82%

Answer (c) 61%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
r=10.4
k=1.4
Eta_Otto=1-1/r^(k-1)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1/r "Taking efficiency to be 1/r"
W2_Eta = 1/r^(k-1)  "Using incorrect relation"
W3_Eta =  1-1/r^(k1-1); k1=1.667 "Using wrong k value"
```

9-197 For specified limits for the maximum and minimum temperatures, the ideal cycle with the lowest thermal efficiency is

(a) Carnot (b) Stirling (c) Ericsson (d) Otto (e) All are the same

Answer (d) Otto

9-198 A Carnot cycle operates between the temperatures limits of 300 K and 2000 K, and produces 600 kW of net power. The rate of entropy change of the working fluid during the heat addition process is

(a) 0 (b) 0.300 kW/K (c) 0.353 kW/K (d) 0.261 kW/K (e) 2.0 kW/K

Answer (c) 0.353 kW/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=300 "K"
TH=2000 "K"
Wnet=600 "kJ/s"
Wnet= (TH-TL)*DS

"Some Wrong Solutions with Common Mistakes:"
W1_DS = Wnet/TH "Using TH instead of TH-TL"
W2_DS = Wnet/TL "Using TL instead of TH-TL"
W3_DS =  Wnet/(TH+TL) "Using TH+TL  instead of TH-TL"
```

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9-199 Air in an ideal Diesel cycle is compressed from 2 L to 0.13 L, and then it expands during the constant pressure heat addition process to 0.30 L. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 41% (b) 59% (c) 66% (d) 70% (e) 78%

Answer (b) 59%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V1=2 "L"
V2= 0.13 "L"
V3= 0.30 "L"
r=V1/V2
rc=V3/V2
k=1.4
Eta_Diesel=1-(1/r^(k-1))*(rc^k-1)/k/(rc-1)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1-(1/r1^(k-1))*(rc^k-1)/k/(rc-1); r1=V1/V3 "Wrong r value"
W2_Eta = 1-Eta_Diesel  "Using incorrect relation"
W3_Eta =  1-(1/r^(k1-1))*(rc^k1-1)/k1/(rc-1); k1=1.667 "Using wrong k value"
W4_Eta = 1-1/r^(k-1) "Using Otto cycle efficiency"
```

9-200 Helium gas in an ideal Otto cycle is compressed from 20°C and 2.5 L to 0.25 L, and its temperature increases by an additional 700°C during the heat addition process. The temperature of helium before the expansion process is

(a) 1790°C (b) 2060°C (c) 1240°C (d) 620°C (e) 820°C

Answer (a) 1790°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
V1=2.5
V2=0.25
r=V1/V2
T1=20+273 "K"
T2=T1*r^(k-1)
T3=T2+700-273 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T3 =T22+700-273; T22=T1*r^(k1-1); k1=1.4 "Using wrong k value"
W2_T3 = T3+273  "Using K instead of C"
W3_T3 = T1+700-273 "Disregarding temp rise during compression"
W4_T3 = T222+700-273; T222=(T1-273)*r^(k-1) "Using C for T1 instead of K"
```

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9-201 In an ideal Otto cycle, air is compressed from 1.20 kg/m^3 and 2.2 L to 0.26 L, and the net work output of the cycle is 440 kJ/kg. The mean effective pressure (MEP) for this cycle is

(a) 612 kPa (b) 599 kPa (c) 528 kPa (d) 416 kPa (e) 367 kPa

Answer (b) 599 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho1=1.20 "kg/m^3"
k=1.4
V1=2.2
V2=0.26
m=rho1*V1/1000 "kg"
w_net=440 "kJ/kg"
Wtotal=m*w_net
MEP=Wtotal/((V1-V2)/1000)

"Some Wrong Solutions with Common Mistakes:"
W1_MEP = w_net/((V1-V2)/1000) "Disregarding mass"
W2_MEP = Wtotal/(V1/1000)  "Using V1 instead of V1-V2"
W3_MEP = (rho1*V2/1000)*w_net/((V1-V2)/1000);  "Finding mass using V2 instead of V1"
W4_MEP = Wtotal/((V1+V2)/1000) "Adding V1 and V2 instead of subtracting"
```

9-202 In an ideal Brayton cycle, air is compressed from 95 kPa and 25°C to 1100 kPa. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 45% (b) 50% (c) 62% (d) 73% (e) 86%

Answer (b) 50%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=95 "kPa"
P2=1100 "kPa"
T1=25+273 "K"
rp=P2/P1
k=1.4
Eta_Brayton=1-1/rp^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1/rp "Taking efficiency to be 1/rp"
W2_Eta = 1/rp^((k-1)/k)  "Using incorrect relation"
W3_Eta = 1-1/rp^((k1-1)/k1); k1=1.667 "Using wrong k value"
```

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9-203 Consider an ideal Brayton cycle executed between the pressure limits of 1200 kPa and 100 kPa and temperature limits of 20°C and 1000°C with argon as the working fluid. The net work output of the cycle is

(a) 68 kJ/kg (b) 93 kJ/kg (c) 158 kJ/kg (d) 186 kJ/kg (e) 310 kJ/kg

Answer (c) 158 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 "kPa"
P2=1200 "kPa"
T1=20+273 "K"
T3=1000+273 "K"
rp=P2/P1
k=1.667
Cp=0.5203 "kJ/kg.K"
Cv=0.3122 "kJ/kg.K"
T2=T1*rp^((k-1)/k)
q_in=Cp*(T3-T2)
Eta_Brayton=1-1/rp^((k-1)/k)
w_net=Eta_Brayton*q_in

"Some Wrong Solutions with Common Mistakes:"
W1_wnet = (1-1/rp^((k-1)/k))*qin1;  qin1=Cv*(T3-T2) "Using Cv instead of Cp"
W2_wnet = (1-1/rp^((k-1)/k))*qin2;  qin2=1.005*(T3-T2)   "Using Cp of air instead of argon"
W3_wnet = (1-1/rp^((k1-1)/k1))*Cp*(T3-T22); T22=T1*rp^((k1-1)/k1); k1=1.4  "Using k of air instead of argon"
W4_wnet = (1-1/rp^((k-1)/k))*Cp*(T3-T222); T222=(T1-273)*rp^((k-1)/k) "Using C for T1 instead of K"
```

9-204 An ideal Brayton cycle has a net work output of 150 kJ/kg and a backwork ratio of 0.4. If both the turbine and the compressor had an isentropic efficiency of 85%, the net work output of the cycle would be

(a) 74 kJ/kg (b) 95 kJ/kg (c) 109 kJ/kg (d) 128 kJ/kg (e) 177 kJ/kg

Answer (b) 95 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
wcomp/wturb=0.4
wturb-wcomp=150 "kJ/kg"
Eff=0.85
w_net=Eff*wturb-wcomp/Eff

"Some Wrong Solutions with Common Mistakes:"
W1_wnet = Eff*wturb-wcomp*Eff "Making a mistake in Wnet relation"
W2_wnet = (wturb-wcomp)/Eff "Using a wrong relation"
W3_wnet = wturb/eff-wcomp*Eff "Using a wrong relation"
```

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9-205 In an ideal Brayton cycle, air is compressed from 100 kPa and 25°C to 1 MPa, and then heated to 927°C before entering the turbine. Under cold air standard conditions, the air temperature at the turbine exit is

(a) 349°C (b) 426°C (c) 622°C (d) 733°C (e) 825°C

Answer (a) 349°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 "kPa"
P2=1000 "kPa"
T1=25+273 "K"
T3=900+273 "K"
rp=P2/P1
k=1.4
T4=T3*(1/rp)^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T4 = T3/rp "Using wrong relation"
W2_T4 = (T3-273)/rp "Using wrong relation"
W3_T4 = T4+273  "Using K instead of C"
W4_T4 = T1+800-273 "Disregarding temp rise during compression"
```

9-206 In an ideal Brayton cycle with regeneration, argon gas is compressed from 100 kPa and 25°C to 400 kPa, and then heated to 1200°C before entering the turbine. The highest temperature that argon can be heated in the regenerator is

(a) 246°C (b) 846°C (c) 689°C (d) 368°C (e) 573°C

Answer (e) 573°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=0.5203 "kJ/kg.K"
P1=100 "kPa"
P2=400 "kPa"
T1=25+273 "K"
T3=1200+273 "K"
"The highest temperature that argon can be heated in the regenerator is the turbine exit temperature,"
rp=P2/P1
T2=T1*rp^((k-1)/k)
T4=T3/rp^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T4 = T3/rp "Using wrong relation"
W2_T4 = (T3-273)/rp^((k-1)/k) "Using C instead of K for T3"
W3_T4 = T4+273  "Using K instead of C"
W4_T4 = T2-273 "Taking compressor exit temp as the answer"
```

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9-207 In an ideal Brayton cycle with regeneration, air is compressed from 80 kPa and 10°C to 400 kPa and 175°C, is heated to 450°C in the regenerator, and then further heated to 1000°C before entering the turbine. Under cold air standard conditions, the effectiveness of the regenerator is

(a) 33% (b) 44% (c) 62% (d) 77% (e) 89%

Answer (d) 77%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
P1=80 "kPa"
P2=400 "kPa"
T1=10+273 "K"
T2=175+273 "K"
T3=1000+273 "K"
T5=450+273 "K"
"The highest temperature that the gas can be heated in the regenerator is the turbine exit temperature,"
rp=P2/P1
T2check=T1*rp^((k-1)/k) "Checking the given value of T2. It checks."
T4=T3/rp^((k-1)/k)
Effective=(T5-T2)/(T4-T2)

"Some Wrong Solutions with Common Mistakes:"
W1_eff = (T5-T2)/(T3-T2) "Using wrong relation"
W2_eff = (T5-T2)/(T44-T2); T44=(T3-273)/rp^((k-1)/k) "Using C instead of K for T3"
W3_eff = (T5-T2)/(T444-T2);  T444=T3/rp "Using wrong relation for T4"
```

9-208 Consider a gas turbine that has a pressure ratio of 6 and operates on the Brayton cycle with regeneration between the temperature limits of 20°C and 900°C. If the specific heat ratio of the working fluid is 1.3, the highest thermal efficiency this gas turbine can have is

(a) 38% (b) 46% (c) 62% (d) 58% (e) 97%

Answer (c) 62%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.3
rp=6
T1=20+273 "K"
T3=900+273 "K"
Eta_regen=1-(T1/T3)*rp^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1-((T1-273)/(T3-273))*rp^((k-1)/k) "Using C for temperatures instead of K"
W2_Eta = (T1/T3)*rp^((k-1)/k)  "Using incorrect relation"
W3_Eta = 1-(T1/T3)*rp^((k1-1)/k1); k1=1.4 "Using wrong k value (the one for air)"
```

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9-209 An ideal gas turbine cycle with many stages of compression and expansion and a regenerator of 100 percent effectiveness has an overall pressure ratio of 10. Air enters every stage of compressor at 290 K, and every stage of turbine at 1200 K. The thermal efficiency of this gas-turbine cycle is

(a) 36% (b) 40% (c) 52% (d) 64% (e) 76%

Answer (e) 76%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
rp=10
T1=290 "K"
T3=1200 "K"
Eff=1-T1/T3

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 100
W2_Eta = 1-1/rp^((k-1)/k) "Using incorrect relation"
W3_Eta = 1-(T1/T3)*rp^((k-1)/k) "Using wrong relation"
W4_Eta = T1/T3 "Using wrong relation"
```

9-210 Air enters a turbojet engine at 320 m/s at a rate of 30 kg/s, and exits at 650 m/s relative to the aircraft. The thrust developed by the engine is

(a) 5 kN (b) 10 kN (c) 15 kN (d) 20 kN (e) 26 kN

Answer (b) 10 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel1=320 "m/s"
Vel2=650 "m/s"
Thrust=m*(Vel2-Vel1)/1000 "kN"
m= 30 "kg/s"

"Some Wrong Solutions with Common Mistakes:"
W1_thrust = (Vel2-Vel1)/1000 "Disregarding mass flow rate"
W2_thrust = m*Vel2/1000 "Using incorrect relation"
```

9-211 ··· 9-219 Design and Essay Problems.

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 10
VAPOR AND COMBINED POWER CYCLES

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Carnot Vapor Cycle

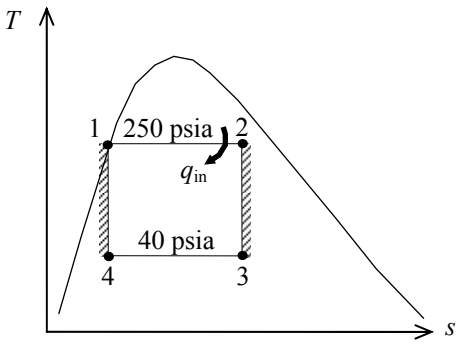
10-1C The Carnot cycle is not a realistic model for steam power plants because (1) limiting the heat transfer processes to two-phase systems to maintain isothermal conditions severely limits the maximum temperature that can be used in the cycle, (2) the turbine will have to handle steam with a high moisture content which causes erosion, and (3) it is not practical to design a compressor that will handle two phases.

10-2E A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the quality at the end of the heat rejection process, and the net work output are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) We note that

$$T_H = T_{\text{sat@250 psia}} = 401°\text{F} = 861\ \text{R}$$
$$T_L = T_{\text{sat@40psia}} = 267.2°\text{F} = 727.2\ \text{R}$$



and

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{727.2\ \text{R}}{861\ \text{R}} = 0.1553 = \mathbf{15.5\%}$$

(*b*) Noting that $s_4 = s_1 = s_{f\,@\,250\ \text{psia}} = 0.56784$ Btu/lbm·R,

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{0.56784 - 0.3921}{1.2845} = \mathbf{0.137}$$

(*c*) The enthalpies before and after the heat addition process are

$$h_1 = h_{f@250\ \text{psia}} = 376.09\ \text{Btu/lbm}$$
$$h_2 = h_f + x_2 h_{fg} = 376.09 + (0.95)(825.47) = 1160.3\ \text{Btu/lbm}$$

Thus,

$$q_{\text{in}} = h_2 - h_1 = 1160.3 - 376.09 = 784.2\ \text{Btu/lbm}$$

and

$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.1553)(784.2\ \text{Btu/lbm}) = \mathbf{122\ Btu/lbm}$$

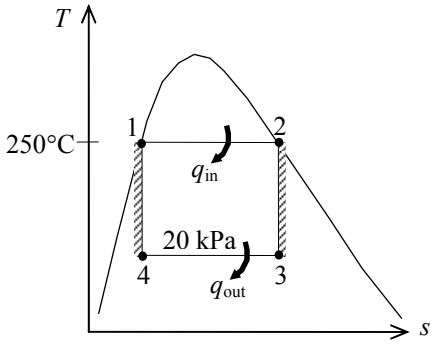
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10-3 A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the amount of heat rejected, and the net work output are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) Noting that T_H = 250°C = 523 K and $T_L = T_{\text{sat @ 20 kPa}}$ = 60.06°C = 333.1 K, the thermal efficiency becomes

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{333.1\ \text{K}}{523\ \text{K}} = 0.3632 = \mathbf{36.3\%}$$



(*b*) The heat supplied during this cycle is simply the enthalpy of vaporization,

$$q_{\text{in}} = h_{fg@\ 250^\circ C} = 1715.3\ \text{kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{333.1\ \text{K}}{523\ \text{K}}\right)(1715.3\ \text{kJ/kg}) = \mathbf{1092.3\ kJ/kg}$$

(*c*) The net work output of this cycle is

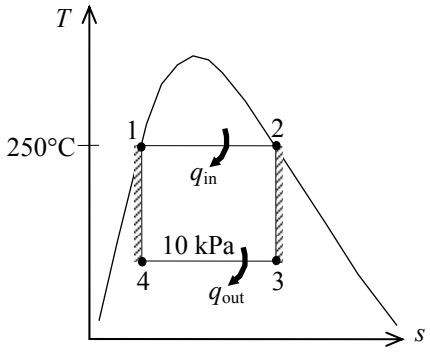
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.3632)(1715.3\ \text{kJ/kg}) = \mathbf{623.0\ kJ/kg}$$

10-4 A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the amount of heat rejected, and the net work output are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) Noting that T_H = 250°C = 523 K and $T_L = T_{\text{sat @ 10 kPa}}$ = 45.81°C = 318.8 K, the thermal efficiency becomes

$$\eta_{\text{th, C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{318.8\ \text{K}}{523\ \text{K}} = \mathbf{39.04\%}$$



(*b*) The heat supplied during this cycle is simply the enthalpy of vaporization,

$$q_{\text{in}} = h_{fg@\ 250^\circ C} = 1715.3\ \text{kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{318.8\ \text{K}}{523\ \text{K}}\right)(1715.3\ \text{kJ/kg}) = \mathbf{1045.6\ kJ/kg}$$

(*c*) The net work output of this cycle is

$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.3904)(1715.3\ \text{kJ/kg}) = \mathbf{669.7\ kJ/kg}$$

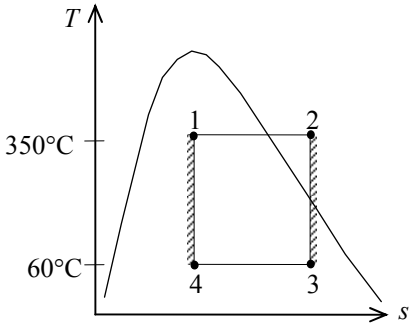
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10-5 A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the pressure at the turbine inlet, and the net work output are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The thermal efficiency is determined from

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{60 + 273 \text{ K}}{350 + 273 \text{ K}} = \mathbf{46.5\%}$$



(*b*) Note that

$$s_2 = s_3 = s_f + x_3 s_{fg}$$
$$= 0.8313 + 0.891 \times 7.0769 = 7.1368 \text{ kJ/kg·K}$$

Thus,

$$\left.\begin{aligned} T_2 &= 350°\text{C} \\ s_2 &= 7.1368 \text{ kJ/kg} \cdot \text{K} \end{aligned}\right\} P_2 \cong \mathbf{1.40\ MPa} \text{ (Table A-6)}$$

(*c*) The net work can be determined by calculating the enclosed area on the *T*-*s* diagram,

$$s_4 = s_f + x_4 s_{fg} = 0.8313 + (0.1)(7.0769) = 1.5390 \text{ kJ/kg} \cdot \text{K}$$

Thus,

$$w_{\text{net}} = \text{Area} = (T_H - T_L)(s_3 - s_4) = (350 - 60)(7.1368 - 1.5390) = \mathbf{1623\ kJ/kg}$$

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The Simple Rankine Cycle

10-6C The four processes that make up the simple ideal cycle are (1) Isentropic compression in a pump, (2) P = constant heat addition in a boiler, (3) Isentropic expansion in a turbine, and (4) P = constant heat rejection in a condenser.

10-7C Heat rejected decreases; everything else increases.

10-8C Heat rejected decreases; everything else increases.

10-9C The pump work remains the same, the moisture content decreases, everything else increases.

10-10C The actual vapor power cycles differ from the idealized ones in that the actual cycles involve friction and pressure drops in various components and the piping, and heat loss to the surrounding medium from these components and piping.

10-11C The boiler exit pressure will be (a) lower than the boiler inlet pressure in actual cycles, and (b) the same as the boiler inlet pressure in ideal cycles.

10-12C We would reject this proposal because $w_{turb} = h_1 - h_2 - q_{out}$, and any heat loss from the steam will adversely affect the turbine work output.

10-13C Yes, because the saturation temperature of steam at 10 kPa is 45.81°C, which is much higher than the temperature of the cooling water.

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10-14 A simple ideal Rankine cycle with R-134a as the working fluid operates between the specified pressure limits. The mass flow rate of R-134a for a given power production and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

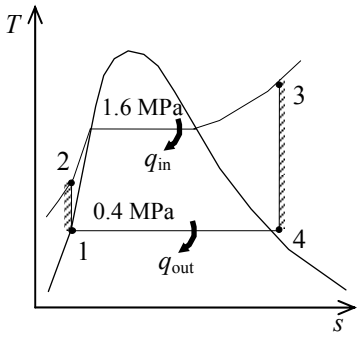
Analysis From the refrigerant tables (Tables A-11, A-12, and A-13),

$$h_1 = h_{f@\,0.4\,\text{MPa}} = 63.94\ \text{kJ/kg}$$
$$\nu_1 = \nu_{f@\,0.4\,\text{MPa}} = 0.0007907\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.0007907\ \text{m}^3/\text{kg})(1600-400)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 0.95\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 63.94 + 0.95 = 64.89\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 1.6\ \text{MPa} \\ T_3 &= 80°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 305.07\ \text{kJ/kg} \\ s_3 &= 0.9875\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 0.4\ \text{MPa} \\ s_4 &= s_3 \end{aligned}\right\} \quad h_4 = 273.21\ \text{kJ/kg}$$



Thus,

$$q_{\text{in}} = h_3 - h_2 = 305.07 - 64.89 = 240.18\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 273.21 - 63.94 = 209.27\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 240.18 - 209.27 = 30.91\ \text{kJ/kg}$$

The mass flow rate of the refrigerant and the thermal efficiency of the cycle are then

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{750\ \text{kJ/s}}{30.91\ \text{kJ/kg}} = \mathbf{24.26\ kg/s}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{209.27}{240.18} = \mathbf{0.129}$$

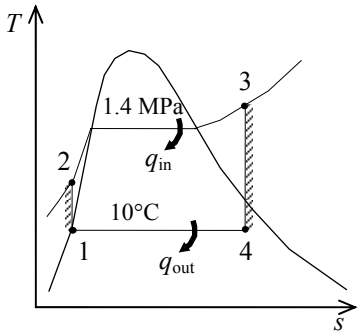
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10-15 A simple ideal Rankine cycle with R-134a as the working fluid is considered. The turbine inlet temperature, the cycle thermal efficiency, and the back-work ratio of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the refrigerant tables (Tables A-11, A-12, and A-13),

$$P_1 = P_{\text{sat @ 10°C}} = 414.89\ \text{kPa}$$
$$h_1 = h_{f\,@\,10°\text{C}} = 65.43\ \text{kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,10°\text{C}} = 0.0007930\ \text{m}^3/\text{kg}$$



$$w_{\text{p,in}} = \nu_1 (P_2 - P_1)$$
$$= (0.0007930\ \text{m}^3/\text{kg})(1400 - 414.89)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 0.78\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 65.43 + 0.78 = 66.21\ \text{kJ/kg}$$

$$\left.\begin{array}{l} T_4 = 10°\text{C} \\ x_4 = 0.98 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 65.43 + (0.98)(190.73) = 252.35\ \text{kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 0.25286 + (0.98)(0.67356) = 0.91295\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 1400\ \text{kPa} \\ s_3 = s_4 = 0.91295\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} h_3 = 276.91\ \text{kJ/kg} \\ T_3 = \mathbf{53.0°C} \end{array}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 276.91 - 66.21 = 210.70\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 252.35 - 65.43 = 186.92\ \text{kJ/kg}$$

The thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{186.92}{210.70} = \mathbf{0.113}$$

The back-work ratio is determined from

$$r_{bw} = \frac{w_{\text{P,in}}}{W_{\text{T,out}}} = \frac{w_{\text{P,in}}}{h_3 - h_4} = \frac{0.78\ \text{kJ/kg}}{(276.91 - 252.35)\ \text{kJ/kg}} = \mathbf{0.0318}$$

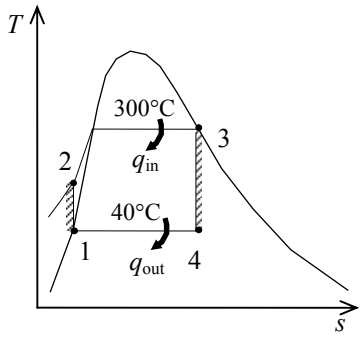
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10-16 A simple ideal Rankine cycle with water as the working fluid is considered. The work output from the turbine, the heat addition in the boiler, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$P_1 = P_{\text{sat @ 40°C}} = 7.385\ \text{kPa}$$
$$P_2 = P_{\text{sat @ 300°C}} = 8588\ \text{kPa}$$
$$h_1 = h_{f\,@\,40°\text{C}} = 167.53\ \text{kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,40°\text{C}} = 0.001008\ \text{m}^3/\text{kg}$$



$$w_{\text{p,in}} = \nu_1 (P_2 - P_1)$$
$$= (0.001008\ \text{m}^3/\text{kg})(8588 - 7.385)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 8.65\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 167.53 + 8.65 = 176.18\ \text{kJ/kg}$$

$$\left.\begin{array}{l} T_3 = 300°\text{C} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2749.6\ \text{kJ/kg} \\ s_3 = 5.7059\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_4 = 40°\text{C} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{5.7059 - 0.5724}{7.6832} = 0.6681 \\ h_4 = h_f + x_4 h_{fg} = 167.53 + (0.6681)(2406.0) = 1775.1\ \text{kJ/kg} \end{array}$$

Thus,

$$w_{\text{T,out}} = h_3 - h_4 = 2749.6 - 1775.1 = \mathbf{974.5\ kJ/kg}$$
$$q_{\text{in}} = h_3 - h_2 = 2749.6 - 176.18 = \mathbf{2573.4\ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 1775.1 - 167.53 = 1607.6\ \text{kJ/kg}$$

The thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1607.6}{2573.4} = \mathbf{0.375}$$

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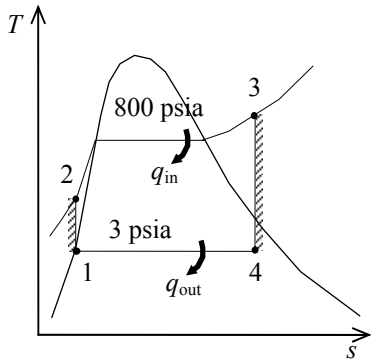
10-17E A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The rates of heat addition and rejection, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,3\,\text{psia}} = 109.40\ \text{Btu/lbm}$$
$$\nu_1 = \nu_{f@\,3\,\text{psia}} = 0.01630\ \text{ft}^3/\text{lbm}$$

$$\begin{aligned} w_{\text{p,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.01630\ \text{ft}^3/\text{lbm})(800-3)\text{psia}\left(\frac{1\,\text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &= 2.40\ \text{Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 109.40 + 2.40 = 111.81\ \text{Btu/lbm}$$



$$\left.\begin{aligned} P_3 &= 800\ \text{psia} \\ T_3 &= 900°\text{F} \end{aligned}\right\} \begin{aligned} h_3 &= 1456.0\ \text{Btu/lbm} \\ s_3 &= 1.6413\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 3\ \text{psia} \\ s_4 &= s_3 \end{aligned}\right\} \begin{aligned} x_4 &= \frac{s_4 - s_f}{s_{fg}} = \frac{1.6413 - 0.2009}{1.6849} = 0.8549 \\ h_4 &= h_f + x_4 h_{fg} = 109.40 + (0.8549)(1012.8) = 975.24\ \text{Btu/lbm} \end{aligned}$$

Knowing the power output from the turbine the mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) \longrightarrow \dot{m} = \frac{\dot{W}_{\text{T,out}}}{h_3 - h_4} = \frac{1750\ \text{kJ/s}}{(1456.0 - 975.24)\text{Btu/lbm}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = 3.450\ \text{lbm/s}$$

The rates of heat addition and rejection are

$$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (3.450\ \text{lbm/s})(1456.0 - 111.81)\text{Btu/lbm} = \mathbf{4637\ Btu/s}$$
$$\dot{Q}_{\text{out}} = \dot{m}(h_4 - h_1) = (3.450\ \text{lbm/s})(975.24 - 109.40)\text{Btu/lbm} = \mathbf{2987\ Btu/s}$$

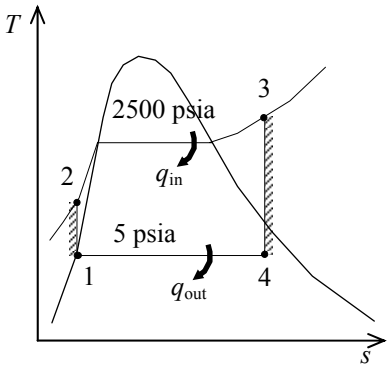
and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{2987}{4637} = 0.3559 = \mathbf{35.6\%}$$

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10-18E A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The turbine inlet temperature and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,5\,\text{psia}} = 130.18\ \text{Btu/lbm}$$
$$\nu_1 = \nu_{f@\,5\,\text{psia}} = 0.01641\ \text{ft}^3/\text{lbm}$$

$$\begin{aligned} w_{\text{p,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.01641\ \text{ft}^3/\text{lbm})(2500-5)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &= 7.58\ \text{Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 130.18 + 7.58 = 137.76\ \text{Btu/lbm}$$

$$\left.\begin{aligned} P_4 &= 5\ \text{psia} \\ x_4 &= 0.80 \end{aligned}\right\} \begin{aligned} h_4 &= h_f + x_4 h_{fg} = 130.18 + (0.80)(1000.5) = 930.58\ \text{Btu/lbm} \\ s_4 &= s_f + x_4 s_{fg} = 0.23488 + (0.80)(1.60894) = 1.52203\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\left.\begin{aligned} P_3 &= 2500\ \text{psia} \\ s_3 &= s_4 = 1.52203\ \text{Btu/lbm}\cdot\text{R} \end{aligned}\right\} \begin{aligned} h_3 &= 1450.8\ \text{Btu/lbm} \\ T_3 &= \mathbf{989.2°F} \end{aligned}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 1450.8 - 137.76 = 1313.0\ \text{Btu/lbm}$$
$$q_{\text{out}} = h_4 - h_1 = 930.58 - 130.18 = 800.4\ \text{Btu/lbm}$$

The thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{800.4}{1313.0} = \mathbf{0.390}$$

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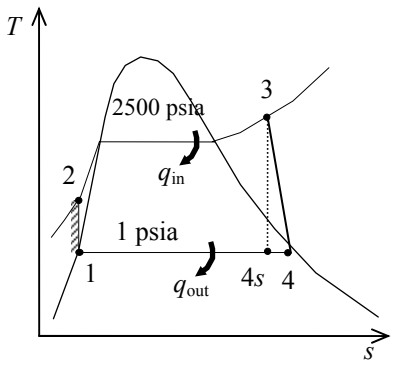
10-19E A simple steam Rankine cycle operates between the specified pressure limits. The mass flow rate, the power produced by the turbine, the rate of heat addition, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,1\,\text{psia}} = 69.72\ \text{Btu/lbm}$$
$$\nu_1 = \nu_{f@\,6\,\text{psia}} = 0.01614\ \text{ft}^3/\text{lbm}$$

$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.01614\ \text{ft}^3/\text{lbm})(2500-1)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right)$$
$$= 7.46\ \text{Btu/lbm}$$
$$h_2 = h_1 + w_{\text{p,in}} = 69.72 + 7.46 = 77.18\ \text{Btu/lbm}$$



$$\left.\begin{array}{l} P_3 = 2500\ \text{psia} \\ T_3 = 800°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1302.0\ \text{Btu/lbm} \\ s_3 = 1.4116\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 1\ \text{psia} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.4116 - 0.13262}{1.84495} = 0.6932 \\ h_{4s} = h_f + x_{4s}h_{fg} = 69.72 + (0.6932)(1035.7) = 787.70\ \text{Btu/lbm} \end{array}$$

$$\eta_{\text{T}} = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_{\text{T}}(h_3 - h_{4\text{s}}) = 1302.0 - (0.90)(1302.0 - 787.70) = 839.13\ \text{kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 1302.0 - 77.18 = 1224.8\ \text{Btu/lbm}$$
$$q_{\text{out}} = h_4 - h_1 = 839.13 - 69.72 = 769.41\ \text{Btu/lbm}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1224.8 - 769.41 = 455.39\ \text{Btu/lbm}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{1000\ \text{kJ/s}}{455.39\ \text{Btu/lbm}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = \mathbf{2.081\ lbm/s}$$

The power output from the turbine and the rate of heat addition are

$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) = (2.081\ \text{lbm/s})(1302.0 - 839.13)\text{Btu/lbm}\left(\frac{1\ \text{kJ}}{0.94782\ \text{Btu}}\right) = \mathbf{1016\ kW}$$
$$\dot{Q}_{\text{in}} = \dot{m}q_{\text{in}} = (2.081\ \text{lbm/s})(1224.8\ \text{Btu/lbm}) = \mathbf{2549\ Btu/s}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1000\ \text{kJ/s}}{2549\ \text{Btu/s}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = \mathbf{0.3718}$$

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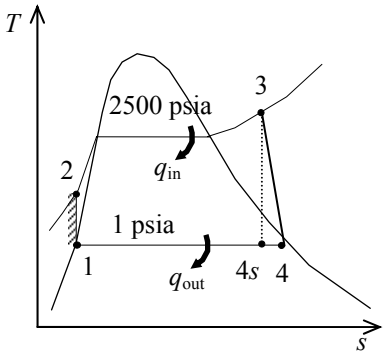
10-20E A simple steam Rankine cycle operates between the specified pressure limits. The mass flow rate, the power produced by the turbine, the rate of heat addition, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,1\,\text{psia}} = 69.72\ \text{Btu/lbm}$$
$$\nu_1 = \nu_{f@\,6\,\text{psia}} = 0.01614\ \text{ft}^3/\text{lbm}$$

$$\begin{aligned} w_{\text{p,in}} &= \nu_1(P_2 - P_1) \\ &= (0.01614\ \text{ft}^3/\text{lbm})(2500-1)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &= 7.46\ \text{Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 69.72 + 7.46 = 77.18\ \text{Btu/lbm}$$



$$\left.\begin{aligned} P_3 &= 2500\ \text{psia} \\ T_3 &= 800°\text{F} \end{aligned}\right\} \begin{aligned} h_3 &= 1302.0\ \text{Btu/lbm} \\ s_3 &= 1.4116\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 1\ \text{psia} \\ s_4 &= s_3 \end{aligned}\right\} \begin{aligned} x_{4s} &= \frac{s_4 - s_f}{s_{fg}} = \frac{1.4116 - 0.13262}{1.84495} = 0.6932 \\ h_{4s} &= h_f + x_{4s}h_{fg} = 69.72 + (0.6932)(1035.7) = 787.70\ \text{Btu/lbm} \end{aligned}$$

$$\eta_{\text{T}} = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_{\text{T}}(h_3 - h_{4\text{s}}) = 1302.0 - (0.90)(1302.0 - 787.70) = 839.13\ \text{kJ/kg}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m}(h_3 - h_4) \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{h_3 - h_4} = \frac{1000\ \text{kJ/s}}{(1302.0 - 839.13)\ \text{Btu/lbm}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = 2.048\ \text{lbm/s}$$

The rate of heat addition is

$$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (2.048\ \text{lbm/s})(1302.0 - 77.18)\text{Btu/lbm}\left(\frac{1\ \text{kJ}}{0.94782\ \text{Btu}}\right) = 2508\ \text{Btu/s}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1000\ \text{kJ/s}}{2508\ \text{Btu/s}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = 0.3779$$

The thermal efficiency in the previous problem was determined to be 0.3718. The error in the thermal efficiency caused by neglecting the pump work is then

$$\text{Error} = \frac{0.3779 - 0.3718}{0.3718} \times 100 = \mathbf{1.64\%}$$

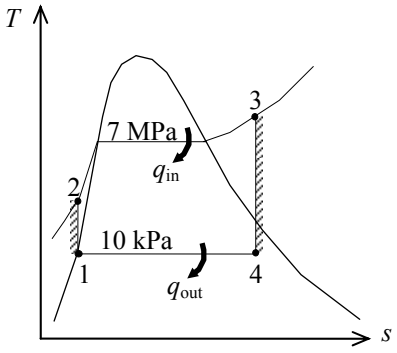
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10-21 A steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle, the mass flow rate of the steam, and the temperature rise of the cooling water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\nu_1 = \nu_{f@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$
$$w_{p,\text{in}} = \nu_1(P_2 - P_1)$$
$$= (0.00101\ \text{m}^3/\text{kg})(7{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 7.06\text{ kJ/kg}$$



$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 7.06 = 198.87\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 7\text{ MPa} \\ T_3 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3411.4\text{ kJ/kg} \\ s_3 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{matrix}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$

$$h_4 = h_f + x_4 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6\text{ kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3411.4 - 198.87 = 3212.5\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2153.6 - 191.81 = 1961.8\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3212.5 - 1961.8 = 1250.7\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1250.7\text{ kJ/kg}}{3212.5\text{ kJ/kg}} = \mathbf{38.9\%}$$

(*b*) $$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{ kJ/s}}{1250.7\text{ kJ/kg}} = \mathbf{36.0\ kg/s}$$

(*c*) The rate of heat rejection to the cooling water and its temperature rise are

$$\dot{Q}_{\text{out}} = \dot{m} q_{\text{out}} = (35.98\text{ kg/s})(1961.8\text{ kJ/kg}) = 70{,}586\ \text{kJ/s}$$
$$\Delta T_{\text{cooling water}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}c)_{\text{cooling water}}} = \frac{70{,}586\text{ kJ/s}}{(2000\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})} = \mathbf{8.4°C}$$

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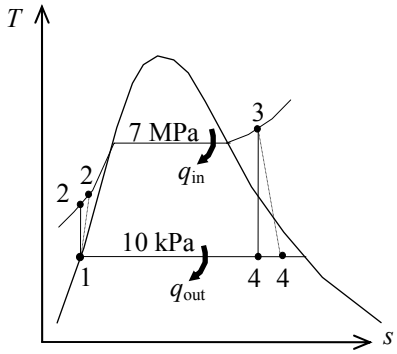
10-22 A steam power plant operates on a simple nonideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle, the mass flow rate of the steam, and the temperature rise of the cooling water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\nu_1 = \nu_{f@\,10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$

$$w_{p,\text{in}} = \nu_1 (P_2 - P_1)/\eta_p$$
$$= (0.00101\text{ m}^3\text{/kg})(7{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.87)$$
$$= 8.11\text{ kJ/kg}$$
$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 8.11 = 199.92\text{ kJ/kg}$$



$$\left.\begin{matrix} P_3 = 7\text{ MPa} \\ T_3 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3411.4\text{ kJ/kg} \\ s_3 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{matrix}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$

$$h_{4s} = h_f + x_4 h_{fg} = 191.81 + (0.820)(2392.1) = 2153.6\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 3411.4 - (0.87)(3411.4 - 2153.6) = 2317.1\text{ kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3411.4 - 199.92 = 3211.5\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2317.1 - 191.81 = 2125.3\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3211.5 - 2125.3 = 1086.2\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1086.2\text{ kJ/kg}}{3211.5\text{ kJ/kg}} = \mathbf{33.8\%}$$

(*b*) $$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{ kJ/s}}{1086.2\text{ kJ/kg}} = \mathbf{41.43\text{ kg/s}}$$

(*c*) The rate of heat rejection to the cooling water and its temperature rise are

$$\dot{Q}_{\text{out}} = \dot{m} q_{\text{out}} = (41.43\text{ kg/s})(2125.3\text{ kJ/kg}) = 88{,}051\text{ kJ/s}$$

$$\Delta T_{\text{coolingwater}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}c)_{\text{coolingwater}}} = \frac{88{,}051\text{ kJ/s}}{(2000\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})} = \mathbf{10.5°C}$$

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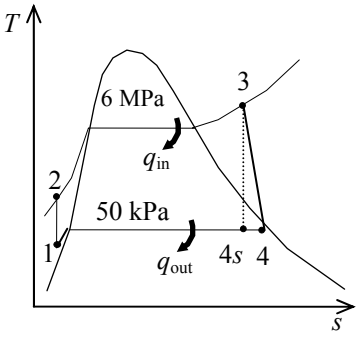
10-23 A simple Rankine cycle with water as the working fluid operates between the specified pressure limits. The rate of heat addition in the boiler, the power input to the pumps, the net power, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$\left.\begin{array}{l} P_1 = 50\ \text{kPa} \\ T_1 = T_{\text{sat}@\,50\,\text{kPa}} - 6.3 = 81.3 - 6.3 = 75°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f\,@\,75°\text{C}} = 314.03\ \text{kJ/kg} \\ \nu_1 = \nu_{f@\,75°\text{C}} = 0.001026\ \text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{p,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.001026\ \text{m}^3/\text{kg})(6000 - 50)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.10\ \text{kJ/kg} \end{aligned}$$



$$h_2 = h_1 + w_{\text{p,in}} = 314.03 + 6.10 = 320.13\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 6000\ \text{kPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3302.9\ \text{kJ/kg} \\ s_3 = 6.7219\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 50\ \text{kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{6.7219 - 1.0912}{6.5019} = 0.8660 \\ h_{4s} = h_f + x_{4s} h_{fg} = 340.54 + (0.8660)(2304.7) = 2336.4\ \text{kJ/kg} \end{array}$$

$$\eta_{\text{T}} = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_{\text{T}}(h_3 - h_{4\text{s}}) = 3302.9 - (0.94)(3302.9 - 2336.4) = 2394.4\ \text{kJ/kg}$$

Thus,

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}(h_3 - h_2) = (20\ \text{kg/s})(3302.9 - 320.13)\text{kJ/kg} = \mathbf{59{,}660\ kW} \\ \dot{W}_{\text{T,out}} &= \dot{m}(h_3 - h_4) = (20\ \text{kg/s})(3302.9 - 2394.4)\text{kJ/kg} = 18{,}170\ \text{kW} \\ \dot{W}_{\text{P,in}} &= \dot{m} w_{\text{P,in}} = (20\ \text{kg/s})(6.10\ \text{kJ/kg}) = \mathbf{122\ kW} \\ \dot{W}_{\text{net}} &= \dot{W}_{\text{T,out}} - \dot{W}_{\text{P,in}} = 18{,}170 - 122 = \mathbf{18{,}050\ kW} \end{aligned}$$

and

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{18{,}050}{59{,}660} = \mathbf{0.3025}$$

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10-24 The change in the thermal efficiency of the cycle in Prob. 10-23 due to a pressure drop in the boiler is to be determined.

Analysis We use the following EES routine to obtain the solution.

```
"Given"
P[2]=6000  [kPa]
DELTAP=50 [kPa]
P[3]=6000-DELTAP [kPa]
T[3]=450 [C]
P[4]=50  [kPa]
Eta_T=0.94
DELTAT_subcool=6.3 [C]
T[1]=temperature(Fluid$, P=P[1], x=x[1])-DELTAT_subcool
m_dot=20 [kg/s]

"Analysis"
Fluid$='steam_iapws'
P[1]=P[4]
x[1]=0
h[1]=enthalpy(Fluid$, P=P[1], T=T[1])
v[1]=volume(Fluid$, P=P[1], T=T[1])
w_p_in=v[1]*(P[2]-P[1])
h[2]=h[1]+w_p_in
h[3]=enthalpy(Fluid$, P=P[3], T=T[3])
s[3]=entropy(Fluid$, P=P[3], T=T[3])
s[4]=s[3]
h_s[4]=enthalpy(Fluid$, P=P[4], s=s[4])
Eta_T=(h[3]-h[4])/(h[3]-h_s[4])
q_in=h[3]-h[2]
q_out=h[4]-h[1]
w_net=q_in-q_out
Eta_th=1-q_out/q_in
```

Solution

DELTAP=50 [kPa]	DELTAT_subcool=6.3 [C]	Eta_T=0.94
Eta_th=**0.3022**	Fluid$='steam_iapws'	h[1]=314.11 [kJ/kg]
h[2]=320.21 [kJ/kg]	h[3]=3303.64 [kJ/kg]	h[4]=2396.01 [kJ/kg]
h_s[4]=2338.1 [kJ/kg]	m_dot=20 [kg/s]	P[1]=50
P[2]=6000	P[3]=5950	P[4]=50
q_in=2983.4 [kJ/kg]	q_out=2081.9 [kJ/kg]	s[3]=6.7265 [kJ/kg-K]
s[4]=6.7265 [kJ/kg-K]	T[1]=75.02 [C]	T[3]=450 [C]
v[1]=0.001026 [m^3/kg]	w_net=901.5 [kJ/kg]	w_p_in=6.104 [kJ/kg]
x[1]=0		

Discussion The thermal efficiency without a pressure drop was obtained to be 0.3025.

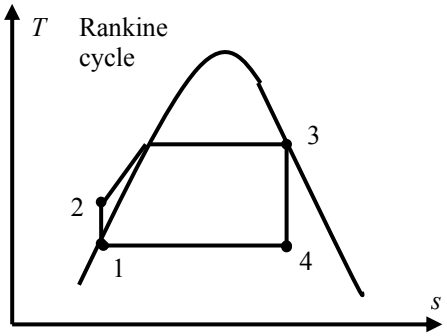
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10-25 The net work outputs and the thermal efficiencies for a Carnot cycle and a simple ideal Rankine cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) Rankine cycle analysis: From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\ 50\ \text{kPa}} = 340.54\ \text{kJ/kg}$$
$$\nu_1 = \nu_{f@\ 50\ \text{kPa}} = 0.001030\ \text{m}^3/\text{kg}$$



$$w_{p,\text{in}} = \nu_1 (P_2 - P_1)$$
$$= (0.001030\ \text{m}^3/\text{kg})(5000 - 50)\ \text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 5.10\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{p,\text{in}} = 340.54 + 5.10 = 345.64\ \text{kJ/kg}$$

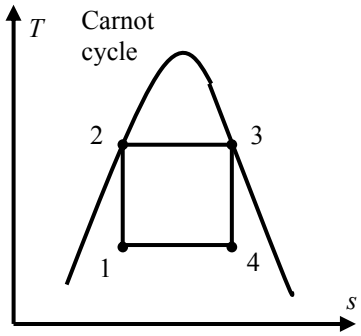
$$\left.\begin{matrix} P_3 = 5\ \text{MPa} \\ x_3 = 1 \end{matrix}\right\} \begin{matrix} h_3 = 2794.2\ \text{kJ/kg} \\ s_3 = 5.9737\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 50\ \text{kPa} \\ s_4 = s_3 \end{matrix}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{5.9737 - 1.09120}{6.5019} = 0.7509$$
$$h_4 = h_f + x_4 h_{fg} = 340.54 + (0.7509)(2304.7)$$
$$= 2071.2\ \text{kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = 2794.2 - 345.64 = 2448.6\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2071.2 - 340.54 = 1730.7\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2448.6 - 1730.7 = \mathbf{717.9\ kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1730.7}{2448.6} = 0.2932 = \mathbf{29.3\%}$$

(*b*) Carnot Cycle analysis:



$$\left.\begin{matrix} P_3 = 5\ \text{MPa} \\ x_3 = 1 \end{matrix}\right\} \begin{matrix} h_3 = 2794.2\ \text{kJ/kg} \\ T_3 = 263.9°\text{C} \end{matrix}$$

$$\left.\begin{matrix} T_2 = T_3 = 263.9°\text{C} \\ x_2 = 0 \end{matrix}\right\} \begin{matrix} h_2 = 1154.5\ \text{kJ/kg} \\ s_2 = 2.9207\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_1 = 50\ \text{kPa} \\ s_1 = s_2 \end{matrix}\right\} \begin{matrix} x_1 = \dfrac{s_1 - s_f}{s_{fg}} = \dfrac{2.9207 - 1.0912}{6.5019} = 0.2814 \\ h_1 = h_f + x_1 h_{fg} \end{matrix}$$
$$= 340.54 + (0.2814)(2304.7) = 989.05\ \text{kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = 2794.2 - 1154.5 = 1639.7\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2071.2 - 340.54 = 1082.2\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1639.7 - 1082.2 = \mathbf{557.5\ kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1082.2}{1639.7} = 0.3400 = \mathbf{34.0\%}$$

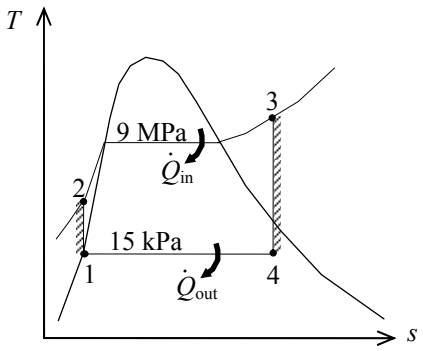
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10-26 A 120-MW coal-fired steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The overall plant efficiency and the required rate of the coal supply are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,15\text{ kPa}} = 225.94\text{ kJ/kg}$$
$$\nu_1 = \nu_{f@\,15\text{ kPa}} = 0.0010140\,\text{m}^3/\text{kg}$$



$$w_{p,\text{in}} = \nu_1(P_2 - P_1)$$
$$= (0.001014\ \text{m}^3/\text{kg})(9000 - 15\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 9.11\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{p,\text{in}} = 225.94 + 9.11 = 235.05\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 9\text{ MPa} \\ T_3 = 550°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3512.0\text{ kJ/kg} \\ s_3 = 6.8164\text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 15\text{ kPa} \\ s_4 = s_3 \end{matrix}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8164 - 0.7549}{7.2522} = 0.8358$$

$$h_4 = h_f + x_4 h_{fg} = 225.94 + (0.8358)(2372.4) = 2208.8\ \text{kJ/kg}$$

The thermal efficiency is determined from

$$q_{\text{in}} = h_3 - h_2 = 3512.0 - 235.05 = 3276.9\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2208.8 - 225.94 = 1982.9\ \text{kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1982.9}{3276.9} = 0.3949$$

Thus,

$$\eta_{\text{overall}} = \eta_{\text{th}} \times \eta_{\text{comb}} \times \eta_{\text{gen}} = (0.3949)(0.75)(0.96) = 0.2843 = \mathbf{28.4\%}$$

(*b*) Then the required rate of coal supply becomes

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{overall}}} = \frac{120{,}000\ \text{kJ/s}}{0.2843} = 422{,}050\ \text{kJ/s}$$

and

$$\dot{m}_{\text{coal}} = \frac{\dot{Q}_{\text{in}}}{C_{\text{coal}}} = \frac{422{,}050\ \text{kJ/s}}{29{,}300\ \text{kJ/kg}} = 14.404\ \text{kg/s} = \mathbf{51.9\ tons/h}$$

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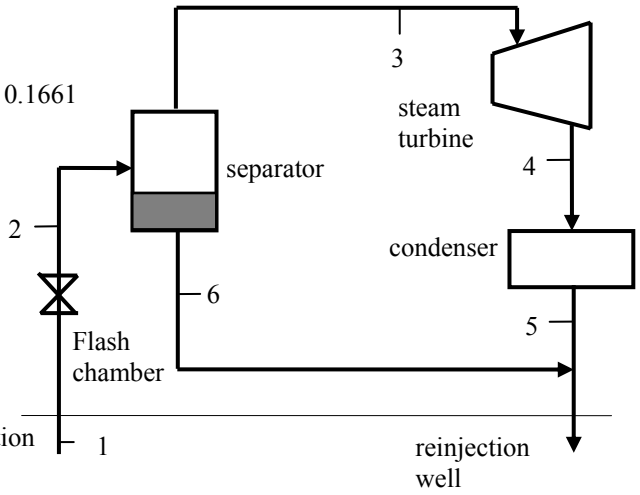
10-27 A single-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The mass flow rate of steam through the turbine, the isentropic efficiency of the turbine, the power output from the turbine, and the thermal efficiency of the plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{matrix} T_1 = 230°\text{C} \\ x_1 = 0 \end{matrix}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{matrix}\right\} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{990.14 - 640.09}{2108} = 0.1661$$



The mass flow rate of steam through the turbine is

$$\dot{m}_3 = x_2 \dot{m}_1 = (0.1661)(230 \text{ kg/s}) = \mathbf{38.20 \text{ kg/s}}$$

(*b*) Turbine:

$$\left.\begin{matrix} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{matrix}\right\} \begin{matrix} h_3 = 2748.1 \text{ kJ/kg} \\ s_3 = 6.8207 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 10 \text{ kPa} \\ s_4 = s_3 \end{matrix}\right\} h_{4s} = 2160.3 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{matrix}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{2748.1 - 2344.7}{2748.1 - 2160.3} = \mathbf{0.686}$$

(*c*) The power output from the turbine is

$$\dot{W}_{\text{T,out}} = \dot{m}_3 (h_3 - h_4) = (38.20 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = \mathbf{15{,}410 \text{ kW}}$$

(*d*) We use saturated liquid state at the standard temperature for dead state enthalpy

$$\left.\begin{matrix} T_0 = 25°\text{C} \\ x_0 = 0 \end{matrix}\right\} h_0 = 104.83 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} = \dot{m}_1 (h_1 - h_0) = (230 \text{ kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{T,out}}}{\dot{E}_{\text{in}}} = \frac{15{,}410}{203{,}622} = 0.0757 = \mathbf{7.6\%}$$

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10-28 A double-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The temperature of the steam at the exit of the second flash chamber, the power produced from the second turbine, and the thermal efficiency of the plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{matrix} T_1 = 230°\text{C} \\ x_1 = 0 \end{matrix}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{matrix}\right\} x_2 = 0.1661$$

$$\dot{m}_3 = x_2 \dot{m}_1 = (0.1661)(230 \text{ kg/s}) = 38.20 \text{ kg/s}$$

$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 0.1661 = 191.80 \text{ kg/s}$$

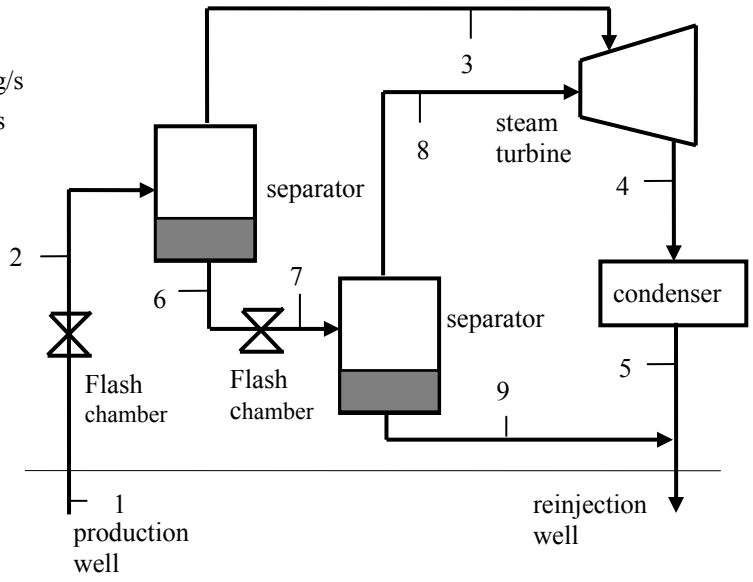
$$\left.\begin{matrix} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{matrix}\right\} h_3 = 2748.1 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{matrix}\right\} h_4 = 2344.7 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{matrix}\right\} h_6 = 640.09 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_7 = 150 \text{ kPa} \\ h_7 = h_6 \end{matrix}\right\} \begin{matrix} T_7 = \mathbf{111.35\ °C} \\ x_7 = 0.0777 \end{matrix}$$

$$\left.\begin{matrix} P_8 = 150 \text{ kPa} \\ x_8 = 1 \end{matrix}\right\} h_8 = 2693.1 \text{ kJ/kg}$$



(*b*) The mass flow rate at the lower stage of the turbine is

$$\dot{m}_8 = x_7 \dot{m}_6 = (0.0777)(191.80 \text{ kg/s}) = 14.90 \text{ kg/s}$$

The power outputs from the high and low pressure stages of the turbine are

$$\dot{W}_{\text{T1,out}} = \dot{m}_3 (h_3 - h_4) = (38.20 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = 15{,}410 \text{ kW}$$

$$\dot{W}_{\text{T2,out}} = \dot{m}_8 (h_8 - h_4) = (14.90 \text{ kJ/kg})(2693.1 - 2344.7)\text{kJ/kg} = \mathbf{5191\ kW}$$

(*c*) We use saturated liquid state at the standard temperature for the dead state enthalpy

$$\left.\begin{matrix} T_0 = 25°\text{C} \\ x_0 = 0 \end{matrix}\right\} h_0 = 104.83 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} = \dot{m}_1 (h_1 - h_0) = (230 \text{ kg/s})(990.14 - 104.83)\text{kJ/kg} = 203{,}621 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{T,out}}}{\dot{E}_{\text{in}}} = \frac{15{,}410 + 5193}{203{,}621} = 0.101 = \mathbf{10.1\%}$$

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10-29 A combined flash-binary geothermal power plant uses hot geothermal water at 230ºC as the heat source. The mass flow rate of isobutane in the binary cycle, the net power outputs from the steam turbine and the binary cycle, and the thermal efficiencies for the binary cycle and the combined plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{matrix} T_1 = 230°\text{C} \\ x_1 = 0 \end{matrix}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{matrix}\right\} x_2 = 0.1661$$

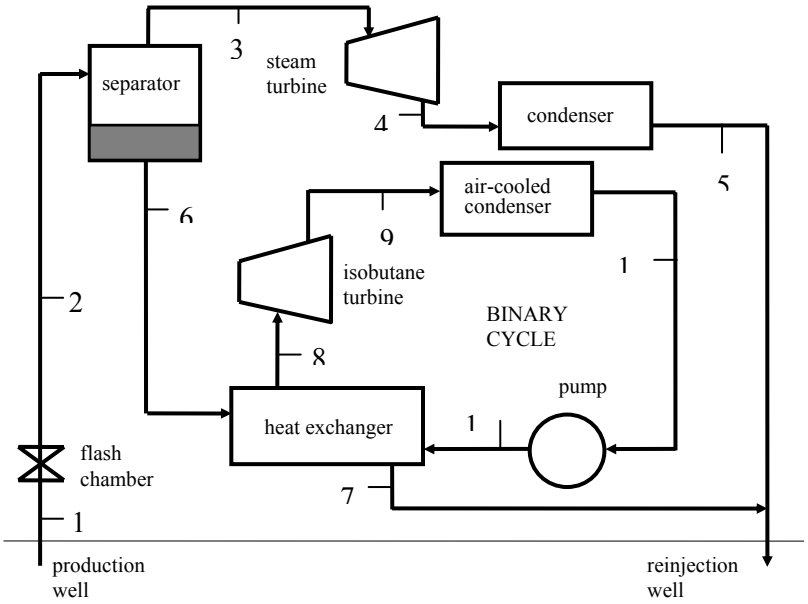
$$\dot{m}_3 = x_2 \dot{m}_1 = (0.1661)(230 \text{ kg/s}) = 38.20 \text{ kg/s}$$
$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 38.20 = 191.80 \text{ kg/s}$$

$$\left.\begin{matrix} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{matrix}\right\} h_3 = 2748.1 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{matrix}\right\} h_4 = 2344.7 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{matrix}\right\} h_6 = 640.09 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_7 = 90°\text{C} \\ x_7 = 0 \end{matrix}\right\} h_7 = 377.04 \text{ kJ/kg}$$



The isobutane properties are obtained from EES:

$$\left.\begin{matrix} P_8 = 3250 \text{ kPa} \\ T_8 = 145°\text{C} \end{matrix}\right\} h_8 = 755.05 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_9 = 400 \text{ kPa} \\ T_9 = 80°\text{C} \end{matrix}\right\} h_9 = 691.01 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_{10} = 400 \text{ kPa} \\ x_{10} = 0 \end{matrix}\right\} \begin{matrix} h_{10} = 270.83 \text{ kJ/kg} \\ \nu_{10} = 0.001839 \text{ m}^3\text{/kg} \end{matrix}$$

$$\begin{aligned} w_{p,\text{in}} &= \nu_{10}(P_{11} - P_{10})/\eta_p \\ &= (0.001819 \text{ m}^3\text{/kg})(3250 - 400)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)/0.90 \\ &= 5.82 \text{ kJ/kg.} \end{aligned}$$
$$h_{11} = h_{10} + w_{p,\text{in}} = 270.83 + 5.82 = 276.65 \text{ kJ/kg}$$

An energy balance on the heat exchanger gives

$$\dot{m}_6 (h_6 - h_7) = \dot{m}_{\text{iso}} (h_8 - h_{11})$$
$$(191.81 \text{ kg/s})(640.09 - 377.04)\text{kJ/kg} = \dot{m}_{\text{iso}} (755.05 - 276.65)\text{kJ/kg} \longrightarrow \dot{m}_{\text{iso}} = \mathbf{105.46 \text{ kg/s}}$$

(*b*) The power outputs from the steam turbine and the binary cycle are

$$\dot{W}_{\text{T,steam}} = \dot{m}_3 (h_3 - h_4) = (38.19 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = \mathbf{15{,}410 \text{ kW}}$$

$$\dot{W}_{\text{T,iso}} = \dot{m}_{iso} (h_8 - h_9) = (105.46 \text{ kJ/kg})(755.05 - 691.01)\text{kJ/kg} = 6753 \text{ kW}$$
$$\dot{W}_{\text{net,binary}} = \dot{W}_{\text{T,iso}} - \dot{m}_{\text{iso}} w_{p,in} = 6753 - (105.46 \text{ kg/s})(5.82 \text{ kJ/kg}) = \mathbf{6139 \text{ kW}}$$

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(*c*) The thermal efficiencies of the binary cycle and the combined plant are

$$\dot{Q}_{\text{in,binary}} = \dot{m}_{\text{iso}}(h_8 - h_{11}) = (105.46\ \text{kJ/kg})(755.05 - 276.65)\text{kJ/kg} = 50{,}454\ \text{kW}$$

$$\eta_{\text{th,binary}} = \frac{\dot{W}_{\text{net,binary}}}{\dot{Q}_{\text{in,binary}}} = \frac{6139}{50{,}454} = 0.122 = \mathbf{12.2\%}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} h_0 = 104.83\ \text{kJ/kg}$$

$$\dot{E}_{\text{in}} = \dot{m}_1(h_1 - h_0) = (230\ \text{kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622\ \text{kW}$$

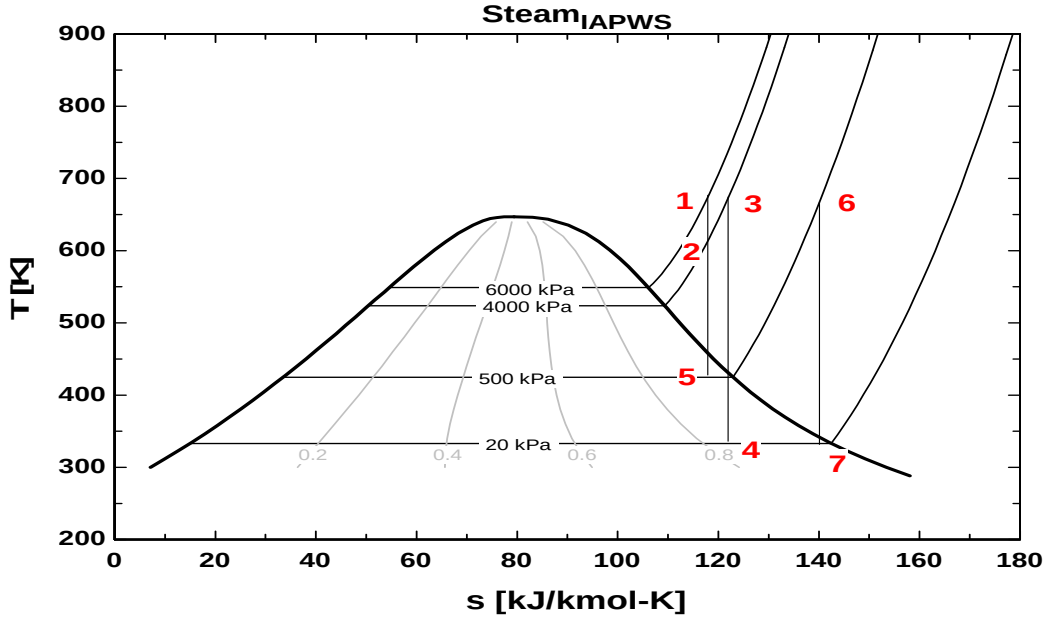
$$\eta_{\text{th,plant}} = \frac{\dot{W}_{\text{T,steam}} + \dot{W}_{\text{net,binary}}}{\dot{E}_{\text{in}}} = \frac{15{,}410 + 6139}{203{,}622} = 0.106 = \mathbf{10.6\%}$$

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The Reheat Rankine Cycle

10-30C The pump work remains the same, the moisture content decreases, everything else increases.

10-31C The *T-s* diagram shows two reheat cases for the reheat Rankine cycle similar to the one shown in Figure 10-11. In the first case there is expansion through the high-pressure turbine from 6000 kPa to 4000 kPa between states 1 and 2 with reheat at 4000 kPa to state 3 and finally expansion in the low-pressure turbine to state 4. In the second case there is expansion through the high-pressure turbine from 6000 kPa to 500 kPa between states 1 and 5 with reheat at 500 kPa to state 6 and finally expansion in the low-pressure turbine to state 7. Increasing the pressure for reheating increases the average temperature for heat addition makes the energy of the steam more available for doing work, see the reheat process 2 to 3 versus the reheat process 5 to 6. Increasing the reheat pressure will increase the cycle efficiency. However, as the reheating pressure increases, the amount of condensation increases during the expansion process in the low-pressure turbine, state 4 versus state 7. An optimal pressure for reheating generally allows for the moisture content of the steam at the low-pressure turbine exit to be in the range of 10 to 15% and this corresponds to quality in the range of 85 to 90%.



10-32C The thermal efficiency of the simple ideal Rankine cycle will probably be higher since the average temperature at which heat is added will be higher in this case.

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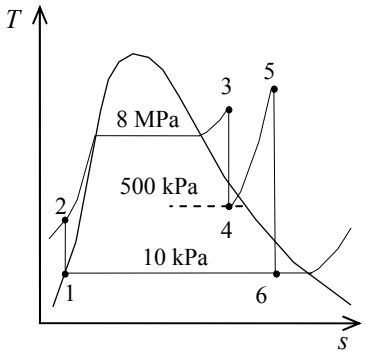
10-33 An ideal reheat steam Rankine cycle produces 5000 kW power. The rates of heat addition and rejection, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@10\,\text{kPa}} = 191.81\,\text{kJ/kg}$$
$$\nu_1 = \nu_{f@10\,\text{kPa}} = 0.001010\,\text{m}^3/\text{kg}$$

$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001010\,\text{m}^3/\text{kg})(8000-10)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 8.07\,\text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 191.81 + 8.07 = 199.88\,\text{kJ/kg}$$



$$\left.\begin{array}{l} P_3 = 8000\,\text{kPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3273.3\,\text{kJ/kg} \\ s_3 = 6.5579\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 500\,\text{kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{6.5579 - 1.8604}{4.9603} = 0.9470 \\ h_4 = h_f + x_4 h_{fg} = 640.09 + (0.9470)(2108.0) = 2636.4\,\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_5 = 500\,\text{kPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3484.5\,\text{kJ/kg} \\ s_5 = 8.0893\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10\,\text{kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{8.0893 - 0.6492}{7.4996} = 0.9921 \\ h_6 = h_f + x_6 h_{fg} = 191.81 + (0.9921)(2392.1) = 2564.9\,\text{kJ/kg} \end{array}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3273.3 - 199.88 + 3484.5 - 2636.4 = 3921.5\,\text{kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2564.9 - 191.81 = 2373.1\,\text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3921.5 - 2373.1 = 1548.5\,\text{kJ/kg}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m}(h_3 - h_4) \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{5000\,\text{kJ/s}}{1548.5\,\text{kJ/kg}} = \mathbf{3.229\,kg/s}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2373.1}{3921.5} = \mathbf{0.395}$$

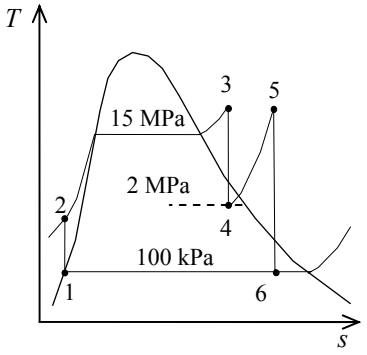
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10-34 An ideal reheat steam Rankine cycle produces 2000 kW power. The mass flow rate of the steam, the rate of heat transfer in the reheater, the power used by the pumps, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),

$$h_1 = h_{f@100\,\text{kPa}} = 417.51\,\text{kJ/kg}$$
$$\nu_1 = \nu_{f@100\,\text{kPa}} = 0.001043\,\text{m}^3/\text{kg}$$



$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001043\,\text{m}^3/\text{kg})(15000-100)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 15.54\,\text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 417.51 + 15.54 = 433.05\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 15{,}000\,\text{kPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3157.9\,\text{kJ/kg} \\ s_3 = 6.1434\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2000\,\text{kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{6.1434 - 2.4467}{3.8923} = 0.9497 \\ h_4 = h_f + x_4 h_{fg} = 908.47 + (0.9497)(1889.8) = 2703.3\,\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_5 = 2000\,\text{kPa} \\ T_5 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3358.2\,\text{kJ/kg} \\ s_5 = 7.2866\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 100\,\text{kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.2866 - 1.3028}{6.0562} = 0.9880 \\ h_6 = h_f + x_6 h_{fg} = 417.51 + (0.9880)(22257.5) = 2648.0\,\text{kJ/kg} \end{array}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3157.9 - 433.05 + 3358.2 - 2703.3 = 3379.8\,\text{kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2648.0 - 417.51 = 2230.5\,\text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 379.8 - 2230.5 = 1149.2\,\text{kJ/kg}$$

The power produced by the cycle is

$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (1.74\,\text{kg/s})(1149.2\,\text{kJ/kg}) = \mathbf{2000\,kW}$$

The rate of heat transfer in the rehetaer is

$$\dot{Q}_{\text{reheater}} = \dot{m}(h_5 - h_4) = (1.740\,\text{kg/s})(3358.2 - 2703.3)\,\text{kJ/kg} = \mathbf{1140\,kW}$$
$$\dot{W}_{\text{P,in}} = \dot{m} w_{\text{P,in}} = (1.740\,\text{kg/s})(15.54\,\text{kJ/kg}) = \mathbf{27\,kW}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2230.5}{3379.8} = \mathbf{0.340}$$

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10-35 A steam power plant that operates on the ideal reheat Rankine cycle is considered. The turbine work output and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

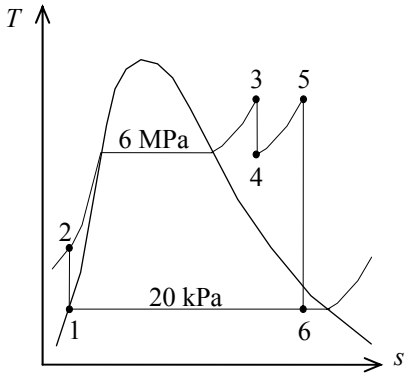
Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\ 20\ \text{kPa}} = 251.42\ \text{kJ/kg}$$

$$\nu_1 = \nu_{f@\ 20\ \text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$w_{p,\text{in}} = \nu_1 (P_2 - P_1)$$
$$= (0.001017\ \text{m}^3/\text{kg})(6000 - 20\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 6.08\ \text{kJ/kg}$$

$$h_2 = h_1 + w_{p,\text{in}} = 251.42 + 6.08 = 257.50\ \text{kJ/kg}$$



$$\left.\begin{matrix} P_3 = 6\ \text{MPa} \\ T_3 = 400°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3178.3\ \text{kJ/kg} \\ s_3 = 6.5432\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 2\ \text{MPa} \\ s_4 = s_3 \end{matrix}\right\} h_4 = 2901.0\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_5 = 2\ \text{MPa} \\ T_5 = 400°\text{C} \end{matrix}\right\} \begin{matrix} h_5 = 3248.4\ \text{kJ/kg} \\ s_5 = 7.1292\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_6 = 20\ \text{kPa} \\ s_6 = s_5 \end{matrix}\right\} \begin{matrix} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.1292 - 0.8320}{7.0752} = 0.8900 \\ h_6 = h_f + x_6 h_{fg} = 251.42 + (0.8900)(2357.5) = 2349.7\ \text{kJ/kg} \end{matrix}$$

The turbine work output and the thermal efficiency are determined from

$$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3178.3 - 2901.0 + 3248.4 - 2349.7 = \mathbf{1176\ kJ/kg}$$

and

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3178.3 - 257.50 + 3248.4 - 2901.0 = 3268\ \text{kJ/kg}$$

$$w_{\text{net}} = w_{T,out} - w_{p,\text{in}} = 1176 - 6.08 = 1170\ \text{kJ/kg}$$

Thus,

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1170\ \text{kJ/kg}}{3268\ \text{kJ/kg}} = 0.358 = \mathbf{35.8\%}$$

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10-36 Problem 10-35 is reconsidered. The problem is to be solved by the diagram window data entry feature of EES by including the effects of the turbine and pump efficiencies and reheat on the steam quality at the low-pressure turbine exit Also, the *T-s* diagram is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data - from diagram window"
{P[6] = 20 [kPa]
P[3] = 6000 [kPa]
T[3] = 400 [C]
P[4] = 2000 [kPa]
T[5] = 400 [C]
Eta_t = 100/100 "Turbine isentropic efficiency"
Eta_p = 100/100 "Pump isentropic efficiency"}

"Pump analysis"
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end

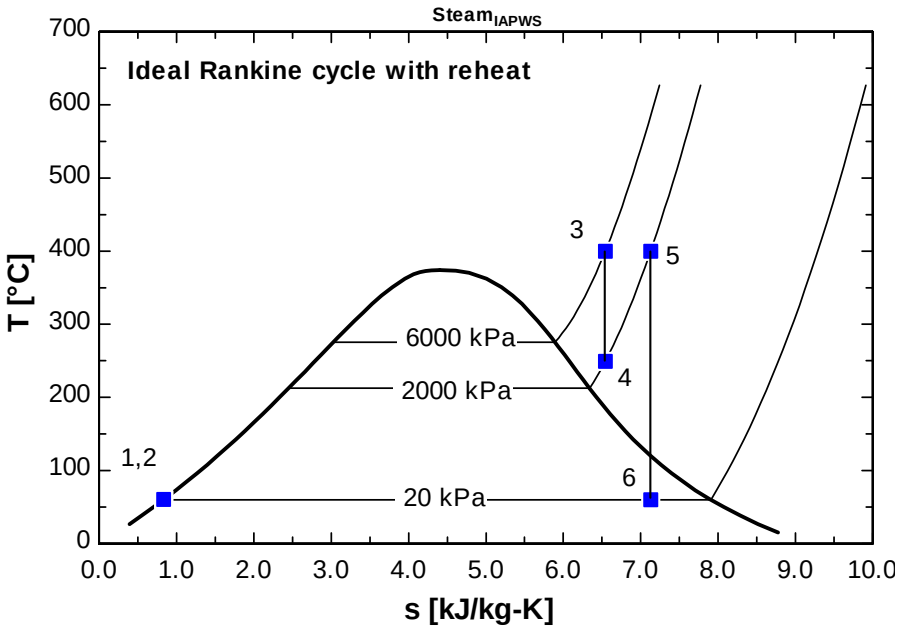
Fluid$='Steam_IAPWS'

P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,T=T[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"
"Low Pressure Turbine analysis"
P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
```

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```
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])

"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eff=W_net/Q_in
```



SOLUTION

```
Eff=0.358
Eta_p=1
Eta_t=1
Fluid$='Steam_IAPWS'
Q_in=3268 [kJ/kg]
Q_out=2098 [kJ/kg]
W_net=1170 [kJ/kg]
W_p=6.083 [kJ/kg]
W_p_s=6.083 [kJ/kg]
W_t_hp=277.2 [kJ/kg]
W_t_lp=898.7 [kJ/kg]
x6s$=''
```

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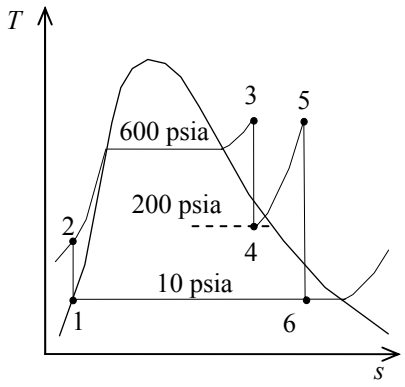
10-37E An ideal reheat steam Rankine cycle produces 5000 kW power. The rates of heat addition and rejection, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4E, A-5E, and A-6E or EES),

$$h_1 = h_{f@10\,\text{psia}} = 161.25\ \text{Btu/lbm}$$
$$\nu_1 = \nu_{f@10\,\text{psia}} = 0.01659\ \text{ft}^3/\text{lbm}$$

$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.01659\ \text{ft}^3/\text{lbm})(600-10)\text{psia}\left(\frac{1\,\text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right)$$
$$= 1.81\ \text{Btu/lbm}$$
$$h_2 = h_1 + w_{\text{p,in}} = 161.25 + 1.81 = 163.06\ \text{Btu/lbm}$$



$$\left.\begin{array}{l} P_3 = 600\ \text{psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1289.9\ \text{Btu/lbm} \\ s_3 = 1.5325\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 200\ \text{psia} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.5325 - 0.54379}{1.00219} = 0.9865 \\ h_4 = h_f + x_4 h_{fg} = 355.46 + (0.9865)(843.33) = 1187.5\ \text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_5 = 200\ \text{psia} \\ T_5 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_5 = 1322.3\ \text{Btu/lbm} \\ s_5 = 1.6771\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10\ \text{psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.6771 - 0.28362}{1.50391} = 0.9266 \\ h_6 = h_f + x_6 h_{fg} = 161.25 + (0.9266)(981.82) = 1071.0\ \text{Btu/lbm} \end{array}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 1289.9 - 163.06 + 1322.3 - 1187.5 = 1261.7\ \text{Btu/lbm}$$
$$q_{\text{out}} = h_6 - h_1 = 1071.0 - 161.25 = 909.7\ \text{Btu/lbm}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1261.7 - 909.8 = 352.0\ \text{Btu/lbm}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{5000\ \text{kJ/s}}{352.0\ \text{Btu/lbm}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = 13.47\ \text{lbm/s}$$

The rates of heat addition and rejection are

$$\dot{Q}_{\text{in}} = \dot{m} q_{\text{in}} = (13.47\ \text{lbm/s})(1261.7\ \text{Btu/lbm}) = \mathbf{16{,}995\ Btu/s}$$
$$\dot{Q}_{\text{out}} = \dot{m} q_{\text{out}} = (13.47\ \text{lbm/s})(909.7\ \text{Btu/lbm}) = \mathbf{12{,}250\ Btu/s}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{5000\ \text{kJ/s}}{16{,}990\ \text{Btu/s}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = \mathbf{0.2790}$$

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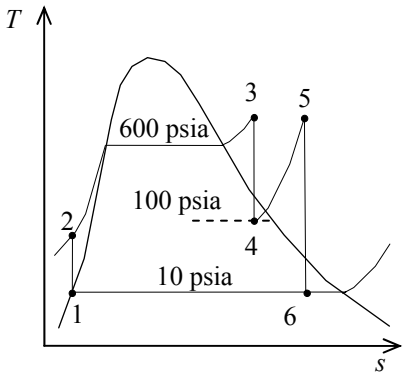
10-38E An ideal reheat steam Rankine cycle produces 5000 kW power. The rates of heat addition and rejection, and the thermal efficiency of the cycle are to be determined for a reheat pressure of 100 psia.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4E, A-5E, and A-6E or EES),

$$h_1 = h_{f@\,10\,\text{psia}} = 161.25\ \text{Btu/lbm}$$
$$\nu_1 = \nu_{f@\,6\,\text{psia}} = 0.01659\ \text{ft}^3/\text{lbm}$$

$$\begin{aligned} w_{\text{p,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.01659\ \text{ft}^3/\text{lbm})(600-10)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &= 1.81\ \text{Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 161.25 + 1.81 = 163.06\ \text{Btu/lbm}$$



$$\left.\begin{array}{l} P_3 = 600\ \text{psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1289.9\ \text{Btu/lbm} \\ s_3 = 1.5325\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 100\ \text{psia} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.5325 - 0.47427}{1.12888} = 0.9374 \\ h_4 = h_f + x_4 h_{fg} = 298.51 + (0.9374)(888.99) = 1131.9\ \text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_5 = 100\ \text{psia} \\ T_5 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_5 = 1329.4\ \text{Btu/lbm} \\ s_5 = 1.7586\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10\ \text{psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{1.7586 - 0.28362}{1.50391} = 0.9808 \\ h_6 = h_f + x_6 h_{fg} = 161.25 + (0.9808)(981.82) = 1124.2\ \text{Btu/lbm} \end{array}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 1289.9 - 163.07 + 1329.4 - 1131.9 = 1324.4\ \text{Btu/lbm}$$
$$q_{\text{out}} = h_6 - h_1 = 1124.2 - 161.25 = 962.9\ \text{Btu/lbm}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1324.4 - 962.9 = 361.5\ \text{Btu/lbm}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{5000\ \text{kJ/s}}{361.5\ \text{Btu/lbm}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = 13.11\ \text{lbm/s}$$

The rates of heat addition and rejection are

$$\dot{Q}_{\text{in}} = \dot{m} q_{\text{in}} = (13.11\ \text{lbm/s})(1324.4\ \text{Btu/lbm}) = \mathbf{17{,}360\ Btu/s}$$
$$\dot{Q}_{\text{out}} = \dot{m} q_{\text{out}} = (13.11\ \text{lbm/s})(962.9\ \text{Btu/lbm}) = \mathbf{12{,}620\ Btu/s}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{5000\ \text{kJ/s}}{17{,}360\ \text{Btu/s}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = \mathbf{0.2729}$$

Discussion The thermal efficiency for 200 psia reheat pressure was determined in the previous problem to be 0.2790. Thus, operating the reheater at 100 psia causes a slight decrease in the thermal efficiency.

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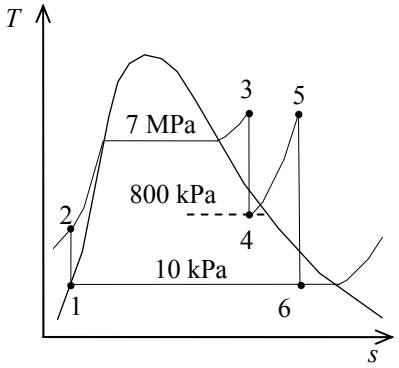
10-39 An ideal reheat Rankine with water as the working fluid is considered. The temperatures at the inlet of both turbines, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@10\,\text{kPa}} = 191.81\,\text{kJ/kg}$$
$$\nu_1 = \nu_{f@10\,\text{kPa}} = 0.001010\,\text{m}^3/\text{kg}$$

$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001010\,\text{m}^3/\text{kg})(7000-10)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 7.06\,\text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 191.81 + 7.06 = 198.87\,\text{kJ/kg}$$



$$\left.\begin{array}{l} P_4 = 800\,\text{kPa} \\ x_4 = 0.93 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 720.87 + (0.93)(2047.5) = 2625.0\,\text{kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 2.0457 + (0.93)(4.6160) = 6.3385\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 7000\,\text{kPa} \\ s_3 = s_4 \end{array}\right\} \begin{array}{l} h_3 = 3085.5\,\text{kJ/kg} \\ T_3 = \mathbf{373.3°C} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10\,\text{kPa} \\ x_6 = 0.90 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.81 + (0.93)(2392.1) = 2416.4\,\text{kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6492 + (0.93)(7.4996) = 7.6239\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 800\,\text{kPa} \\ s_5 = s_6 \end{array}\right\} \begin{array}{l} h_5 = 3302.0\,\text{kJ/kg} \\ T_5 = \mathbf{416.2°C} \end{array}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3085.5 - 198.87 + 3302.0 - 2625.0 = 3563.6\,\text{kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2416.4 - 191.81 = 2224.6\,\text{kJ/kg}$$

and

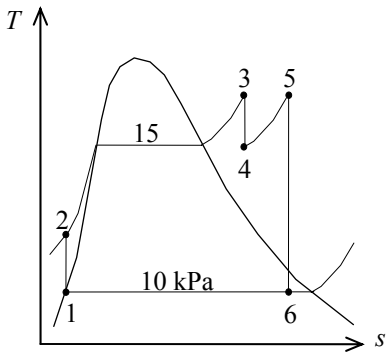
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2224.6}{3563.6} = 0.3757 = \mathbf{37.6\%}$$

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10-40 A steam power plant that operates on an ideal reheat Rankine cycle between the specified pressure limits is considered. The pressure at which reheating takes place, the total rate of heat input in the boiler, and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),



$$h_1 = h_{\text{sat@ 10 kPa}} = 191.81 \text{ kJ/kg}$$

$$\nu_1 = \nu_{\text{sat@ 10 kPa}} = 0.00101 \text{ m}^3/\text{kg}$$

$$w_{p,\text{in}} = \nu_1 (P_2 - P_1)$$
$$= (0.00101 \text{ m}^3/\text{kg})(15{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$
$$= 15.14 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 15.14 = 206.95 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 15 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3310.8 \text{ kJ/kg} \\ s_3 = 6.3480 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7 \text{ kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6492 + (0.90)(7.4996) = 7.3988 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} T_5 = 500°\text{C} \\ s_5 = s_6 \end{array}\right\} \begin{array}{l} P_5 = \mathbf{2150 \text{ kPa}} \text{ (the reheat pressure)} \\ h_5 = 3466.61 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2.15 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 2817.2 \text{ kJ/kg}$$

(*b*) The rate of heat supply is

$$\dot{Q}_{\text{in}} = \dot{m}[(h_3 - h_2) + (h_5 - h_4)]$$
$$= (12 \text{ kg/s})(3310.8 - 206.95 + 3466.61 - 2817.2)\text{kJ/kg}$$
$$= \mathbf{45{,}039 \text{ kW}}$$

(*c*) The thermal efficiency is determined from

$$\dot{Q}_{\text{out}} = \dot{m}(h_6 - h_1) = (12 \text{ kJ/s})(2344.7 - 191.81)\text{kJ/kg} = 25{,}835 \text{ kJ/s}$$

Thus,

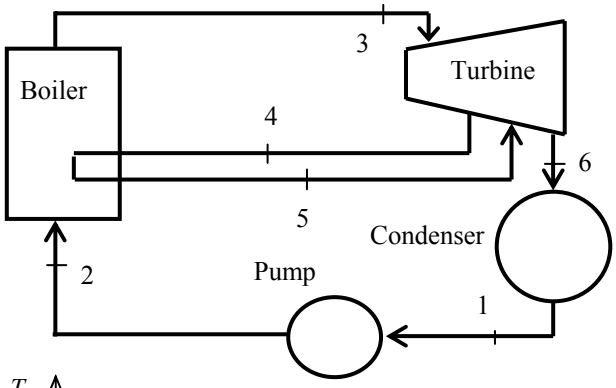
$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{25{,}834 \text{ kJ/s}}{45{,}039 \text{ kJ/s}} = \mathbf{42.6\%}$$

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10-41 A steam power plant that operates on a reheat Rankine cycle is considered. The condenser pressure, the net power output, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),



$$\left.\begin{matrix} P_3 = 12.5 \text{ MPa} \\ T_3 = 550°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3476.5 \text{ kJ/kg} \\ s_3 = 6.6317 \text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 2 \text{ MPa} \\ s_{4s} = s_3 \end{matrix}\right\} h_{4s} = 2948.1 \text{ kJ/kg}$$

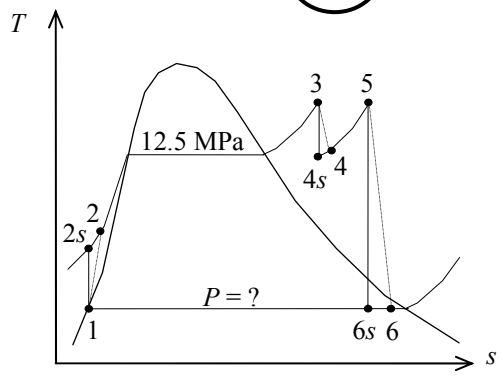
$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$\rightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 3476.5 - (0.85)(3476.5 - 2948.1)$$
$$= 3027.3 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_5 = 2 \text{ MPa} \\ T_5 = 450°\text{C} \end{matrix}\right\} \begin{matrix} h_5 = 3358.2 \text{ kJ/kg} \\ s_5 = 7.2815 \text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_6 = ? \\ x_6 = 0.95 \end{matrix}\right\} h_6 = \quad \text{(Eq. 1)}$$

$$\left.\begin{matrix} P_6 = ? \\ s_6 = s_5 \end{matrix}\right\} h_{6s} = \quad \text{(Eq. 2)}$$



$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s}) = 3358.2 - (0.85)(3358.2 - h_{6s}) \quad \text{(Eq. 3)}$$

The pressure at state 6 may be determined by a trial-error approach from the steam tables or by using EES from the above three equations:

$$P_6 = \mathbf{9.73 \text{ kPa}}, \quad h_6 = 2463.3 \text{ kJ/kg},$$

(*b*) Then,

$$h_1 = h_{f@\ 9.73 \text{ kPa}} = 189.57 \text{ kJ/kg}$$

$$\nu_1 = \nu_{f@\ 10 \text{ kPa}} = 0.001010 \text{ m}^3\text{/kg}$$

$$w_{p,\text{in}} = \nu_1 (P_2 - P_1)/\eta_p$$
$$= (0.00101 \text{ m}^3\text{/kg})(12{,}500 - 9.73 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/(0.90)$$
$$= 14.02 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{p,\text{in}} = 189.57 + 14.02 = 203.59 \text{ kJ/kg}$$

Cycle analysis:

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3476.5 - 203.59 + 3358.2 - 2463.3 = 3603.8 \text{ kJ/kg}$$

$$q_{\text{out}} = h_6 - h_1 = 2463.3 - 189.57 = 2273.7 \text{ kJ/kg}$$

$$\dot{W}_{\text{net}} = \dot{m}(q_{\text{in}} - q_{\text{out}}) = (7.7 \text{ kg/s})(3603.8 - 2273.7)\text{kJ/kg} = \mathbf{10{,}242 \text{ kW}}$$

(*c*) The thermal efficiency is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2273.7 \text{ kJ/kg}}{3603.8 \text{ kJ/kg}} = 0.369 = \mathbf{36.9\%}$$

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Regenerative Rankine Cycle

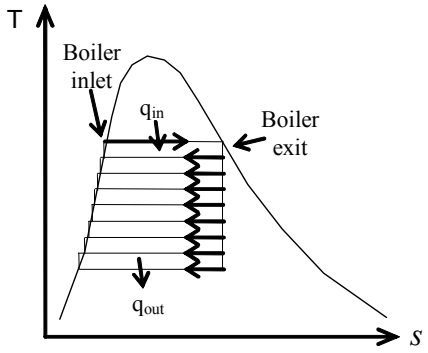
10-42C Moisture content remains the same, everything else decreases.

10-43C This is a smart idea because we waste little work potential but we save a lot from the heat input. The extracted steam has little work potential left, and most of its energy would be part of the heat rejected anyway. Therefore, by regeneration, we utilize a considerable amount of heat by sacrificing little work output.

10-44C In open feedwater heaters, the two fluids actually mix, but in closed feedwater heaters there is no mixing.

10-45C Both cycles would have the same efficiency.

10-46C To have the same thermal efficiency as the Carnot cycle, the cycle must receive and reject heat isothermally. Thus the liquid should be brought to the saturated liquid state at the boiler pressure isothermally, and the steam must be a saturated vapor at the turbine inlet. This will require an infinite number of heat exchangers (feedwater heaters), as shown on the *T-s* diagram.



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10-47E Feedwater is heated by steam in a feedwater heater of a regenerative Rankine cycle. The ratio of the bleed steam mass flow rate to the inlet feedwater mass flow rate is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4E through A-6E or EES),

$$h_1 \cong h_{f@\,110°\text{F}} = 78.02 \text{ Btu/lbm}$$

$$\left.\begin{matrix} P_2 = 20 \text{ psia} \\ T_2 = 250°\text{F} \end{matrix}\right\} \; h_2 = 1167.2 \text{ Btu/lbm}$$

$$h_3 \cong h_{f@\,225°\text{F}} = 193.32 \text{ Btu/lbm}$$

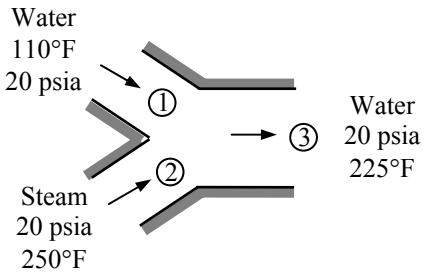
Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$



Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}} \overset{\not\nearrow 0 \text{ (steady)}}{}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

Solving for the bleed steam mass flow rate to the inlet feedwater mass flow rate, and substituting gives

$$\frac{\dot{m}_2}{\dot{m}_1} = \frac{h_1 - h_3}{h_3 - h_2} = \frac{(78.02 - 193.32) \text{ kJ/kg}}{(193.32 - 1167.2) \text{ kJ/kg}} = \mathbf{0.118}$$

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10-48 In a regenerative Rankine cycle, the closed feedwater heater with a pump as shown in the figure is arranged so that the water at state 5 is mixed with the water at state 2 to form a feedwater which is a saturated liquid. The amount of bleed steam required to heat 1 kg of feedwater is to be determined.

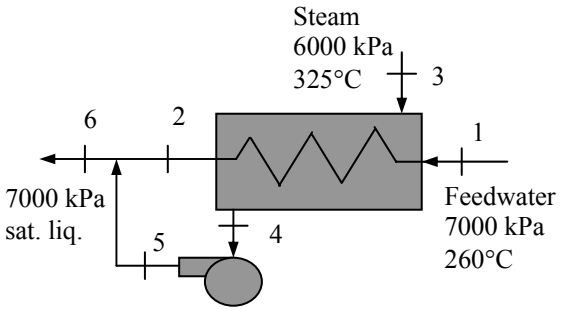
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through A-6),

$$\left.\begin{matrix} P_1 = 7000\ \text{kPa} \\ T_1 = 260°\text{C} \end{matrix}\right\} \quad h_1 \cong h_{f\,@\,260°\text{C}} = 1134.8\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 6000\ \text{kPa} \\ T_3 = 325°\text{C} \end{matrix}\right\} \quad h_3 = 2969.5\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_6 = 7000\ \text{kPa} \\ x_6 = 0 \end{matrix}\right\} \quad h_6 = h_{f\,@\,7000\ \text{kPa}} = 1267.5\ \text{kJ/kg}$$



Analysis We take the entire unit as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 + \dot{m}_3 w_{\text{P,in}} = \dot{m}_6 h_6$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 + \dot{m}_3 w_{\text{P,in}} = (\dot{m}_1 + \dot{m}_3) h_6$$

Solving this for $\dot{m}_3$,

$$\dot{m}_3 = \dot{m}_1 \frac{h_6 - h_1}{(h_3 - h_6) + w_{\text{P,in}}} = (1\ \text{kg/s}) \frac{1267.5 - 1134.8}{2969.5 - 1267.5 + 1.319} = \mathbf{0.0779\ kg/s}$$

where

$$w_{\text{P,in}} = \nu_4 (P_5 - P_4) = \nu_{f\,@\,6000\ \text{kPa}} (P_5 - P_4)$$

$$= (0.001319\ \text{m}^3/\text{kg})(7000 - 6000)\ \text{kPa} \left(\frac{1\ \text{kPa}}{1\ \text{kPa} \cdot \text{m}^3} \right) = 1.319\ \text{kJ/kg}$$

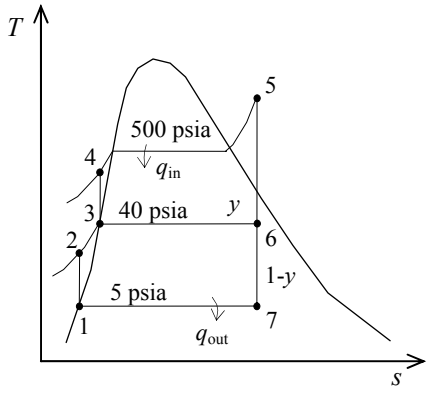
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10-49E An ideal regenerative Rankine cycle with an open feedwater heater is considered. The work produced by the turbine, the work consumed by the pumps, and the heat rejected in the condenser are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,5\,\text{psia}} = 130.18\ \text{Btu/lbm}$$
$$\upsilon_1 = \upsilon_{f@\,5\,\text{psia}} = 0.01641\ \text{ft}^3/\text{lbm}$$



$$w_{\text{PI,in}} = \upsilon_1 (P_2 - P_1)$$
$$= (0.01641\ \text{ft}^3/\text{lbm})(40-5)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right)$$
$$= 0.11\ \text{Btu/lbm}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 130.18 + 0.11 = 130.29\ \text{Btu/lbm}$$

$$h_3 = h_{f@\,40\,\text{psia}} = 236.14\ \text{Btu/lbm}$$
$$\upsilon_3 = \upsilon_{f@\,40\,\text{psia}} = 0.01715\ \text{ft}^3/\text{lbm}$$

$$w_{\text{PII,in}} = \upsilon_3 (P_4 - P_3)$$
$$= (0.01715\ \text{ft}^3/\text{lbm})(500-40)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right)$$
$$= 1.46\ \text{Btu/lbm}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 236.14 + 1.46 = 237.60\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_5 = 500\ \text{psia} \\ T_5 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_5 = 1298.6\ \text{Btu/lbm} \\ s_5 = 1.5590\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 40\ \text{psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{1.5590 - 0.39213}{1.28448} = 0.9085 \\ h_6 = h_f + x_6 h_{fg} = 236.14 + (0.9085)(933.69) = 1084.4\ \text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_7 = 5\ \text{psia} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.5590 - 0.23488}{1.60894} = 0.8230 \\ h_7 = h_f + x_7 h_{fg} = 130.18 + (0.8230)(1000.5) = 953.63\ \text{Btu/lbm} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heater. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system} \overset{\nearrow 0\ \text{(steady)}}{} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y) h_2 = 1 h_3$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{236.14 - 130.29}{1084.4 - 130.29} = 0.1109$$

Then,

$$w_{\text{T,out}} = h_5 - h_6 + (1-y)(h_6 - h_7) = 1298.6 - 1084.4 + (1-0.1109)(1084.4 - 953.63) = \mathbf{330.5\ Btu/lbm}$$
$$w_{\text{P,in}} = w_{\text{PI,in}} + w_{\text{PII,in}} = 0.11 + 1.46 = \mathbf{1.57\ Btu/lbm}$$
$$q_{\text{out}} = (1-y)(h_7 - h_1) = (1-0.1109)(953.63 - 130.18) = \mathbf{732.1 Btu/lbm}$$

Also,

$$q_{\text{in}} = h_5 - h_4 = 1298.6 - 237.60 = 1061\ \text{Btu/lbm}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{732.1}{1061} = 0.3100$$

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10-50E An ideal regenerative Rankine cycle with an open feedwater heater is considered. The change in thermal efficiency when the steam supplied to the open feedwater heater is at 60 psia rather than 40 psia is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

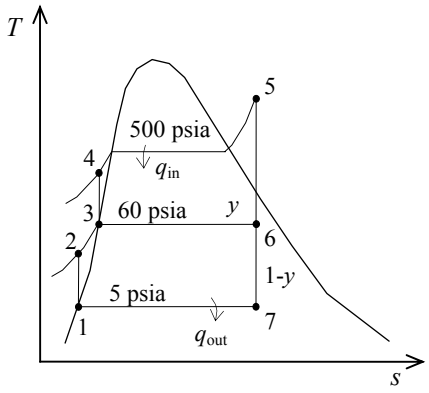
Analysis From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,5\,\text{psia}} = 130.18\ \text{Btu/lbm}$$
$$\nu_1 = \nu_{f@\,5\,\text{psia}} = 0.01641\ \text{ft}^3/\text{lbm}$$

$$w_{\text{PI,in}} = \nu_1 (P_2 - P_1)$$
$$= (0.01641\ \text{ft}^3/\text{lbm})(60-5)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right)$$
$$= 0.17\ \text{Btu/lbm}$$
$$h_2 = h_1 + w_{\text{p,in}} = 130.18 + 0.17 = 130.35\ \text{Btu/lbm}$$

$$h_3 = h_{f@\,60\,\text{psia}} = 262.20\ \text{Btu/lbm}$$
$$\nu_3 = \nu_{f@\,60\,\text{psia}} = 0.01738\ \text{ft}^3/\text{lbm}$$

$$w_{\text{PII,in}} = \nu_3 (P_4 - P_3)$$
$$= (0.01738\ \text{ft}^3/\text{lbm})(500-60)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right)$$
$$= 1.42\ \text{Btu/lbm}$$
$$h_4 = h_3 + w_{\text{p,in}} = 262.20 + 1.42 = 263.62\ \text{Btu/lbm}$$



$$\left.\begin{array}{l} P_5 = 500\ \text{psia} \\ T_5 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_5 = 1298.6\ \text{Btu/lbm} \\ s_5 = 1.5590\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 60\ \text{psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.5590 - 0.42728}{1.21697} = 0.9300 \\ h_6 = h_f + x_6 h_{fg} = 262.20 + (0.9300)(915.61) = 1113.7\ \text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_7 = 5\ \text{psia} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.5590 - 0.23488}{1.60894} = 0.8230 \\ h_7 = h_f + x_7 h_{fg} = 130.18 + (0.8230)(1000.5) = 953.63\ \text{Btu/lbm} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heater. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\nearrow 0\ \text{(steady)}} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y) h_2 = 1 h_3$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{262.20 - 130.35}{1113.7 - 130.35} = 0.1341$$

Then,

$$q_{\text{in}} = h_5 - h_4 = 1298.6 - 263.62 = 1035\ \text{Btu/lbm}$$
$$q_{\text{out}} = (1-y)(h_7 - h_1) = (1 - 0.1341)(953.63 - 130.18) = 713.0\ \text{Btu/lbm}$$

and $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{713.0}{1035} = \mathbf{0.3111}$$

When the reheater pressure is increased from 40 psia to 60 psia, the thermal efficiency increases from **0.3100** to **0.3111**, which is an increase of **0.35%**.

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10-51E The optimum bleed pressure for the open feedwater heater that maximizes the thermal efficiency of the cycle is to be determined by EES.

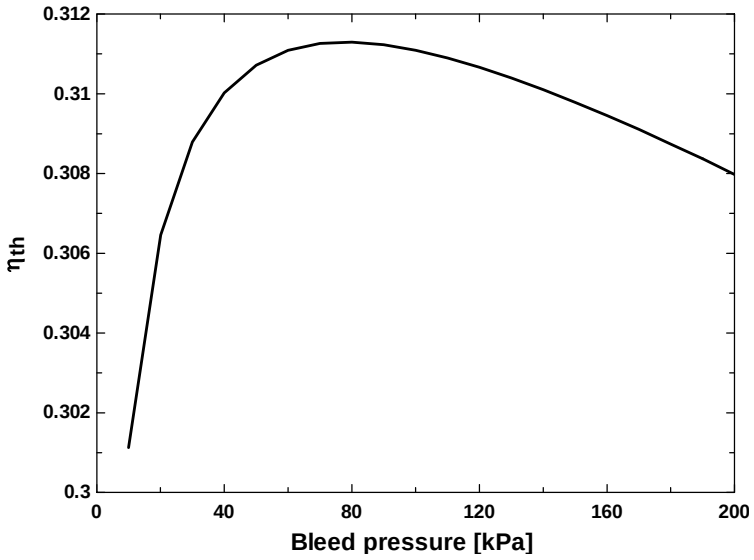
Analysis The EES program used to solve this problem as well as the solutions are given below.

```
P[5]=500 [psia]
T[5]=600 [F]
P[6]=60 [psia]
P[7]=5 [psia]
x[3]=0

"Analysis"
Fluid$='steam_iapws'
"pump I"
P[1]=P[7]
x[1]=0
h[1]=enthalpy(Fluid$, P=P[1], x=x[1])
v[1]=volume(Fluid$, P=P[1], x=x[1])
P[3]=P[6]
P[2]=P[3]
w_pI_in=v[1]*(P[2]-P[1])*Convert(psia-ft^3, Btu)
h[2]=h[1]+w_pI_in
"pump II"
h[3]=enthalpy(Fluid$, P=P[3], x=x[3])
v[3]=volume(Fluid$, P=P[3], x=x[3])
P[4]=P[5]
w_pII_in=v[3]*(P[4]-P[3])*Convert(psia-ft^3, Btu)
h[4]=h[3]+w_pII_in
"turbine"
h[5]=enthalpy(Fluid$, P=P[5], T=T[5])
s[5]=entropy(Fluid$, P=P[5], T=T[5])
s[6]=s[5]
h[6]=enthalpy(Fluid$, P=P[6], s=s[6])
x[6]=quality(Fluid$, P=P[6], s=s[6])
s[7]=s[5]
h[7]=enthalpy(Fluid$, P=P[7], s=s[7])
x[7]=quality(Fluid$, P=P[7], s=s[7])
"open feedwater heater"
y*h[6]+(1-y)*h[2]=h[3]  "y=m_dot_6/m_dot_3"
"cycle"
q_in=h[5]-h[4]
q_out=(1-y)*(h[7]-h[1])
w_net=q_in-q_out
Eta_th=1-q_out/q_in
```

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P_6 [kPa]	η_{th}
10	0.3011
20	0.3065
30	0.3088
40	0.3100
50	0.3107
60	0.3111
70	0.31126
80	**0.31129**
90	0.3112
100	0.3111
110	0.3109
120	0.3107
130	0.3104
140	0.3101
150	0.3098
160	0.3095
170	0.3091
180	0.3087
190	0.3084
200	0.3080

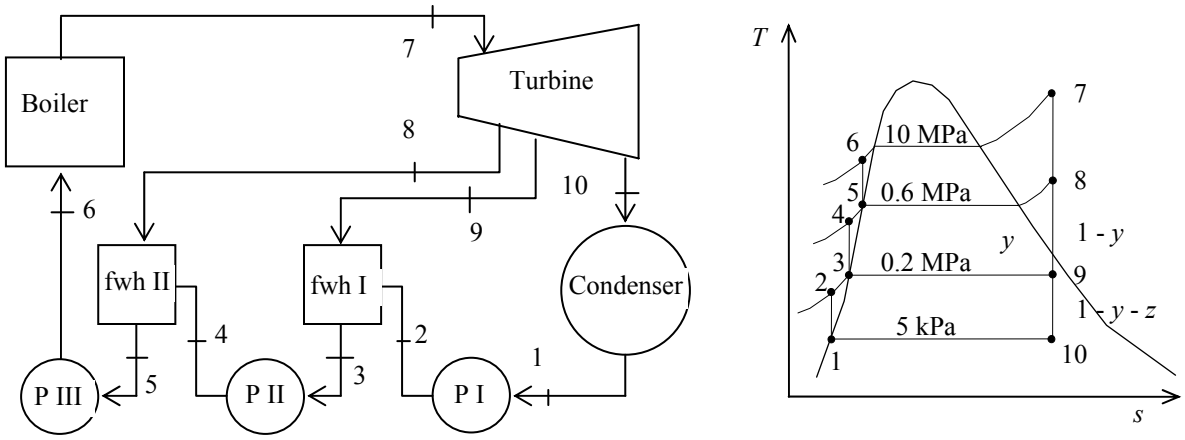


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10-52 A steam power plant operates on an ideal regenerative Rankine cycle with two open feedwater heaters. The net power output of the power plant and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis



(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\ 5\ \text{kPa}} = 137.75\ \text{kJ/kg}$$
$$\nu_1 = \nu_{f@\ 5\ \text{kPa}} = 0.001005\ \text{m}^3/\text{kg}$$
$$w_{pI,\text{in}} = \nu_1 (P_2 - P_1) = (0.001005\ \text{m}^3/\text{kg})(200 - 5\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 0.20\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 137.75 + 0.20 = 137.95\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.2\ \text{MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\ 0.2\ \text{MPa}} = 504.71\ \text{kJ/kg} \\ \nu_3 = \nu_{f@\ 0.2\ \text{MPa}} = 0.001061\ \text{m}^3/\text{kg} \end{array}$$

$$w_{pII,\text{in}} = \nu_3 (P_4 - P_3) = (0.001061\ \text{m}^3/\text{kg})(600 - 200\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 0.42\ \text{kJ/kg}$$
$$h_4 = h_3 + w_{pII,\text{in}} = 504.71 + 0.42 = 505.13\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 0.6\ \text{MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_5 = h_{f@\ 0.6\ \text{MPa}} = 670.38\ \text{kJ/kg} \\ \nu_5 = \nu_{f@\ 0.6\ \text{MPa}} = 0.001101\ \text{m}^3/\text{kg} \end{array}$$

$$w_{pIII,\text{in}} = \nu_5 (P_6 - P_5) = (0.001101\ \text{m}^3/\text{kg})(10{,}000 - 600\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 10.35\ \text{kJ/kg}$$

$$h_6 = h_5 + w_{pIII,\text{in}} = 670.38 + 10.35 = 680.73\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 10\ \text{MPa} \\ T_7 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3625.8\ \text{kJ/kg} \\ s_7 = 6.9045\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 0.6\ \text{MPa} \\ s_8 = s_7 \end{array}\right\} h_8 = 2821.8\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 0.2\ \text{MPa} \\ s_9 = s_7 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{6.9045 - 1.5302}{5.5968} = 0.9602 \\ h_9 = h_f + x_9 h_{fg} = 504.71 + (0.9602)(2201.6) = 2618.7\ \text{kJ/kg} \end{array}$$

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$$\left.\begin{array}{l} P_{10} = 5 \text{ kPa} \\ s_{10} = s_7 \end{array}\right\} \begin{array}{l} x_{10} = \dfrac{s_{10} - s_f}{s_{fg}} = \dfrac{6.9045 - 0.4762}{7.9176} = 0.8119 \\ h_{10} = h_f + x_{10} h_{fg} = 137.75 + (0.8119)(2423.0) = 2105.0 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

FWH-2:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_4 h_4 = \dot{m}_5 h_5 \longrightarrow y h_8 + (1 - y) h_4 = 1(h_5)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_5$). Solving for y,

$$y = \frac{h_5 - h_4}{h_8 - h_4} = \frac{670.38 - 505.13}{2821.8 - 505.13} = 0.07133$$

FWH-1:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_9 h_9 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow z h_9 + (1 - y - z) h_2 = (1 - y) h_3$$

where z is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for z,

$$z = \frac{h_3 - h_2}{h_9 - h_2}(1 - y) = \frac{504.71 - 137.95}{2618.7 - 137.95}(1 - 0.07136) = 0.1373$$

Then,

$$q_{\text{in}} = h_7 - h_6 = 3625.8 - 680.73 = 2945.0 \text{ kJ/kg}$$

$$q_{\text{out}} = (1 - y - z)(h_{10} - h_1) = (1 - 0.07133 - 0.1373)(2105.0 - 137.75) = 1556.8 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2945.0 - 1556.8 = 1388.2 \text{ kJ/kg}$$

and

$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (22 \text{ kg/s})(1388.2 \text{ kJ/kg}) = 30{,}540 \text{ kW} \cong \mathbf{30.5\ MW}$$

(*b*) The thermal efficiency is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1556.8 \text{ kJ/kg}}{2945.0 \text{ kJ/kg}} = \mathbf{47.1\%}$$

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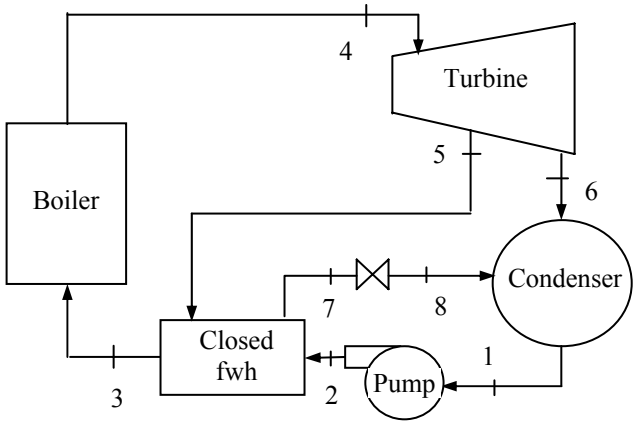
10-53 An ideal regenerative Rankine cycle with a closed feedwater heater is considered. The work produced by the turbine, the work consumed by the pumps, and the heat added in the boiler are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,20\,\text{kPa}} = 251.42\,\text{kJ/kg}$$
$$\nu_1 = \nu_{f@\,20\,\text{kPa}} = 0.001017\,\text{m}^3/\text{kg}$$

$$w_{\text{p,in}} = \nu_1 (P_2 - P_1)$$
$$= (0.001017\,\text{m}^3/\text{kg})(3000-20)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 3.03\,\text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 3.03 = 254.45\,\text{kJ/kg}$$

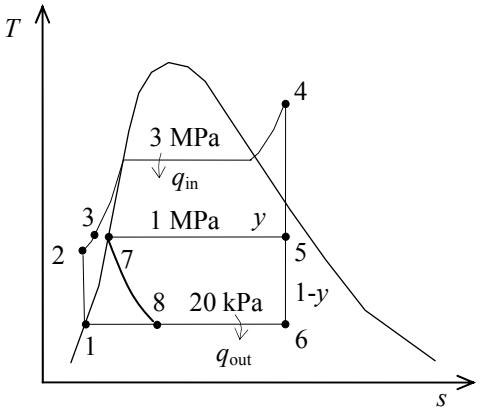


$$\left.\begin{array}{l} P_4 = 3000\,\text{kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1\,\text{kJ/kg} \\ s_4 = 6.7450\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 1000\,\text{kPa} \\ s_5 = s_4 \end{array}\right\} h_5 = 2851.9\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 20\,\text{kPa} \\ s_6 = s_4 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.7450 - 0.8320}{7.0752} = 0.8357 \\ h_6 = h_f + x_6 h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7\,\text{kJ/kg} \end{array}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.



$$\left.\begin{array}{l} P_7 = 1000\,\text{kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 762.51\,\text{kJ/kg} \\ T_7 = 179.9°\text{C} \end{array}$$

$$h_8 = h_7 = 762.51\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 3000\,\text{kPa} \\ T_3 = T_7 = 209.9°\text{C} \end{array}\right\} h_3 = 763.53\,\text{kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$
$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 254.45}{2851.9 - 762.51} = 0.2437$$

Then,

$$w_{\text{T,out}} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2851.9 + (1-0.2437)(2851.9 - 2221.7) = \mathbf{740.9\,kJ/kg}$$
$$w_{\text{P,in}} = \mathbf{3.03\,kJ/kg}$$
$$q_{\text{in}} = h_4 - h_3 = 3116.1 - 763.53 = \mathbf{2353\,kJ/kg}$$

Also, $w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 740.9 - 3.03 = 737.8\,\text{kJ/kg}$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{737.8}{2353} = 0.3136$$

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10-54 Problem 10-53 is reconsidered. The optimum bleed pressure for the open feedwater heater that maximizes the thermal efficiency of the cycle is to be determined by EES.

Analysis The EES program used to solve this problem as well as the solutions are given below.

```
"Given"
P[4]=3000 [kPa]
T[4]=350 [C]
P[5]=600 [kPa]
P[6]=20 [kPa]

P[3]=P[4]
P[2]=P[3]
P[7]=P[5]
P[1]=P[6]

"Analysis"
Fluid$='steam_iapws'

"pump I"
x[1]=0
h[1]=enthalpy(Fluid$, P=P[1], x=x[1])
v[1]=volume(Fluid$, P=P[1], x=x[1])
w_p_in=v[1]*(P[2]-P[1])
h[2]=h[1]+w_p_in

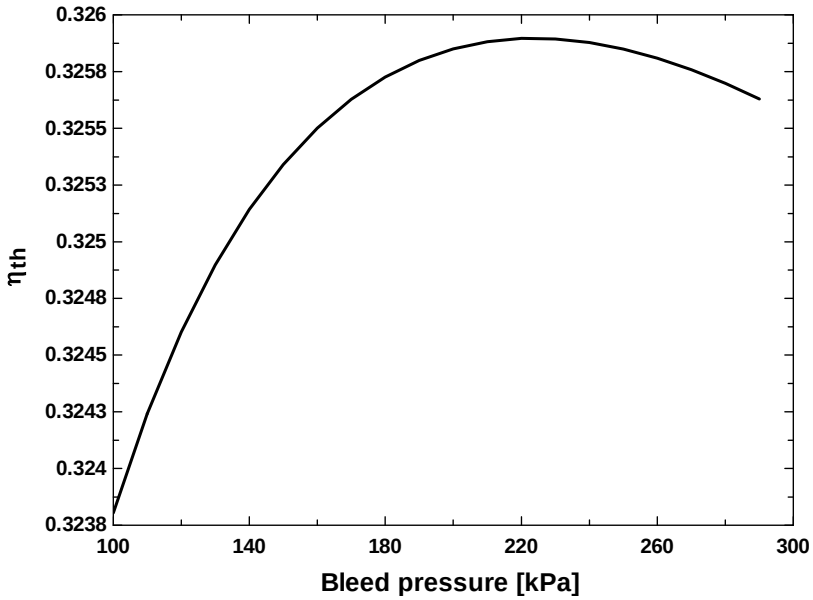
"turbine"
h[4]=enthalpy(Fluid$, P=P[4], T=T[4])
s[4]=entropy(Fluid$, P=P[4], T=T[4])
s[5]=s[4]
h[5]=enthalpy(Fluid$, P=P[5], s=s[5])
T[5]=temperature(Fluid$, P=P[5], s=s[5])
x[5]=quality(Fluid$, P=P[5], s=s[5])
s[6]=s[4]
h[6]=enthalpy(Fluid$, P=P[6], s=s[6])
x[6]=quality(Fluid$, P=P[6], s=s[6])

"closed feedwater heater"
x[7]=0
h[7]=enthalpy(Fluid$, P=P[7], x=x[7])
T[7]=temperature(Fluid$, P=P[7], x=x[7])
T[3]=T[7]
h[3]=enthalpy(Fluid$, P=P[3], T=T[3])
y=(h[3]-h[2])/(h[5]-h[7])  "y=m_dot_5/m_dot_4"

"cycle"
q_in=h[4]-h[3]
w_T_out=h[4]-h[5]+(1-y)*(h[5]-h[6])
w_net=w_T_out-w_p_in
Eta_th=w_net/q_in
```

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P_6 [kPa]	η_{th}
100	0.32380
110	0.32424
120	0.32460
130	0.32490
140	0.32514
150	0.32534
160	0.32550
170	0.32563
180	0.32573
190	0.32580
200	0.32585
210	0.32588
220	**0.32590**
230	0.32589
240	0.32588
250	0.32585
260	0.32581
270	0.32576
280	0.32570
290	0.32563



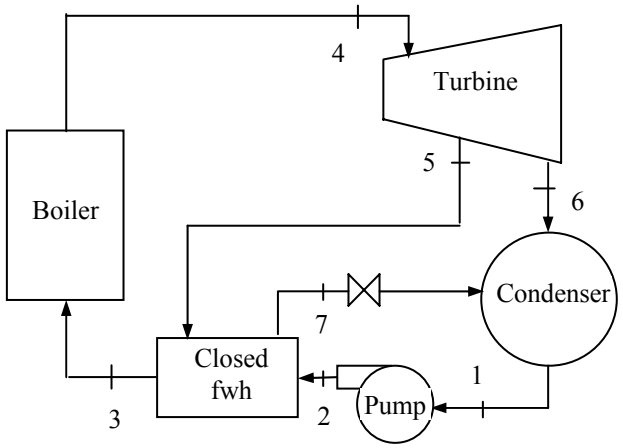
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10-55 A regenerative Rankine cycle with a closed feedwater heater is considered. The thermal efficiency is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),

$$h_1 = h_{f@\,20\,\text{kPa}} = 251.42\ \text{kJ/kg}$$
$$\nu_1 = \nu_{f@\,20\,\text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$



$$w_{\text{p,in}} = \nu_1 (P_2 - P_1)$$
$$= (0.001017\ \text{m}^3/\text{kg})(3000-20)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 3.03\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 3.03 = 254.45\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000\ \text{kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1\ \text{kJ/kg} \\ s_4 = 6.7450\ \text{kJ/kg}\cdot\text{K} \end{array}$$

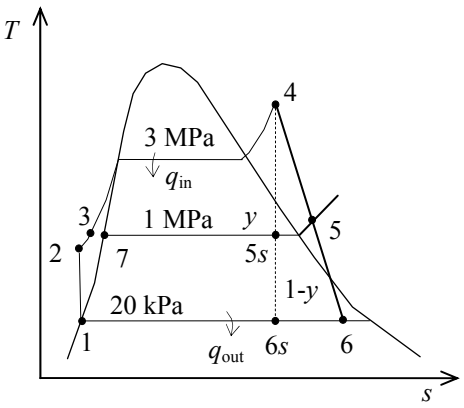
$$\left.\begin{array}{l} P_5 = 1000\ \text{kPa} \\ s_{5s} = s_4 \end{array}\right\} \quad h_{5s} = 2851.9\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 20\ \text{kPa} \\ s_{6s} = s_4 \end{array}\right\} \quad x_{6s} = \frac{s_{6s} - s_f}{s_{fg}} = \frac{6.7450 - 0.8320}{7.0752} = 0.8357$$
$$h_{6s} = h_f + x_{6s} h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7\ \text{kJ/kg}$$

$$\eta_{\text{T}} = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow h_5 = h_4 - \eta_{\text{T}}(h_4 - h_{5s}) = 3116.1 - (0.90)(3116.1 - 2851.9) = 2878.3\ \text{kJ/kg}$$

$$\eta_{\text{T}} = \frac{h_4 - h_6}{h_4 - h_{6s}} \longrightarrow h_6 = h_4 - \eta_{\text{T}}(h_4 - h_{6s}) = 3116.1 - (0.90)(3116.1 - 2221.7) = 2311.1\ \text{kJ/kg}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.



$$\left.\begin{array}{l} P_7 = 1000\ \text{kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 762.51\ \text{kJ/kg} \\ T_7 = 179.9°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 3000\ \text{kPa} \\ T_3 = T_7 = 209.9°\text{C} \end{array}\right\} \quad h_3 = 763.53\ \text{kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$
$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 254.45}{2878.3 - 762.51} = 0.2406$$

Then,

$$w_{\text{T,out}} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2878.3 + (1 - 0.2406)(2878.3 - 2311.1) = 668.5\ \text{kJ/kg}$$
$$w_{\text{P,in}} = 3.03\ \text{kJ/kg}$$
$$q_{\text{in}} = h_4 - h_3 = 3116.1 - 763.53 = 2353\ \text{kJ/kg}$$

Also, $w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 668.5 - 3.03 = 665.5\ \text{kJ/kg}$

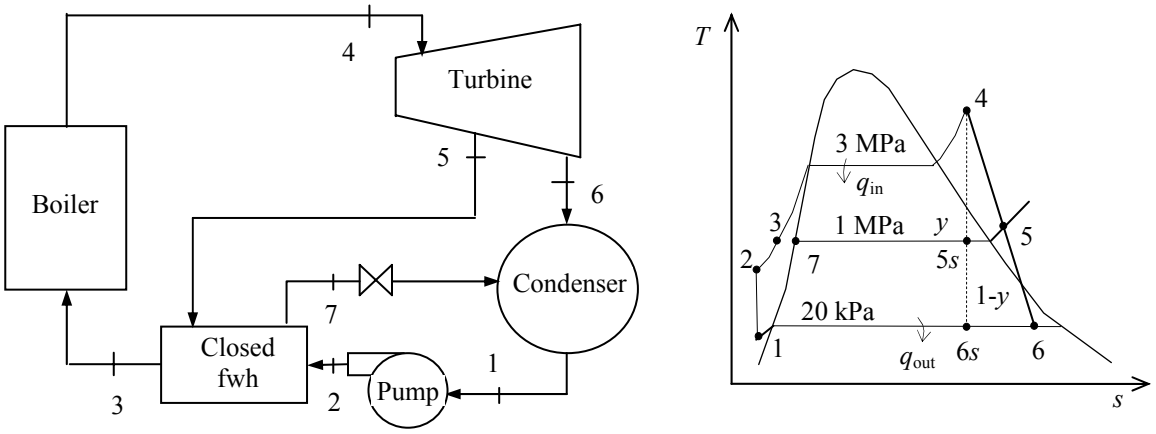
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{665.5}{2353} = 0.2829 = \mathbf{28.3\%}$$

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10-56 A regenerative Rankine cycle with a closed feedwater heater is considered. The thermal efficiency is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),



When the liquid enters the pump 10°C cooler than a saturated liquid at the condenser pressure, the enthalpies become

$$\left.\begin{array}{l} P_1 = 20\,\text{kPa} \\ T_1 = T_{\text{sat @ 20 kPa}} - 10 = 60.06 - 10 \cong 50°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f\,@\,50°\text{C}} = 209.34\,\text{kJ/kg} \\ \boldsymbol{\nu}_1 \cong \boldsymbol{\nu}_{f\,@\,50°\text{C}} = 0.001012\,\text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{\nu}_1 (P_2 - P_1) \\ &= (0.001012\,\text{m}^3/\text{kg})(3000 - 20)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) \\ &= 3.02\,\text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 209.34 + 3.02 = 212.36\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000\,\text{kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1\,\text{kJ/kg} \\ s_4 = 6.7450\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 1000\,\text{kPa} \\ s_{5s} = s_4 \end{array}\right\} \quad h_{5s} = 2851.9\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 20\,\text{kPa} \\ s_{6s} = s_4 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.7450 - 0.8320}{7.0752} = 0.8357 \\ h_{6s} = h_f + x_{6s} h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7\,\text{kJ/kg} \end{array}$$

$$\eta_{\text{T}} = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow h_5 = h_4 - \eta_{\text{T}}(h_4 - h_{5s}) = 3116.1 - (0.90)(3116.1 - 2851.9) = 2878.3\,\text{kJ/kg}$$

$$\eta_{\text{T}} = \frac{h_4 - h_6}{h_4 - h_{6s}} \longrightarrow h_6 = h_4 - \eta_{\text{T}}(h_4 - h_{6s}) = 3116.1 - (0.90)(3116.1 - 2221.7) = 2311.1\,\text{kJ/kg}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.

$$\left.\begin{array}{l} P_7 = 1000\,\text{kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 762.51\,\text{kJ/kg} \\ T_7 = 179.9°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 3000\,\text{kPa} \\ T_3 = T_7 = 209.9°\text{C} \end{array}\right\} \quad h_3 = 763.53\,\text{kJ/kg}$$

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An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$
$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 212.36}{2878.3 - 762.51} = 0.2605$$

Then,

$$w_{\text{T,out}} = h_4 - h_5 + (1 - y)(h_5 - h_6) = 3116.1 - 2878.3 + (1 - 0.2605)(2878.3 - 2311.1) = 657.2 \text{ kJ/kg}$$
$$w_{\text{P,in}} = 3.03 \text{ kJ/kg}$$
$$q_{\text{in}} = h_4 - h_3 = 3116.1 - 763.53 = 2353 \text{ kJ/kg}$$

Also,

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 657.2 - 3.03 = 654.2 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{654.2}{2353} = \mathbf{0.2781}$$

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10-57 The effect of pressure drop and non-isentropic turbine on the rate of heat input is to be determined for a given power plant.

Analysis The EES program used to solve this problem as well as the solutions are given below.

```
"Given"
P[3]=3000 [kPa]
DELTAP_boiler=10 [kPa]
P[4]=P[3]-DELTAP_boiler
T[4]=350 [C]
P[5]=1000 [kPa]
P[6]=20 [kPa]
eta_T=0.90

P[2]=P[3]
P[7]=P[5]
P[1]=P[6]

"Analysis"
Fluid$='steam_iapws'
"(a)"
"pump I"
x[1]=0
h[1]=enthalpy(Fluid$, P=P[1], x=x[1])
v[1]=volume(Fluid$, P=P[1], x=x[1])
w_p_in=v[1]*(P[2]-P[1])
h[2]=h[1]+w_p_in
"turbine"
h[4]=enthalpy(Fluid$, P=P[4], T=T[4])
s[4]=entropy(Fluid$, P=P[4], T=T[4])
s[5]=s[4]
h_s[5]=enthalpy(Fluid$, P=P[5], s=s[5])
T[5]=temperature(Fluid$, P=P[5], s=s[5])
x_s[5]=quality(Fluid$, P=P[5], s=s[5])
s[6]=s[4]
h_s[6]=enthalpy(Fluid$, P=P[6], s=s[6])
x_s[6]=quality(Fluid$, P=P[6], s=s[6])

h[5]=h[4]-eta_T*(h[4]-h_s[5])
h[6]=h[4]-eta_T*(h[4]-h_s[6])
x[5]=quality(Fluid$, P=P[5], h=h[5])
x[6]=quality(Fluid$, P=P[6], h=h[6])

"closed feedwater heater"
x[7]=0
h[7]=enthalpy(Fluid$, P=P[7], x=x[7])
T[7]=temperature(Fluid$, P=P[7], x=x[7])
T[3]=T[7]
h[3]=enthalpy(Fluid$, P=P[3], T=T[3])
y=(h[3]-h[2])/(h[5]-h[7])  "y=m_dot_5/m_dot_4"

"cycle"
q_in=h[4]-h[3]
w_T_out=h[4]-h[5]+(1-y)*(h[5]-h[6])
w_net=w_T_out-w_p_in
Eta_th=w_net/q_in
```

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Solution with 10 kPa pressure drop in the boiler:

DELTAP_boiler=10 [kPa]	eta_T=0.9
Eta_th=0.2827	Fluid$='steam_iapws'
P[3]=3000 [kPa]	P[4]=2990 [kPa]
q_in=2352.8 [kJ/kg]	w_net=665.1 [kJ/kg]
w_p_in=3.031 [m^3-kPa/kg]	w_T_out=668.1 [kJ/kg]
y=0.2405	

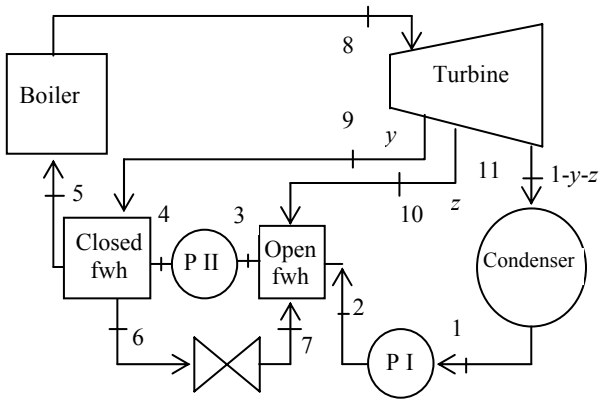
Solution without any pressure drop in the boiler:

DELTAP_boiler=0 [kPa]	eta_T=1
Eta_th=0.3136	Fluid$='steam_iapws'
P[3]=3000 [kPa]	P[4]=3000 [kPa]
q_in=2352.5 [kJ/kg]	w_net=737.8 [kJ/kg]
w_p_in=3.031 [m^3-kPa/kg]	w_T_out=740.9 [kJ/kg]
y=0.2437	

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10-58 A steam power plant operates on an ideal regenerative Rankine cycle with two feedwater heaters, one closed and one open. The mass flow rate of steam through the boiler for a net power output of 400 MW and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

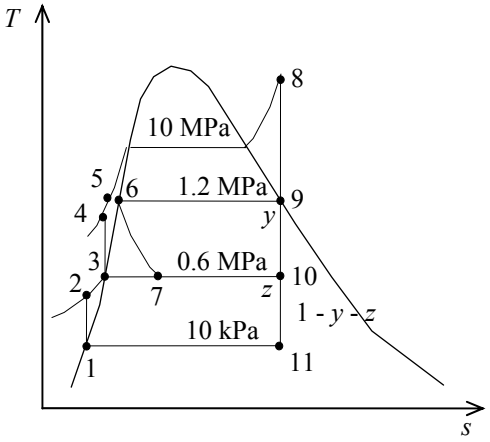


Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\nu_1 = \nu_{f@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$w_{pI,\text{in}} = \nu_1(P_2 - P_1)$$
$$= (0.00101\ \text{m}^3/\text{kg})(600 - 10\ \text{kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.60\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 191.81 + 0.60 = 192.40\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.6\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.3\text{ MPa}} = 670.38\text{ kJ/kg} \\ \nu_3 = \nu_{f@\,0.3\text{ MPa}} = 0.001101\ \text{m}^3/\text{kg} \end{array}$$
$$w_{pII,\text{in}} = \nu_3(P_4 - P_3)$$
$$= (0.001101\ \text{m}^3/\text{kg})(10000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.35\text{ kJ/kg}$$
$$h_4 = h_3 + w_{pII,\text{in}} = 670.38 + 10.35 = 680.73\text{ kJ/kg}$$



$$\left.\begin{array}{l} P_6 = 1.2\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_6 = h_7 = h_{f@\,1.2\text{ MPa}} = 798.33\text{ kJ/kg} \\ T_6 = T_{\text{sat@ }1.2\text{ MPa}} = 188.0°\text{C} \end{array}$$

$$T_6 = T_5,\ P_5 = 10\text{ MPa} \rightarrow h_5 = 798.33\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 10\text{ MPa} \\ T_8 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_8 = 3625.8\text{ kJ/kg} \\ s_8 = 6.9045\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_9 = 1.2\text{ MPa} \\ s_9 = s_8 \end{array}\right\} h_9 = 2974.5\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{10} = 0.6\text{ MPa} \\ s_{10} = s_8 \end{array}\right\} h_{10} = 2820.9\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{11} = 10\text{ kPa} \\ s_{11} = s_8 \end{array}\right\} \begin{array}{l} x_{11} = \dfrac{s_{11} - s_f}{s_{fg}} = \dfrac{6.9045 - 0.6492}{7.4996} = 0.8341 \\ h_{11} = h_f + x_{11}h_{fg} = 191.81 + (0.8341)(2392.1) = 2187.0\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_9(h_9 - h_6) = \dot{m}_5(h_5 - h_4) \longrightarrow y(h_9 - h_6) = (h_5 - h_4)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_{10}/\dot{m}_5$). Solving for y,

$$y = \frac{h_5 - h_4}{h_9 - h_6} = \frac{798.33 - 680.73}{2974.5 - 798.33} = 0.05404$$

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For the open FWH,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_7 h_7 + \dot{m}_2 h_2 + \dot{m}_{10} h_{10} = \dot{m}_3 h_3 \longrightarrow y h_7 + (1 - y - z) h_2 + z h_{10} = (1) h_3$$

where z is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for z,

$$z = \frac{(h_3 - h_2) - y(h_7 - h_2)}{h_{10} - h_2} = \frac{670.38 - 192.40 - (0.05404)(798.33 - 192.40)}{2820.9 - 192.40} = 0.1694$$

Then,

$$q_{\text{in}} = h_8 - h_5 = 3625.8 - 798.33 = 2827 \text{ kJ/kg}$$

$$q_{\text{out}} = (1 - y - z)(h_{11} - h_1) = (1 - 0.05404 - 0.1694)(2187.0 - 191.81) = 1549 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2827 - 1549 = 1278 \text{ kJ/kg}$$

and

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{400{,}000 \text{ kJ/s}}{1278 \text{ kJ/kg}} = \mathbf{313.0 \text{ kg/s}}$$

(*b*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1549 \text{ kJ/kg}}{2827 \text{ kJ/kg}} = 0.452 = \mathbf{45.2\%}$$

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10-59 Problem 10-58 is reconsidered. The effects of turbine and pump efficiencies on the mass flow rate and thermal efficiency are to be investigated. Also, the *T-s* diagram is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[8] = 10000 [kPa]
T[8] = 600 [C]
P[9] = 1200 [kPa]
P_cfwh=600 [kPa]
P[10] = P_cfwh
P_cond=10 [kPa]
P[11] = P_cond
W_dot_net=400 [MW]*Convert(MW, kW)
Eta_turb= 100/100 "Turbine isentropic efficiency"
Eta_turb_hp = Eta_turb "Turbine isentropic efficiency for high pressure stages"
Eta_turb_ip = Eta_turb "Turbine isentropic efficiency for intermediate pressure stages"
Eta_turb_lp = Eta_turb "Turbine isentropic efficiency for low pressure stages"
Eta_pump = 100/100 "Pump isentropic efficiency"

"Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
P[1] = P[11]
P[2]=P[10]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Open Feedwater Heater analysis"
z*h[10] + y*h[7] + (1-y-z)*h[2] = 1*h[3] "Steady-flow conservation of energy"
h[3]=enthalpy(Fluid$,P=P[3],x=0)
T[3]=temperature(Fluid$,P=P[3],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[3]=entropy(Fluid$,P=P[3],x=0)

"Boiler condensate pump  or Pump 2 analysis"
P[5]=P[8]
P[4] = P[5]
P[3]=P[10]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])

"Closed Feedwater Heater analysis"
P[6]=P[9]
 y*h[9] + 1*h[4] = 1*h[5] + y*h[6] "Steady-flow conservation of energy"
h[5]=enthalpy(Fluid$,P=P[6],x=0) "h[5] = h(T[5], P[5]) where T[5]=Tsat at P[9]"
T[5]=temperature(Fluid$,P=P[5],h=h[5]) "Condensate leaves heater as sat. liquid at P[6]"
s[5]=entropy(Fluid$,P=P[6],h=h[5])
h[6]=enthalpy(Fluid$,P=P[6],x=0)
T[6]=temperature(Fluid$,P=P[6],x=0) "Condensate leaves heater as sat. liquid at P[6]"
```

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```
s[6]=entropy(Fluid$,P=P[6],x=0)

"Trap analysis"
P[7] = P[10]
y*h[6] = y*h[7]  "Steady-flow conservation of energy for the trap operating as a throttle"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])

"Boiler analysis"
q_in + h[5]=h[8]"SSSF conservation of energy for the Boiler"
h[8]=enthalpy(Fluid$, T=T[8], P=P[8])
s[8]=entropy(Fluid$, T=T[8], P=P[8])

"Turbine analysis"
ss[9]=s[8]
hs[9]=enthalpy(Fluid$,s=ss[9],P=P[9])
Ts[9]=temperature(Fluid$,s=ss[9],P=P[9])
h[9]=h[8]-Eta_turb_hp*(h[8]-hs[9])"Definition of turbine efficiency for high pressure stages"
T[9]=temperature(Fluid$,P=P[9],h=h[9])
s[9]=entropy(Fluid$,P=P[9],h=h[9])
ss[10]=s[8]
hs[10]=enthalpy(Fluid$,s=ss[10],P=P[10])
Ts[10]=temperature(Fluid$,s=ss[10],P=P[10])
h[10]=h[9]-Eta_turb_ip*(h[9]-hs[10])"Definition of turbine efficiency for Intermediate pressure stages"
T[10]=temperature(Fluid$,P=P[10],h=h[10])
s[10]=entropy(Fluid$,P=P[10],h=h[10])
ss[11]=s[8]
hs[11]=enthalpy(Fluid$,s=ss[11],P=P[11])
Ts[11]=temperature(Fluid$,s=ss[11],P=P[11])
h[11]=h[10]-Eta_turb_lp*(h[10]-hs[11])"Definition of turbine efficiency for low pressure stages"
T[11]=temperature(Fluid$,P=P[11],h=h[11])
s[11]=entropy(Fluid$,P=P[11],h=h[11])
h[8] =y*h[9] + z*h[10] + (1-y-z)*h[11] + w_turb "SSSF conservation of energy for turbine"

"Condenser analysis"
(1-y-z)*h[11]=q_out+(1-y-z)*h[1]"SSSF First Law for the Condenser"

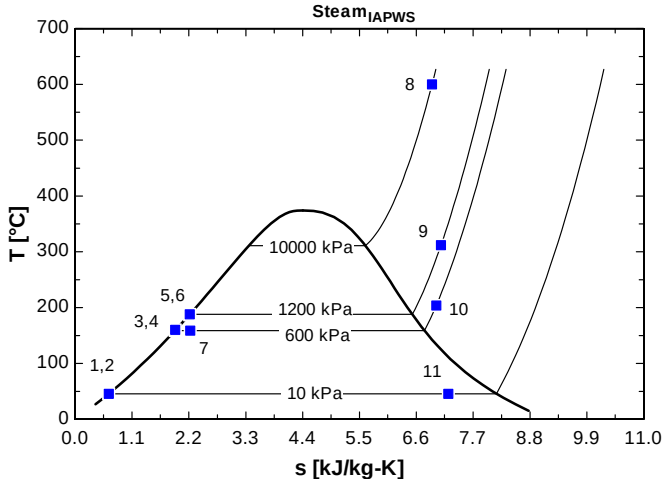
"Cycle Statistics"
w_net=w_turb - ((1-y-z)*w_pump1+ w_pump2)
Eta_th=w_net/q_in
W_dot_net = m_dot * w_net
```

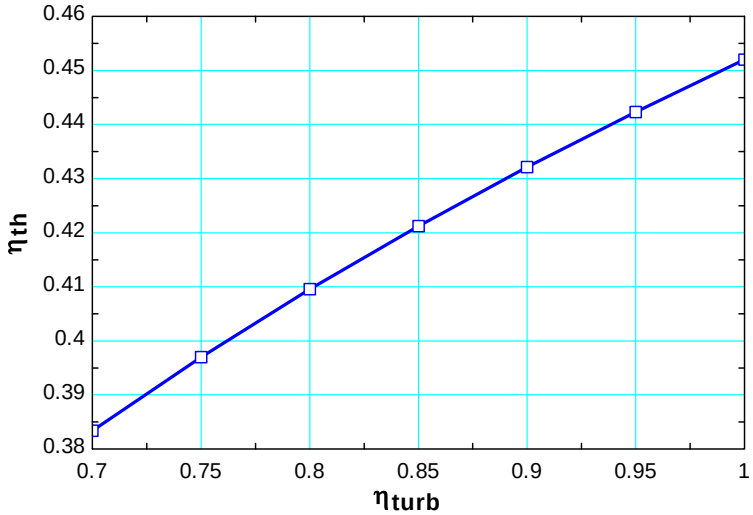
η_{turb}	η_{th}	m [kg/s]
0.7	0.3834	369
0.75	0.397	356.3
0.8	0.4096	345.4
0.85	0.4212	335.8
0.9	0.4321	327.4
0.95	0.4423	319.8
1	0.452	313

η_{pump}	η_{th}	m [kg/s]
0.7	0.4509	313.8
0.75	0.4511	313.6
0.8	0.4513	313.4
0.85	0.4515	313.3
0.9	0.4517	313.2
0.95	0.4519	313.1
1	0.452	313

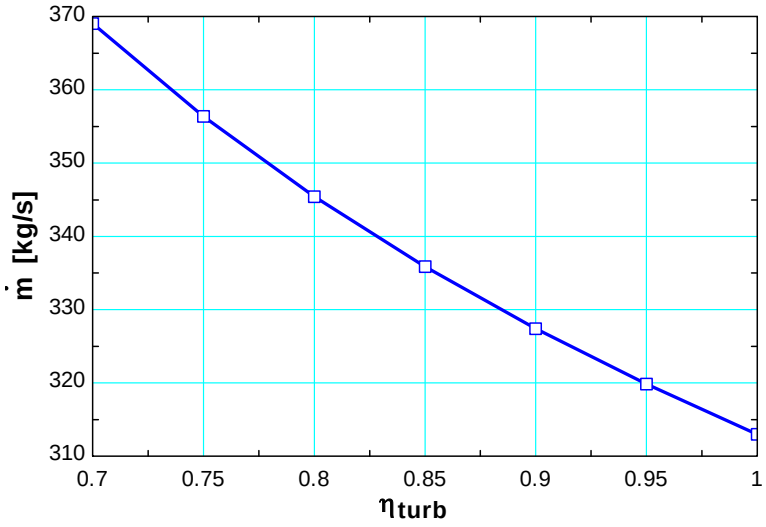
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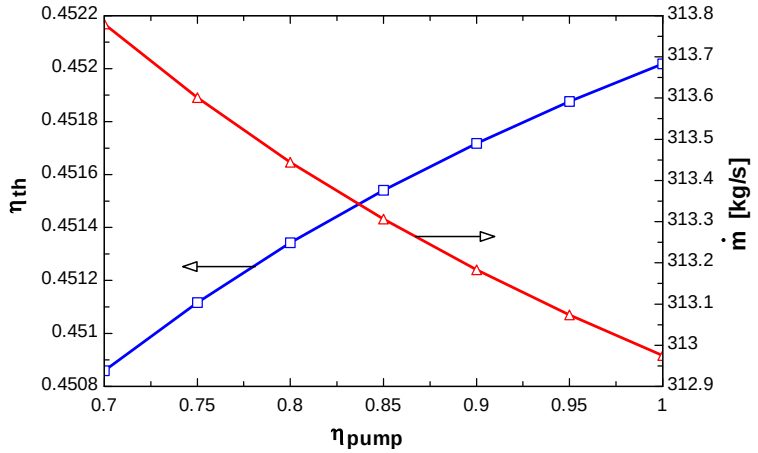
P_{cfwh} [kPa]	y	z
100	0.1702	0.05289
200	0.1301	0.09634
300	0.1041	0.1226
400	0.08421	0.1418
500	0.06794	0.1569
600	0.05404	0.1694
700	0.04182	0.1801
800	0.03088	0.1895
900	0.02094	0.1979
1000	0.01179	0.2054

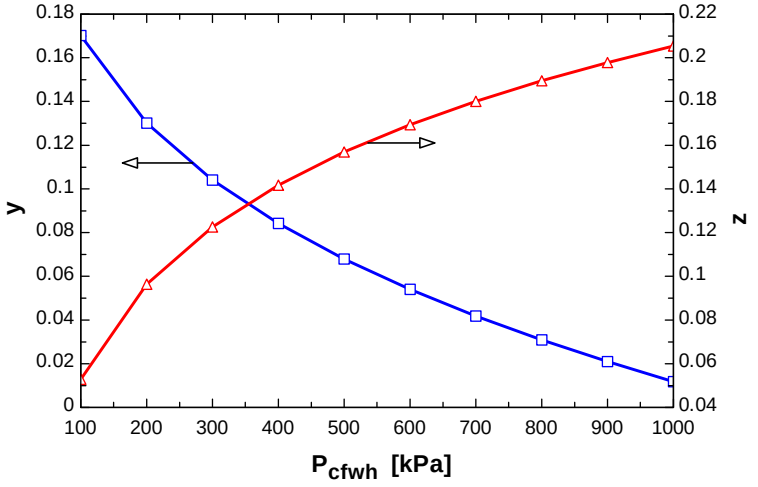




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10-60 A steam power plant that operates on an ideal regenerative Rankine cycle with a closed feedwater heater is considered. The temperature of the steam at the inlet of the closed feedwater heater, the mass flow rate of the steam extracted from the turbine for the closed feedwater heater, the net power output, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

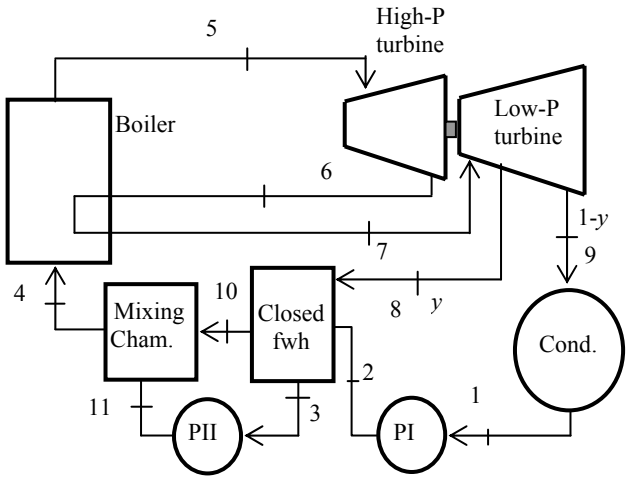
Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,20\text{ kPa}} = 251.42\text{ kJ/kg}$$

$$\nu_1 = \nu_{f@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$w_{pI,\text{in}} = \nu_1(P_2 - P_1)/\eta_p$$
$$= (0.001017\ \text{m}^3/\text{kg})(8000 - 20\text{ kPa})\frac{1}{0.88}$$
$$= 9.22\text{ kJ/kg}$$

$$h_2 = h_1 + w_{pI,\text{in}}$$
$$= 251.42 + 9.223$$
$$= 260.65\text{ kJ/kg}$$



$$\left.\begin{array}{l} P_3 = 1\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,1\text{ MPa}} = 762.51\text{ kJ/kg} \\ \nu_3 = \nu_{f@\,1\text{ MPa}} = 0.001127\ \text{m}^3/\text{kg} \end{array}$$

$$w_{pII,\text{in}} = \nu_3(P_{11} - P_3)/\eta_p$$
$$= (0.001127\ \text{m}^3/\text{kg})(8000 - 1000\text{ kPa})/0.88$$
$$= 8.97\text{ kJ/kg}$$

$$h_{11} = h_3 + w_{pII,\text{in}} = 762.51 + 8.97 = 771.48\text{ kJ/kg}$$

Also, $h_4 = h_{10} = h_{11} = 771.48$ kJ/kg since the two fluid streams which are being mixed have the same enthalpy.

$$\left.\begin{array}{l} P_5 = 8\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3399.5\text{ kJ/kg} \\ s_5 = 6.7266\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 3\text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_{6s} = 3104.7\text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T(h_5 - h_{6s})$$
$$= 3399.5 - (0.88)(3399.5 - 3104.7) = 3140.1\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 3\text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3457.2\text{ kJ/kg} \\ s_7 = 7.2359\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 1\text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_{8s} = 3117.1\text{ kJ/kg}$$

$$\eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} \longrightarrow h_8 = h_7 - \eta_T(h_7 - h_{8s})$$
$$= 3457.2 - (0.88)(3457.2 - 3117.1) = 3157.9\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 1\text{ MPa} \\ h_8 = 3157.9\text{ kJ/kg} \end{array}\right\} T_8 = \mathbf{349.9°C}$$

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$$\left.\begin{array}{l} P_9 = 20\ \text{kPa} \\ s_9 = s_7 \end{array}\right\} h_{9s} = 2385.2\ \text{kJ/kg}$$

$$\eta_T = \frac{h_7 - h_9}{h_7 - h_{9s}} \longrightarrow h_9 = h_7 - \eta_T (h_7 - h_{9s})$$
$$= 3457.2 - (0.88)(3457.2 - 2385.2) = 2513.9\ \text{kJ/kg}$$

The fraction of steam extracted from the low pressure turbine for closed feedwater heater is determined from the steady-flow energy balance equation applied to the feedwater heater. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$(1 - y)(h_{10} - h_2) = y(h_8 - h_3)$$
$$(1 - y)(771.48 - 260.65) = y(3157.9 - 762.51) \longrightarrow y = 0.1758$$

The corresponding mass flow rate is

$$\dot{m}_8 = y\dot{m}_5 = (0.1758)(15\ \text{kg/s}) = \mathbf{2.637\ kg/s}$$

(*c*) Then,

$$q_{\text{in}} = h_5 - h_4 + h_7 - h_6 = 3399.5 - 771.48 + 3457.2 - 3140.1 = 2945.2\ \text{kJ/kg}$$
$$q_{\text{out}} = (1 - y)(h_9 - h_1) = (1 - 0.1758)(2513.9 - 251.42) = 1864.8\ \text{kJ/kg}$$

and

$$\dot{W}_{\text{net}} = \dot{m}(q_{\text{in}} - q_{\text{out}}) = (15\ \text{kg/s})(2945.8 - 1864.8)\text{kJ/kg} = \mathbf{16{,}206\ kW}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1864.8\ \text{kJ/kg}}{2945.8\ \text{kJ/kg}} = 0.3668 = \mathbf{36.7\%}$$

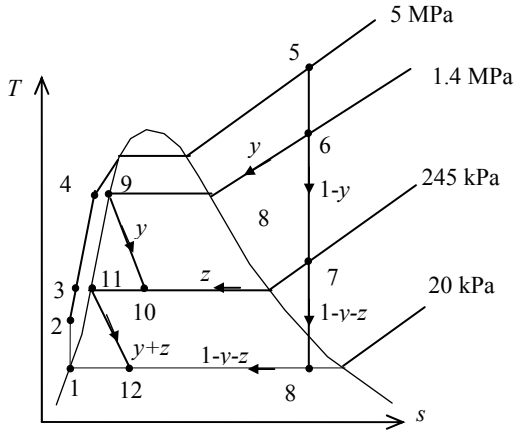
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10-61 A Rankine steam cycle modified with two closed feedwater heaters is considered. The *T-s* diagram for the ideal cycle is to be sketched. The fraction of mass extracted for the closed feedwater heater z and the cooling water flow rate are to be determined. Also, the net power output and the thermal efficiency of the plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*b*) Using the data from the problem statement, the enthalpies at various states are

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,20\text{ kPa}} = 0.00102\ \text{m}^3/\text{kg}$$
$$w_{\text{pI,in}} = \nu_1(P_2 - P_1)$$
$$= (0.00102\ \text{m}^3/\text{kg})(5000 - 20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 5.1\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 251 + 5.1 = 256.1\text{ kJ/kg}$$



Also,

$$h_3 = h_{11} = h_{f\,@\,245\text{ kPa}} = 533\text{ kJ/kg}$$
$$h_{12} = h_{11}\quad \text{(throttle valve operation)}$$
$$h_4 = h_9 = h_{f\,@\,1400\text{ kPa}} = 830\text{ kJ/kg}$$
$$h_{10} = h_9\quad \text{(throttle valve operation)}$$

An energy balance on the closed feedwater heater gives

$$1h_2 + zh_7 + yh_{10} = 1h_3 + (y+z)h_{11}$$

where *z* is the fraction of steam extracted from the low-pressure turbine. Solving for *z*,

$$z = \frac{(h_3 - h_2) + y(h_{11} - h_{10})}{h_7 - h_{11}} = \frac{(533 - 256.1) + (0.1153)(533 - 830)}{2918 - 533} = \mathbf{0.1017}$$

(c) An energy balance on the condenser gives

$$\dot{m}_8 h_8 + \dot{m}_w h_{w1} + \dot{m}_{12} h_{12} = \dot{m}_1 h_1 + \dot{m}_w h_{w2}$$
$$\dot{m}_w (h_{w2} - h_{w2}) = \dot{m}_8 h_8 + \dot{m}_{12} h_{12} - \dot{m}_1 h_1$$

Solving for the mass flow rate of cooling water, and substituting with correct units,

$$\dot{m}_w = \frac{\dot{m}_5\left[(1 - y - z)h_8 + (y+z)h_{12} - 1h_1\right]}{c_{pw}\Delta T_w}$$
$$= \frac{(50)\left[(1 - 0.1153 - 0.1017)(2477) + (0.1153 + 0.1017)(533) - 1(251)\right]}{(4.18)(10)}$$
$$= \mathbf{2158\ kg/s}$$

(d) The work output from the turbines is

$$w_{\text{T,out}} = h_5 - yh_6 - zh_7 - (1 - y - z)h_8$$
$$= 3900 - (0.1153)(3406) - (0.1017)(2918) - (1 - 0.1153 - 0.1017)(2477)$$
$$= 1271\text{ kJ/kg}$$

The net work output from the cycle is

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}}$$
$$= 1271 - 5.1 = 1265.9\text{ kJ/kg}$$

The net power output is

$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (50\text{ kg/s})(1265.9\text{ kJ/kg}) = 63{,}300\text{ kW} = \mathbf{63.3\ MW}$$

The rate of heat input in the boiler is

$$\dot{Q}_{\text{in}} = \dot{m}(h_5 - h_4) = (50\text{ kg/s})(3900 - 830)\text{ kJ/kg} = 153{,}500\text{ kW}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{63{,}300\text{ kW}}{153{,}500\text{ kW}} = 0.412 = \mathbf{41.2\%}$$

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Second-Law Analysis of Vapor Power Cycles

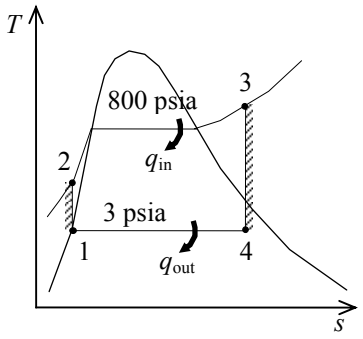
10-62C In the simple ideal Rankine cycle, irreversibilities occur during heat addition and heat rejection processes in the boiler and the condenser, respectively, and both are due to temperature difference. Therefore, the irreversibilities can be decreased and thus the 2nd law efficiency can be increased by minimizing the temperature differences during heat transfer in the boiler and the condenser. One way of doing that is regeneration.

10-63E The exergy destructions associated with each of the processes of the Rankine cycle described in Prob. 10-17E are to be determined for the specified source and sink temperatures.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Problem 10-17E,

$$s_1 = s_2 = s_{f@3\,\text{psia}} = 0.2009\ \text{Btu/lbm}\cdot\text{R}$$
$$s_3 = s_4 = 1.6413\ \text{Btu/lbm}\cdot\text{R}$$
$$q_{\text{in}} = h_3 - h_2 = 1456.0 - 111.81 = 1344.2\ \text{Btu/lbm}$$
$$q_{\text{out}} = h_4 - h_1 = 975.24 - 109.40 = 865.8\ \text{Btu/lbm}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 - \frac{q_{\text{in}}}{T_{\text{source}}}\right) = (500\ \text{R})\left(1.6413 - 0.2009 - \frac{1344.2\ \text{Btu/lbm}}{1960\ \text{R}}\right) = \mathbf{377\ Btu/lbm}$$

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{\text{out}}}{T_{\text{sink}}}\right) = (500\ \text{R})\left(0.2009 - 1.6413 + \frac{865.8\ \text{Btu/lbm}}{500\ \text{R}}\right) = \mathbf{146\ Btu/lbm}$$

Processes 1-2 and 3-4 are isentropic, and thus

$$x_{\text{destroyed},12} = \mathbf{0}$$
$$x_{\text{destroyed},34} = \mathbf{0}$$

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10-64 The exergy destruction associated with the heat rejection process in Prob. 10-21 is to be determined for the specified source and sink temperatures. The exergy of the steam at the boiler exit is also to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Problem 10-21,

$$s_1 = s_2 = s_{f@10\text{kPa}} = 0.6492 \text{ kJ/kg} \cdot \text{K}$$
$$s_3 = s_4 = 6.8000 \text{ kJ/kg} \cdot \text{K}$$
$$h_3 = 3411.4 \text{ kJ/kg}$$
$$q_{\text{out}} = 1961.8 \text{ kJ/kg}$$

The exergy destruction associated with the heat rejection process is

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (290 \text{ K})\left(0.6492 - 6.8000 + \frac{1961.8 \text{ kJ/kg}}{290 \text{ K}}\right) = \mathbf{178.0 \text{ kJ/kg}}$$

The exergy of the steam at the boiler exit is simply the flow exergy,

$$\psi_3 = (h_3 - h_0) - T_0(s_3 - s_0) + \frac{V_3^2}{2}^{\nearrow 0} + qz_3^{\nearrow 0}$$
$$= (h_3 - h_0) - T_0(s_3 - s_0)$$

where

$$h_0 = h_{@(290 \text{ K},\ 100 \text{ kPa})} \cong h_{f\ @\ 290 \text{ K}} = 71.95 \text{ kJ/kg}$$
$$s_0 = s_{@(290 \text{ K},100 \text{ kPa})} \cong s_{f\ @\ 290 \text{ K}} = 0.2533 \text{ kJ/kg} \cdot \text{K}$$

Thus,

$$\psi_3 = (3411.4 - 71.95) \text{ kJ/kg} - (290 \text{ K})(6.800 - 0.2532) \text{ kJ/kg} \cdot \text{K} = \mathbf{1440.9 \text{ kJ/kg}}$$

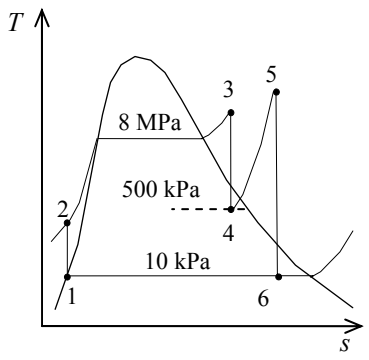
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10-65 The component of the ideal reheat Rankine cycle described in Prob. 10-33 with the largest exergy destruction is to be identified.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Prob. 10-33,

$$s_1 = s_2 = s_{f@10\,\text{kPa}} = 0.6492\,\text{kJ/kg}\cdot\text{K}$$
$$s_3 = s_4 = 6.5579\,\text{kJ/kg}\cdot\text{K}$$
$$s_5 = s_6 = 8.0893\,\text{kJ/kg}\cdot\text{K}$$
$$q_{\text{in, 2-3}} = h_3 - h_2 = 3273.3 - 199.88 = 3073.4\,\text{kJ/kg}$$
$$q_{\text{in, 4-5}} = h_5 - h_4 = 3485.4 - 2636.4 = 848.1\,\text{kJ/kg}$$
$$q_{\text{out}} = 2373.1\,\text{kJ/kg}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed, 23}} = T_0\left(s_3 - s_2 - \frac{q_{\text{in, 2-3}}}{T_{\text{source}}}\right) = (283\,\text{K})\left(6.5579 - 0.6492 - \frac{3073.4\,\text{kJ/kg}}{883\,\text{K}}\right) = \mathbf{687.1\,kJ/kg}$$

$$x_{\text{destroyed, 45}} = T_0\left(s_5 - s_4 - \frac{q_{\text{in, 4-5}}}{T_{\text{source}}}\right) = (283\,\text{K})\left(8.0893 - 6.5579 - \frac{848.1\,\text{kJ/kg}}{883\,\text{K}}\right) = \mathbf{161.6\,kJ/kg}$$

$$x_{\text{destroyed, 61}} = T_0\left(s_1 - s_6 + \frac{q_{\text{out}}}{T_{\text{sink}}}\right) = (283\,\text{K})\left(0.6492 - 8.0893 + \frac{2373.1\,\text{kJ/kg}}{283\,\text{K}}\right) = \mathbf{267.6\,kJ/kg}$$

Processes 1-2, 3-4, and 5-6 are isentropic, and thus,

$$x_{\text{destroyed,12}} = \mathbf{0}$$
$$x_{\text{destroyed, 34}} = \mathbf{0}$$
$$x_{\text{destroyed, 56}} = \mathbf{0}$$

The greatest exergy destruction occurs during heat addition process 2-3.

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10-66 The exergy destructions associated with each of the processes of the reheat Rankine cycle described in Prob. 10-35 are to be determined for the specified source and sink temperatures.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Problem 10-35,

$$s_1 = s_2 = s_{f@20\text{kPa}} = 0.8320 \text{ kJ/kg} \cdot \text{K}$$
$$s_3 = s_4 = 6.5432 \text{ kJ/kg} \cdot \text{K}$$
$$s_5 = s_6 = 7.1292 \text{ kJ/kg} \cdot \text{K}$$
$$q_{23,\text{in}} = 3178.3 - 257.50 = 2920.8 \text{ kJ/kg}$$
$$q_{45,\text{in}} = 3248.4 - 2901.0 = 347.3 \text{ kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2349.7 - 251.42 = 2098.3 \text{ kJ/kg}$$

Processes 1-2, 3-4, and 5-6 are isentropic. Thus, $i_{12} = i_{34} = i_{56} = \mathbf{0}$. Also,

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = (295 \text{ K})\left(6.5432 - 0.8320 + \frac{-2920.8 \text{ kJ/kg}}{1500 \text{ K}}\right) = \mathbf{1110 \text{ kJ/kg}}$$

$$x_{\text{destroyed},45} = T_0\left(s_5 - s_4 + \frac{q_{R,45}}{T_R}\right) = (295 \text{ K})\left(7.1292 - 6.5432 + \frac{-347.3 \text{ kJ/kg}}{1500 \text{ K}}\right) = \mathbf{104.6 \text{ kJ/kg}}$$

$$x_{\text{destroyed},61} = T_0\left(s_1 - s_6 + \frac{q_{R,61}}{T_R}\right) = (295 \text{ K})\left(0.8320 - 7.1292 + \frac{2098.3 \text{ kJ/kg}}{295 \text{ K}}\right) = \mathbf{240.6 \text{ kJ/kg}}$$

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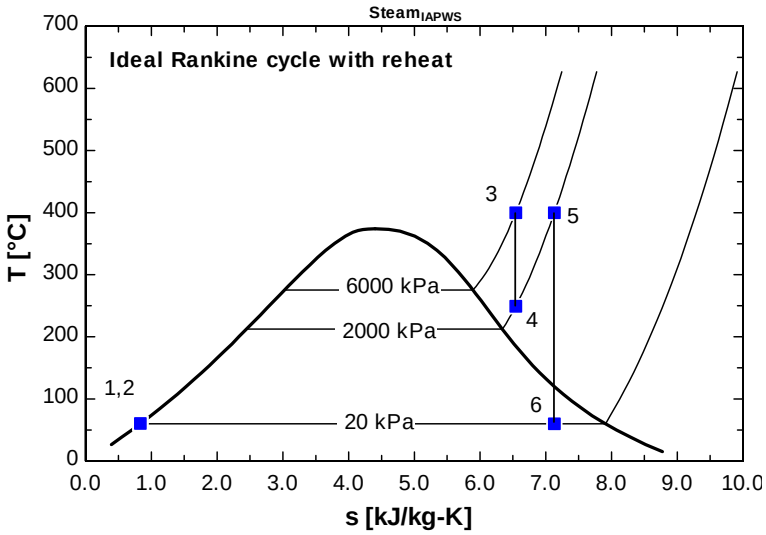
10-67 Problem 10-66 is reconsidered. The problem is to be solved by the diagram window data entry feature of EES by including the effects of the turbine and pump efficiencies. Also, the *T-s* diagram is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end
"Input Data - from diagram window"
{P[6] = 20 [kPa]
P[3] = 6000 [kPa]
T[3] = 400 [C]
P[4] = 2000 [kPa]
T[5] = 400 [C]
Eta_t = 100/100 "Turbine isentropic efficiency"
Eta_p = 100/100 "Pump isentropic efficiency"}
"Data for the irreversibility calculations:"
T_o = 295 [K]
T_R_L = 295 [K]
T_R_H = 1500 [K]
"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,T=T[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"
"Low Pressure Turbine analysis"
P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
```

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```
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])
"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eff=W_net/Q_in
"The irreversibilities (or exergy destruction) for each of the processes are:"
q_R_23 = - (h[3] - h[2]) "Heat transfer for the high temperature reservoir to process 2-3"
i_23 = T_o*(s[3] -s[2] + q_R_23/T_R_H)
q_R_45 = - (h[5] - h[4]) "Heat transfer for the high temperature reservoir to process 4-5"
i_45 = T_o*(s[5] -s[4] + q_R_45/T_R_H)
q_R_61 =  (h[6] - h[1]) "Heat transfer to the low temperature reservoir in process 6-1"
i_61 = T_o*(s[1] -s[6] + q_R_61/T_R_L)
i_34 = T_o*(s[4] -s[3])
i_56 = T_o*(s[6] -s[5])
i_12 = T_o*(s[2] -s[1])
```



SOLUTION

Eff=0.358	Eta_p=1
Eta_t=1	Fluid$='Steam_IAPWS'
i_12=0.007 [kJ/kg]	i_23=1110.378 [kJ/kg]
i_34=-0.000 [kJ/kg]	i_45=104.554 [kJ/kg]
i_56=0.000 [kJ/kg]	i_61=240.601 [kJ/kg]
Q_in=3268 [kJ/kg]	Q_out=2098 [kJ/kg]
q_R_23=-2921 [kJ/kg]	q_R_45=-347.3 [kJ/kg]
q_R_61=2098 [kJ/kg]	T_o=295 [K]
T_R_H=1500 [K]	T_R_L=295 [K]
W_net=1170 [kJ/kg]	W_p=6.083 [kJ/kg]
W_p_s=6.083 [kJ/kg]	W_t_hp=277.2 [kJ/kg]
W_t_lp=898.7 [kJ/kg]	

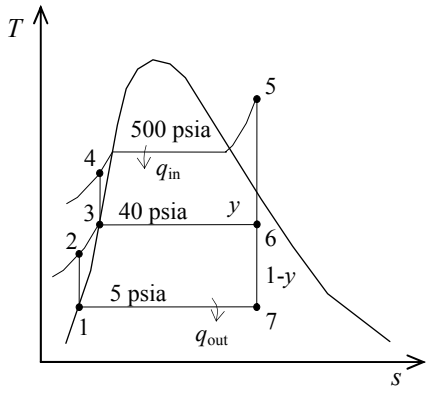
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10-68E The component of the ideal regenerative Rankine cycle described in Prob. 10-49E with the largest exergy destruction is to be identified.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Prob. 10-49E,

$$s_1 = s_2 = s_{f@5\,\text{psia}} = 0.23488\ \text{Btu/lbm}\cdot\text{R}$$
$$s_3 = s_4 = s_{f@40\,\text{psia}} = 0.39213\ \text{Btu/lbm}\cdot\text{R}$$
$$s_5 = s_6 = s_7 = 1.5590\ \text{Btu/lbm}\cdot\text{R}$$
$$q_{\text{in}} = h_5 - h_4 = 1298.6 - 237.6 = 1061\ \text{kJ/kg}$$
$$q_{\text{out}} = 732.1\ \text{Btu/lbm}$$
$$y = 0.1109$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed},45} = T_0\left(s_5 - s_4 - \frac{q_{\text{in}}}{T_{\text{source}}}\right) = (520\ \text{R})\left(1.5590 - 0.39213 - \frac{1061\ \text{Btu/lbm}}{1260\ \text{R}}\right) = \mathbf{168.9\ Btu/lbm}$$

$$x_{\text{destroyed},71} = T_0\left(s_1 - s_7 + \frac{q_{\text{out}}}{T_{\text{sink}}}\right) = (520\ \text{R})\left(0.23488 - 1.5590 + \frac{732.1\ \text{Btu/lbm}}{520\ \text{R}}\right) = \mathbf{43.6\ Btu/lbm}$$

For open feedwater heater, we have

$$\begin{aligned} x_{\text{destroyed, FWH}} &= T_0\left[s_3 - ys_6 - (1-y)s_2\right] \\ &= (520\ \text{R})\left[0.39213 - (0.1109)(1.5590) - (1-0.1109)(0.23488)\right] \\ &= \mathbf{5.4\ Btu/lbm} \end{aligned}$$

Processes 1-2, 3-4, 5-6, and 6-7 are isentropic, and thus

$$x_{\text{destroyed},12} = \mathbf{0} \qquad x_{\text{destroyed},56} = \mathbf{0}$$
$$x_{\text{destroyed},34} = \mathbf{0} \qquad x_{\text{destroyed},67} = \mathbf{0}$$

The greatest exergy destruction occurs during heat addition process 4-5.

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10-69 A single-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The power output from the turbine, the thermal efficiency of the plant, the exergy of the geothermal liquid at the exit of the flash chamber, and the exergy destructions and exergy efficiencies for the flash chamber, the turbine, and the entire plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) We use properties of water for geothermal water (Tables A-4, A-5, and A-6)

$$\left.\begin{matrix} T_1 = 230°\text{C} \\ x_1 = 0 \end{matrix}\right\} \begin{matrix} h_1 = 990.14\ \text{kJ/kg} \\ s_1 = 2.6100\ \text{kJ/kg.K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 500\ \text{kPa} \\ h_2 = h_1 = 990.14\ \text{kJ/kg} \end{matrix}\right\} \begin{matrix} x_2 = 0.1661 \\ s_2 = 2.6841\ \text{kJ/kg.K} \end{matrix}$$

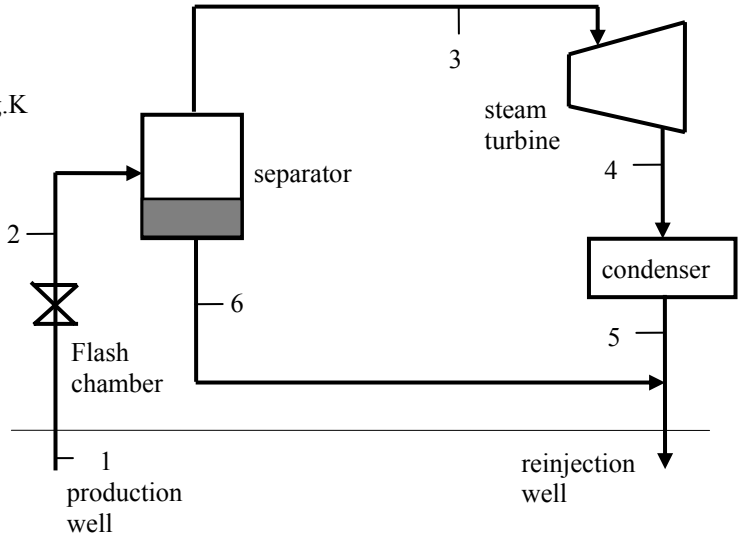
$$\begin{aligned} \dot{m}_3 &= x_2 \dot{m}_1 \\ &= (0.1661)(230\ \text{kg/s}) \\ &= 38.19\ \text{kg/s} \end{aligned}$$

$$\left.\begin{matrix} P_3 = 500\ \text{kPa} \\ x_3 = 1 \end{matrix}\right\} \begin{matrix} h_3 = 2748.1\ \text{kJ/kg} \\ s_3 = 6.8207\ \text{kJ/kg} \cdot \text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 10\ \text{kPa} \\ x_4 = 0.95 \end{matrix}\right\} \begin{matrix} h_4 = 2464.3\ \text{kJ/kg} \\ s_4 = 7.7739\ \text{kJ/kg} \cdot \text{K} \end{matrix}$$

$$\left.\begin{matrix} P_6 = 500\ \text{kPa} \\ x_6 = 0 \end{matrix}\right\} \begin{matrix} h_6 = 640.09\ \text{kJ/kg} \\ s_6 = 1.8604\ \text{kJ/kg} \cdot \text{K} \end{matrix}$$

$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 38.19 = 191.81\ \text{kg/s}$$



The power output from the turbine is

$$\dot{W}_{\text{T}} = \dot{m}_3 (h_3 - h_4) = (38.19\ \text{kJ/kg})(2748.1 - 2464.3)\text{kJ/kg} = \mathbf{10{,}842\ kW}$$

We use saturated liquid state at the standard temperature for dead state properties

$$\left.\begin{matrix} T_0 = 25°\text{C} \\ x_0 = 0 \end{matrix}\right\} \begin{matrix} h_0 = 104.83\ \text{kJ/kg} \\ s_0 = 0.3672\ \text{kJ/kg} \end{matrix}$$

$$\dot{E}_{\text{in}} = \dot{m}_1 (h_1 - h_0) = (230\ \text{kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622\ \text{kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{T,out}}}{\dot{E}_{\text{in}}} = \frac{10{,}842}{203{,}622} = 0.0532 = \mathbf{5.3\%}$$

(*b*) The specific exergies at various states are

$$\psi_1 = h_1 - h_0 - T_0 (s_1 - s_0) = (990.14 - 104.83)\text{kJ/kg} - (298\ \text{K})(2.6100 - 0.3672)\text{kJ/kg.K} = 216.53\ \text{kJ/kg}$$

$$\psi_2 = h_2 - h_0 - T_0 (s_2 - s_0) = (990.14 - 104.83)\text{kJ/kg} - (298\ \text{K})(2.6841 - 0.3672)\text{kJ/kg.K} = 194.44\ \text{kJ/kg}$$

$$\psi_3 = h_3 - h_0 - T_0 (s_3 - s_0) = (2748.1 - 104.83)\text{kJ/kg} - (298\ \text{K})(6.8207 - 0.3672)\text{kJ/kg.K} = 719.10\ \text{kJ/kg}$$

$$\psi_4 = h_4 - h_0 - T_0 (s_4 - s_0) = (2464.3 - 104.83)\text{kJ/kg} - (298\ \text{K})(7.7739 - 0.3672)\text{kJ/kg.K} = 151.05\ \text{kJ/kg}$$

$$\psi_6 = h_6 - h_0 - T_0 (s_6 - s_0) = (640.09 - 104.83)\text{kJ/kg} - (298\ \text{K})(1.8604 - 0.3672)\text{kJ/kg.K} = 89.97\ \text{kJ/kg}$$

The exergy of geothermal water at state 6 is

$$\dot{X}_6 = \dot{m}_6 \psi_6 = (191.81\ \text{kg/s})(89.97\ \text{kJ/kg}) = \mathbf{17{,}257\ kW}$$

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(*c*) Flash chamber:

$$\dot{X}_{\text{dest, FC}} = \dot{m}_1(\psi_1 - \psi_2) = (230\ \text{kg/s})(216.53 - 194.44)\text{kJ/kg} = \mathbf{5080\ kW}$$

$$\eta_{\text{II,FC}} = \frac{\psi_2}{\psi_1} = \frac{194.44}{216.53} = 0.898 = \mathbf{89.8\%}$$

(*d*) Turbine:

$$\dot{X}_{\text{dest,T}} = \dot{m}_3(\psi_3 - \psi_4) - \dot{W}_{\text{T}} = (38.19\ \text{kg/s})(719.10 - 151.05)\text{kJ/kg} - 10{,}842\ \text{kW} = \mathbf{10{,}854\ kW}$$

$$\eta_{\text{II,T}} = \frac{\dot{W}_{\text{T}}}{\dot{m}_3(\psi_3 - \psi_4)} = \frac{10{,}842\ \text{kW}}{(38.19\ \text{kg/s})(719.10 - 151.05)\text{kJ/kg}} = 0.500 = \mathbf{50.0\%}$$

(*e*) Plant:

$$\dot{X}_{\text{in,Plant}} = \dot{m}_1\psi_1 = (230\ \text{kg/s})(216.53\ \text{kJ/kg}) = 49{,}802\ \text{kW}$$

$$\dot{X}_{\text{dest,Plant}} = \dot{X}_{\text{in,Plant}} - \dot{W}_{\text{T}} = 49{,}802 - 10{,}842 = \mathbf{38{,}960\ kW}$$

$$\eta_{\text{II,Plant}} = \frac{\dot{W}_{\text{T}}}{\dot{X}_{\text{in,Plant}}} = \frac{10{,}842\ \text{kW}}{49{,}802\ \text{kW}} = 0.2177 = \mathbf{21.8\%}$$

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Cogeneration

10-70C The utilization factor of a cogeneration plant is the ratio of the energy utilized for a useful purpose to the total energy supplied. It could be unity for a plant that does not produce any power.

10-71C No. A cogeneration plant may involve throttling, friction, and heat transfer through a finite temperature difference, and still have a utilization factor of unity.

10-72C Yes, if the cycle involves no irreversibilities such as throttling, friction, and heat transfer through a finite temperature difference.

10-73C Cogeneration is the production of more than one useful form of energy from the same energy source. Regeneration is the transfer of heat from the working fluid at some stage to the working fluid at some other stage.

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10-74 A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The net power produced and the utilization factor of the plant are to be determined.

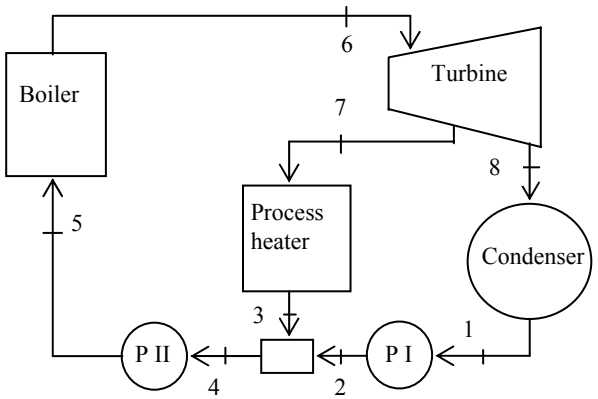
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \nu_1(P_2 - P_1)$$
$$= (0.00101\ \text{m}^3/\text{kg})(600 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.60\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.60 = 192.41\text{ kJ/kg}$$

$$h_3 = h_{f\,@\,0.6\text{ MPa}} = 670.38\text{ kJ/kg}$$



Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_4 h_4 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$

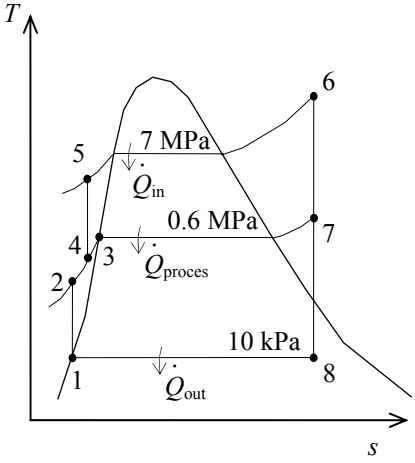
or,
$$h_4 = \frac{\dot{m}_2 h_2 + \dot{m}_3 h_3}{\dot{m}_4} = \frac{(22.50)(192.41) + (7.50)(670.38)}{30} = 311.90\text{ kJ/kg}$$
$$\nu_4 \cong \nu_{f\,@\,h_f = 311.90\text{ kJ/kg}} = 0.001026\ \text{m}^3/\text{kg}$$

$$w_{p\text{II,in}} = \nu_4(P_5 - P_4)$$
$$= (0.001026\ \text{m}^3/\text{kg})(7000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 6.57\text{ kJ/kg}$$
$$h_5 = h_4 + w_{p\text{II,in}} = 311.90 + 6.57 = 318.47\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 7\text{ MPa} \\ T_6 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_6 = 3411.4\text{ kJ/kg} \\ s_6 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_7 = 0.6\text{ MPa} \\ s_7 = s_6 \end{array}\right\} h_7 = 2774.6\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 10\text{ kPa} \\ s_8 = s_6 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{6.8000 - 0.6492}{7.4996} = 0.8201 \\ h_8 = h_f + x_8 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6\text{ kJ/kg} \end{array}$$



Then,

$$\dot{W}_{\text{T,out}} = \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8)$$
$$= (30\text{ kg/s})(3411.4 - 2774.6)\text{kJ/kg} + (22.5\text{ kg/s})(2774.6 - 2153.6)\text{kJ/kg} = 33{,}077\text{ kW}$$

$$\dot{W}_{\text{p,in}} = \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{p\text{II,in}} = (22.5\text{ kg/s})(0.60\text{ kJ/kg}) + (30\text{ kg/s})(6.57\text{ kJ/kg}) = 210.6\text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 33{,}077 - 210.6 = \mathbf{32{,}866\ kW}$$

Also, $$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (7.5\text{ kg/s})(2774.6 - 670.38)\text{ kJ/kg} = 15{,}782\text{ kW}$$

$$\dot{Q}_{\text{in}} = \dot{m}_5(h_6 - h_5) = (30\text{ kg/s})(3411.4 - 318.47) = 92{,}788\text{ kW}$$

and
$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{32{,}866 + 15{,}782}{92{,}788} = \mathbf{52.4\%}$$

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10-75E A large food-processing plant requires steam at a relatively high pressure, which is extracted from the turbine of a cogeneration plant. The rate of heat transfer to the boiler and the power output of the cogeneration plant are to be determined.

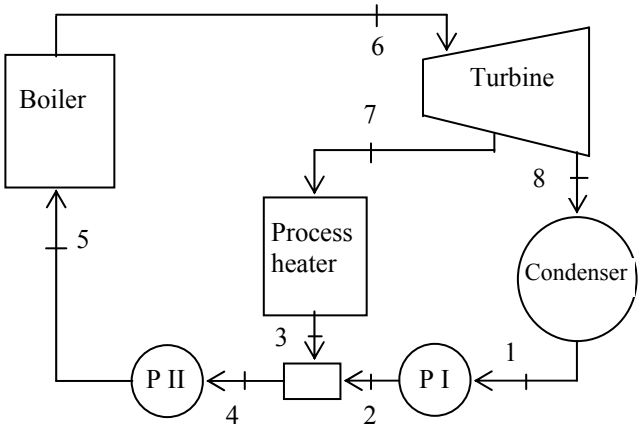
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f\,@\,2\text{ psia}} = 94.02 \text{ Btu/lbm}$$
$$\upsilon_1 = \upsilon_{f\,@\,2\text{ psia}} = 0.01623 \text{ ft}^3\text{/lbm}$$

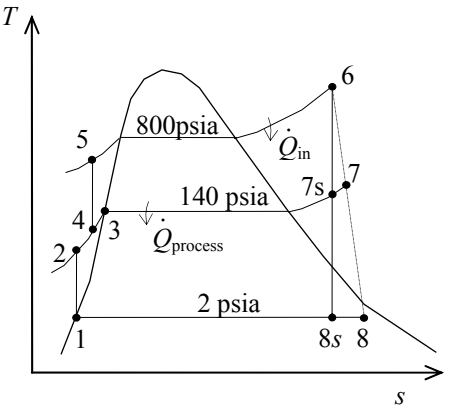
$$w_{\text{pI,in}} = \upsilon_1 (P_2 - P_1)/\eta_p$$
$$= \frac{1}{0.86}(0.01623 \text{ ft}^3\text{/lbm})(140-2)\text{psia} \times \left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right)$$
$$= 0.48 \text{ Btu/lbm}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 94.02 + 0.48 = 94.50 \text{ Btu/lbm}$$

$$h_3 = h_{f\,@\,140\text{ psia}} = 324.92 \text{ Btu/lbm}$$



Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad \longrightarrow \quad \dot{m}_4 h_4 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$



or,

$$h_4 = \frac{\dot{m}_2 h_2 + \dot{m}_3 h_3}{\dot{m}_4} = \frac{(8.5)(94.50) + (1.5)(324.92)}{10} = 129.07 \text{ Btu/lbm}$$
$$\upsilon_4 \cong \upsilon_{f\,@\,h_f = 129.07\text{ Btu/lbm}} = 0.01640 \text{ ft}^3\text{/lbm}$$

$$w_{\text{pII,in}} = \upsilon_4 (P_5 - P_4)/\eta_p$$
$$= (0.01640 \text{ ft}^3\text{/lbm})(800 - 140 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right)/(0.86)$$
$$= 2.33 \text{ Btu/lbm}$$
$$h_5 = h_4 + w_{pII,\text{in}} = 129.07 + 2.33 = 131.39 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_6 = 800 \text{ psia} \\ T_6 = 1000°\text{F} \end{array}\right\} \begin{array}{l} h_6 = 1512.2 \text{ Btu/lbm} \\ s_6 = 1.6812 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_{7s} = 140 \text{ psia} \\ s_{7s} = s_6 \end{array}\right\} h_{7s} = 1287.5 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_{8s} = 2 \text{ psia} \\ s_{8s} = s_6 \end{array}\right\} \begin{array}{l} x_{8s} = \dfrac{s_{8s} - s_f}{s_{fg}} = \dfrac{1.6812 - 0.17499}{1.74444} = 0.8634 \\ h_{8s} = h_f + x_{8s} h_{fg} = 94.02 + (0.8634)(1021.7) = 976.21 \text{ Btu/lbm} \end{array}$$

Then, $\dot{Q}_{\text{in}} = \dot{m}_5 (h_6 - h_5) = (10 \text{ lbm/s})(1512.2 - 131.39)\text{Btu/lbm} = \mathbf{13{,}810\ Btu/s}$

(*b*) $\dot{W}_{\text{T,out}} = \eta_T \dot{W}_{T,s} = \eta_T [\dot{m}_6 (h_6 - h_{7s}) + \dot{m}_8 (h_{7s} - h_{8s})]$
$= (0.86)[(10 \text{ lbm/s})(1512.2 - 1287.5) \text{ Btu/lbm} + (1.5 \text{ lbm/s})(1287.5 - 976.21) \text{ Btu/lbm}]$
$= 4208 \text{ Btu/s} = \mathbf{4440\ kW}$

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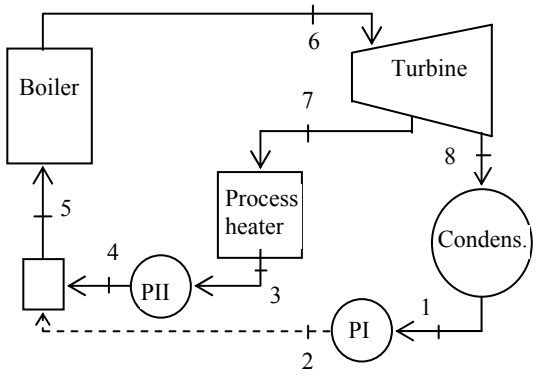
10-76 A cogeneration plant has two modes of operation. In the first mode, all the steam leaving the turbine at a relatively high pressure is routed to the process heater. In the second mode, 60 percent of the steam is routed to the process heater and remaining is expanded to the condenser pressure. The power produced and the rate at which process heat is supplied in the first mode, and the power produced and the rate of process heat supplied in the second mode are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

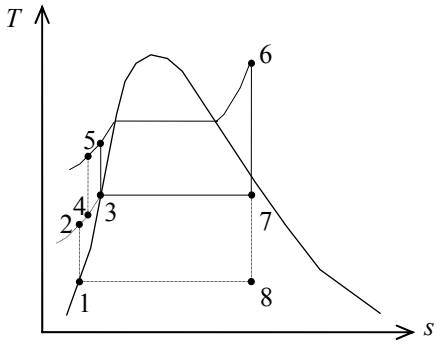
Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.42\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001017\ \text{m}^3/\text{kg})(10{,}000 - 20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.15\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 10.15 = 261.57\text{ kJ/kg}$$
$$h_3 = h_{f\,@\,0.5\text{ MPa}} = 640.09\ \text{kJ/kg}$$
$$\nu_3 = \nu_{f\,@\,0.5\text{ MPa}} = 0.001093\ \text{m}^3/\text{kg}$$



$$w_{\text{pII,in}} = \nu_3(P_4 - P_3)$$
$$= (0.001093\ \text{m}^3/\text{kg})(10{,}000 - 500\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.38\text{ kJ/kg}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.38 = 650.47\text{ kJ/kg}$$



Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}\overset{\nearrow 0\ \text{(steady)}}{} = 0 \ \rightarrow\ \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad \longrightarrow \quad \dot{m}_5 h_5 = \dot{m}_2 h_2 + \dot{m}_4 h_4$$

or,
$$h_5 = \frac{\dot{m}_2 h_2 + \dot{m}_4 h_4}{\dot{m}_5} = \frac{(2)(261.57) + (3)(650.47)}{5} = 494.91\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 10\text{ MPa} \\ T_6 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_6 = 3242.4\text{ kJ/kg} \\ s_6 = 6.4219\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_7 = 0.5\text{ MPa} \\ s_7 = s_6 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.4219 - 1.8604}{4.9603} = 0.9196 \\ h_7 = h_f + x_7 h_{fg} = 640.09 + (0.9196)(2108.0) = 2578.6\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_8 = 20\text{ kPa} \\ s_8 = s_6 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{6.4219 - 0.8320}{7.0752} = 0.7901 \\ h_8 = h_f + x_8 h_{fg} = 251.42 + (0.7901)(2357.5) = 2114.0\text{ kJ/kg} \end{array}$$

When the entire steam is routed to the process heater,

$$\dot{W}_{\text{T,out}} = \dot{m}_6(h_6 - h_7) = (5\text{ kg/s})(3242.4 - 2578.6)\text{kJ/kg} = \mathbf{3319\ kW}$$
$$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (5\text{ kg/s})(2578.6 - 640.09)\text{kJ/kg} = \mathbf{9693\ kW}$$

(*b*) When only 60% of the steam is routed to the process heater,

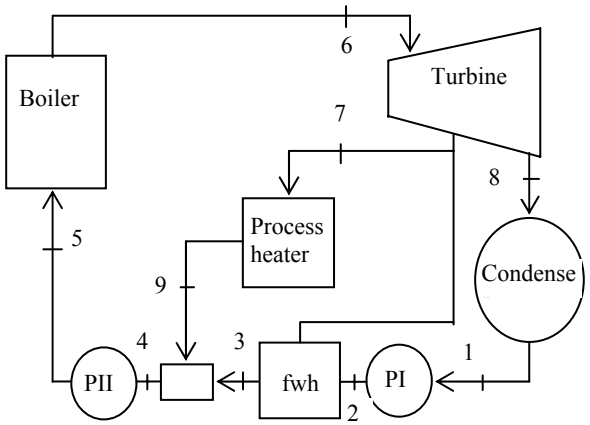
$$\dot{W}_{\text{T,out}} = \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8)$$
$$= (5\text{ kg/s})(3242.4 - 2578.6)\text{ kJ/kg} + (2\text{ kg/s})(2578.6 - 2114.0)\text{ kJ/kg} = \mathbf{4248\ kW}$$
$$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (3\text{ kg/s})(2578.6 - 640.09)\text{ kJ/kg} = \mathbf{5816\ kW}$$

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10-77 A cogeneration plant modified with regeneration is to generate power and process heat. The mass flow rate of steam through the boiler for a net power output of 25 MW is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),



$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\upsilon_1 = \upsilon_{f\,@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \upsilon_1(P_2 - P_1) \\ &= (0.00101\ \text{m}^3/\text{kg})(1600 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 1.61\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 1.61 = 193.41\text{ kJ/kg}$$

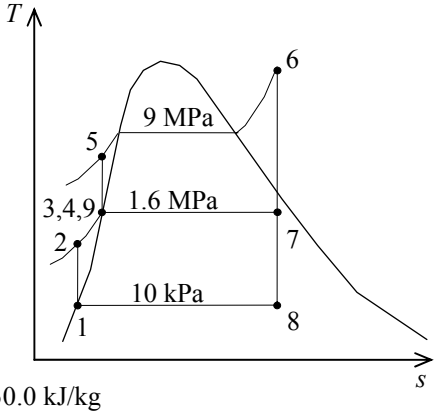
$$h_3 = h_4 = h_9 = h_{f\,@\,1.6\text{ MPa}} = 858.44\text{ kJ/kg}$$

$$\upsilon_4 = \upsilon_{f\,@\,1.6\text{ MPa}} = 0.001159\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pII,in}} &= \upsilon_4(P_5 - P_4) \\ &= (0.001159\ \text{m}^3/\text{kg})(9000 - 400\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 8.57\text{ kJ/kg} \end{aligned}$$

$$h_5 = h_4 + w_{\text{pII,in}} = 858.44 + 8.57 = 867.02\text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 9\text{ MPa} \\ T_6 &= 400°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3118.8\text{ kJ/kg} \\ s_6 &= 6.2876\text{ kJ/kg}\cdot\text{K} \end{aligned}$$



$$\left.\begin{aligned} P_7 &= 1.6\text{ MPa} \\ s_7 &= s_6 \end{aligned}\right\} \begin{aligned} x_7 &= \frac{s_7 - s_f}{s_{fg}} = \frac{6.2876 - 2.3435}{4.0765} = 0.9675 \\ h_7 &= h_f + x_7 h_{fg} = 858.44 + (0.9675)(1934.4) = 2730.0\text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_8 &= 10\text{ kPa} \\ s_8 &= s_6 \end{aligned}\right\} \begin{aligned} x_8 &= \frac{s_8 - s_f}{s_{fg}} = \frac{6.2876 - 0.6492}{7.4996} = 0.7518 \\ h_8 &= h_f + x_8 h_{fg} = 191.81 + (0.7518)(2392.1) = 1990.2\text{ kJ/kg} \end{aligned}$$

Then, per kg of steam flowing through the boiler, we have

$$\begin{aligned} w_{\text{T,out}} &= (h_6 - h_7) + (1 - y)(h_7 - h_8) \\ &= (3118.8 - 2730.0)\text{ kJ/kg} + (1 - 0.35)(2730.0 - 1990.2)\text{ kJ/kg} \\ &= 869.7\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{p,in}} &= (1 - y)w_{\text{pI,in}} + w_{\text{pII,in}} \\ &= (1 - 0.35)(1.61\text{ kJ/kg}) + (8.57\text{ kJ/kg}) \\ &= 9.62\text{ kJ/kg} \end{aligned}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{p,in}} = 869.7 - 9.62 = 860.1\text{ kJ/kg}$$

Thus,

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{25{,}000\text{ kJ/s}}{860.1\text{ kJ/kg}} = \mathbf{29.1\ kg/s}$$

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10-78 Problem 10-77 is reconsidered. The effect of the extraction pressure for removing steam from the turbine to be used for the process heater and open feedwater heater on the required mass flow rate is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
y = 0.35 "fraction of steam extracted from turbine for feedwater heater and process heater"
P[6] = 9000 [kPa]
T[6] = 400 [C]
P_extract=1600 [kPa]
P[7] = P_extract
P_cond=10 [kPa]
P[8] = P_cond
W_dot_net=25 [MW]*Convert(MW, kW)
Eta_turb= 100/100 "Turbine isentropic efficiency"
Eta_pump = 100/100 "Pump isentropic efficiency"
P[1] = P[8]
P[2]=P[7]
P[3]=P[7]
P[4] = P[7]
P[5]=P[6]
P[9] = P[7]

"Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'

h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Open Feedwater Heater analysis:"
z*h[7]  + (1- y)*h[2] = (1- y + z)*h[3] "Steady-flow conservation of energy"
h[3]=enthalpy(Fluid$,P=P[3],x=0)
T[3]=temperature(Fluid$,P=P[3],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[3]=entropy(Fluid$,P=P[3],x=0)

"Process heater analysis:"
(y - z)*h[7]  = q_process + (y - z)*h[9] "Steady-flow conservation of energy"
Q_dot_process = m_dot*(y - z)*q_process"[kW]"
h[9]=enthalpy(Fluid$,P=P[9],x=0)
T[9]=temperature(Fluid$,P=P[9],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[9]=entropy(Fluid$,P=P[9],x=0)

"Mixing chamber at 3, 4, and 9:"
(y-z)*h[9]  + (1-y+z)*h[3] = 1*h[4] "Steady-flow conservation of energy"
T[4]=temperature(Fluid$,P=P[4],h=h[4]) "Condensate leaves heater as sat. liquid at P[3]"
s[4]=entropy(Fluid$,P=P[4],h=h[4])

"Boiler condensate pump  or Pump 2 analysis"
v4=volume(Fluid$,P=P[4],x=0)
w_pump2_s=v4*(P[5]-P[4])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[4]+w_pump2= h[5]  "Steady-flow conservation of energy"
```

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```
s[5]=entropy(Fluid$,P=P[5],h=h[5])
T[5]=temperature(Fluid$,P=P[5],h=h[5])

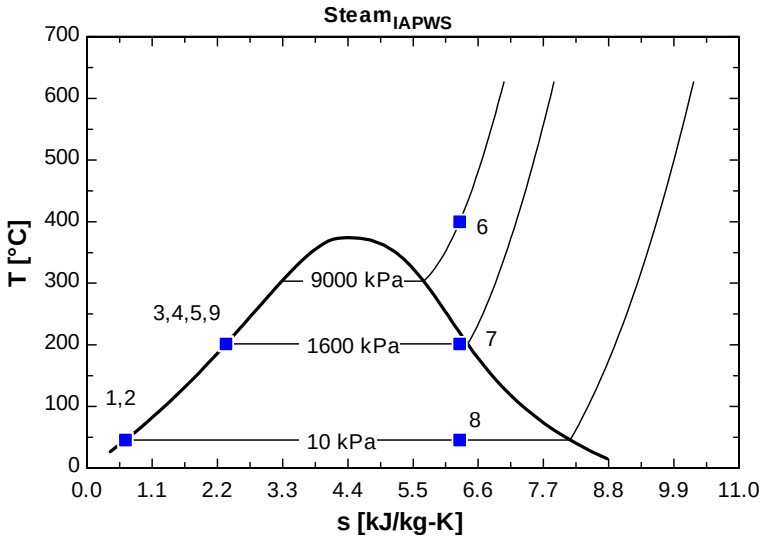
"Boiler analysis"
q_in + h[5]=h[6]"SSSF conservation of energy for the Boiler"
h[6]=enthalpy(Fluid$, T=T[6], P=P[6])
s[6]=entropy(Fluid$, T=T[6], P=P[6])

"Turbine analysis"
ss[7]=s[6]
hs[7]=enthalpy(Fluid$,s=ss[7],P=P[7])
Ts[7]=temperature(Fluid$,s=ss[7],P=P[7])
h[7]=h[6]-Eta_turb*(h[6]-hs[7])"Definition of turbine efficiency for high pressure stages"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
ss[8]=s[7]
hs[8]=enthalpy(Fluid$,s=ss[8],P=P[8])
Ts[8]=temperature(Fluid$,s=ss[8],P=P[8])
h[8]=h[7]-Eta_turb*(h[7]-hs[8])"Definition of turbine efficiency for low pressure stages"
T[8]=temperature(Fluid$,P=P[8],h=h[8])
s[8]=entropy(Fluid$,P=P[8],h=h[8])
h[6] =y*h[7] + (1- y)*h[8] + w_turb "SSSF conservation of energy for turbine"

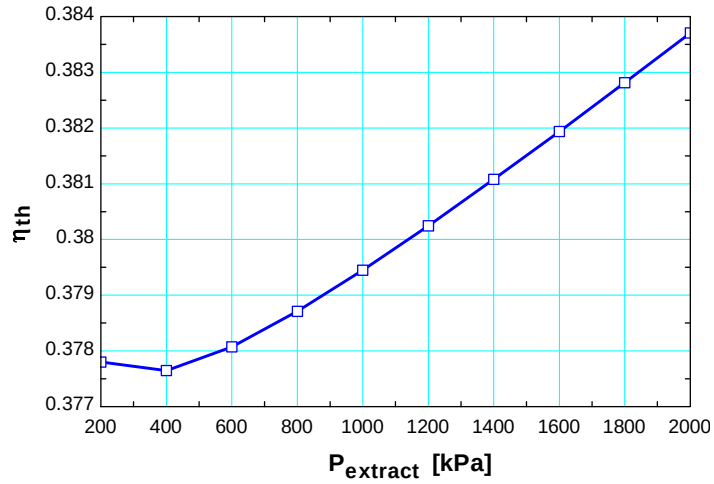
"Condenser analysis"
(1- y)*h[8]=q_out+(1- y)*h[1]"SSSF First Law for the Condenser"

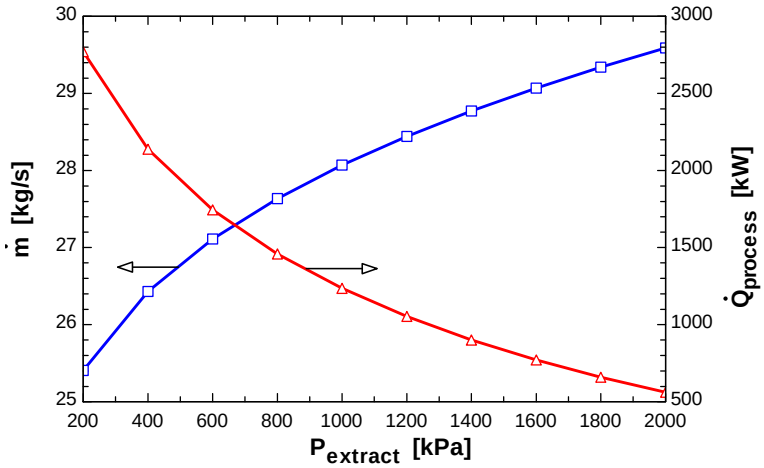
"Cycle Statistics"
w_net=w_turb - ((1- y)*w_pump1+ w_pump2)
Eta_th=w_net/q_in
W_dot_net = m_dot * w_net
```

$P_{extract}$ [kPa]	η_{th}	m [kg/s]	$Q_{process}$ [kW]
200	0.3778	25.4	2770
400	0.3776	26.43	2137
600	0.3781	27.11	1745
800	0.3787	27.63	1459
1000	0.3794	28.07	1235
1200	0.3802	28.44	1053
1400	0.3811	28.77	900.7
1600	0.3819	29.07	770.9
1800	0.3828	29.34	659
2000	0.3837	29.59	561.8



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10-79E A cogeneration plant is to generate power while meeting the process steam requirements for a certain industrial application. The net power produced, the rate of process heat supply, and the utilization factor of this plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 \cong h_{f\,@\,240°\text{F}} = 208.49 \text{ Btu/lbm}$$
$$h_2 \cong h_1$$

$$\left.\begin{matrix} P_3 = 600 \text{ psia} \\ T_3 = 800°\text{F} \end{matrix}\right\} \begin{matrix} h_3 = 1408.0 \text{ Btu/lbm} \\ s_3 = s_5 = s_7 = 1.6348 \text{ Btu/lbm} \cdot \text{R} \end{matrix}$$

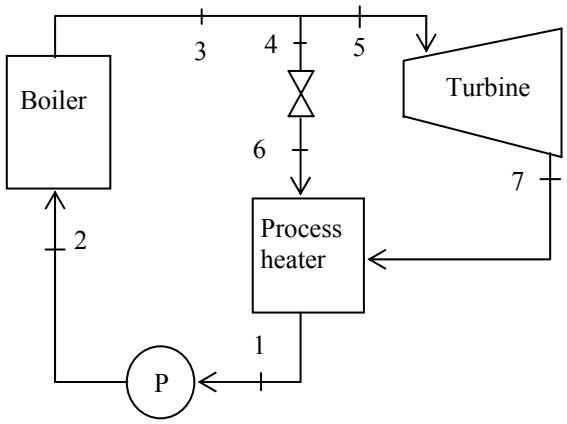
$$h_3 = h_4 = h_5 = h_6$$

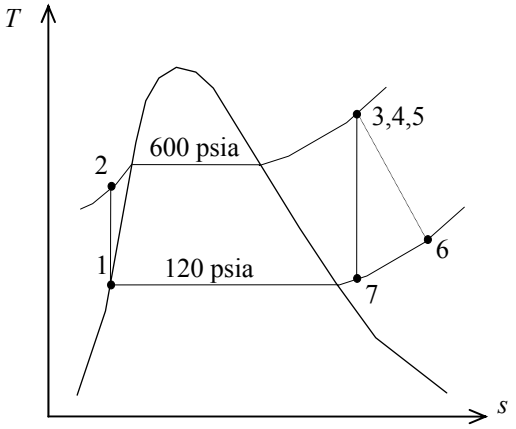
$$\left.\begin{matrix} P_7 = 120 \text{ psia} \\ s_7 = s_3 \end{matrix}\right\} h_7 = 1229.5 \text{ Btu/lbm}$$

$$\dot{W}_{\text{net}} = \dot{m}_5 (h_5 - h_7)$$
$$= (12 \text{ lbm/s})(1408.0 - 1229.5) \text{ Btu/lbm}$$
$$= 2142 \text{ Btu/s} = \mathbf{2260\ kW}$$

(*b*) $$\dot{Q}_{\text{process}} = \sum \dot{m}_i h_i - \sum \dot{m}_e h_e$$
$$= \dot{m}_6 h_6 + \dot{m}_7 h_7 - \dot{m}_1 h_1 -$$
$$= (6)(1408.0) + (12)(1229.5) - (18)(208.49)$$
$$= \mathbf{19{,}450\ Btu/s}$$

(*c*) $\varepsilon_u = \mathbf{1}$ since all the energy is utilized.





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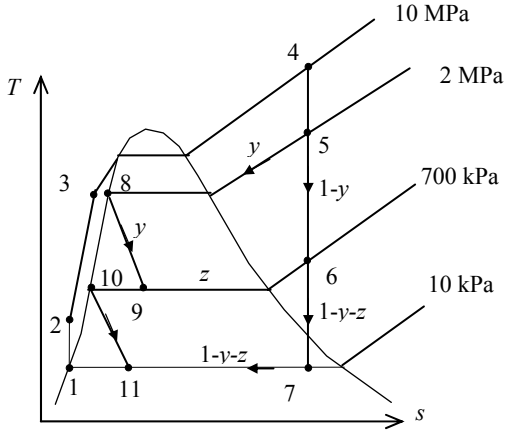
10-80 A Rankine steam cycle modified for a closed feedwater heater and a process heater is considered. The *T*-*s* diagram for the ideal cycle is to be sketched; the mass flow rate of the cooling water; and the utilization efficiency of the plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*b*) Using the data from the problem statement, the enthalpies at various states are

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,20\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$
$$w_{\text{pI,in}} = \nu_1(P_2 - P_1)$$
$$= (0.00101\text{ m}^3\text{/kg})(10000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.1\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 10.1 = 201.9\text{ kJ/kg}$$

$$h_3 = h_8 = h_9 = h_{f\,@\,2000\text{ kPa}} = 908.47\text{ kJ/kg}$$
$$h_{10} = h_{11} = h_{f\,@\,700\text{ kPa}} = 697.00\text{ kJ/kg}$$



An energy balance on the closed feedwater heater gives

$$y(h_5 - h_8) = h_3 - h_2$$
$$y = \frac{h_3 - h_2}{h_5 - h_8} = \frac{908.47 - 201.9}{2930 - 908.47} = 0.3495$$

The process heat is expressed as

$$\dot{Q}_{\text{process}} = z\dot{m}(h_6 - h_{10}) = \dot{m}_w c_p \Delta T_w$$
$$\dot{m}_w = \frac{z\dot{m}(h_6 - h_{10})}{c_p \Delta T_w} = \frac{0.05(100\text{ kg/s})(2714 - 697.00)\text{ kJ/kg}}{(4.18\text{ kJ/kg}\cdot°\text{C})(40°\text{C})} = \mathbf{60.3\text{ kg/s}}$$

(c) The net power output is determined from

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T}} - \dot{W}_{\text{P}}$$
$$= \dot{m}\left[y(h_4 - h_5) + z(h_4 - h_6) + (1 - y - z)(h_4 - h_7) - w_P\right]$$
$$= (100\text{ kg/s})\begin{bmatrix}0.3495(3374 - 2930)\text{ kJ/kg} + 0.05(3374 - 2714)\text{ kJ/kg} \\ +(1 - 0.3495 - 0.05)(3374 - 2089)\text{ kJ/kg} - (10.1\text{ kJ/kg})\end{bmatrix}$$
$$= 94{,}970\text{ kW}$$

The rate of heat input in the boiler is

$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = (100\text{ kg/s})(3874 - 908.47)\text{ kJ/kg} = 296{,}550\text{ kW}$$

The rate of process heat is

$$\dot{Q}_{\text{process}} = 0.05\dot{m}(h_6 - h_{10}) = 0.05(100\text{ kg/s})(2714 - 697.00)\text{ kJ/kg} = 10{,}085\text{ kW}$$

The utilization efficiency of this cogeneration plant is

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{(94{,}970 + 10{,}085)\text{ kW}}{296{,}550\text{ kW}} = 0.354 = \mathbf{35.4\%}$$

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Combined Gas-Vapor Power Cycles

10-81C The energy source of the steam is the waste energy of the exhausted combustion gases.

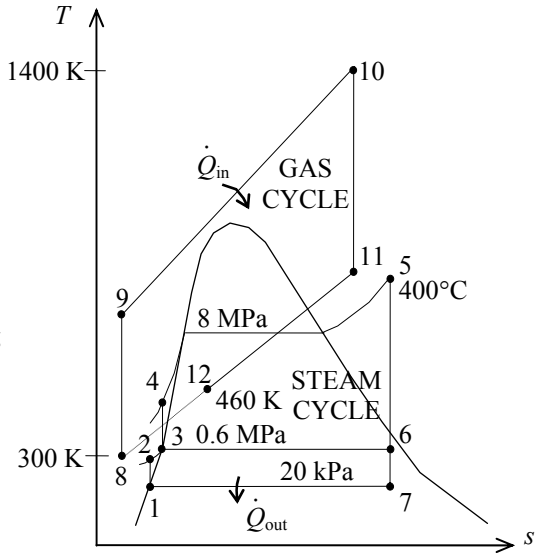
10-82C Because the combined gas-steam cycle takes advantage of the desirable characteristics of the gas cycle at high temperature, and those of steam cycle at low temperature, and combines them. The result is a cycle that is more efficient than either cycle executed operated alone.

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10-83 A 450-MW combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is an ideal Rankine cycle with an open feedwater heater. The mass flow rate of air to steam, the required rate of heat input in the combustion chamber, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) The analysis of gas cycle yields (Table A-17)



$$T_8 = 300\ \text{K} \longrightarrow \begin{matrix} h_8 = 300.19\ \text{kJ/kg} \\ P_{r_8} = 1.386 \end{matrix}$$

$$P_{r_9} = \frac{P_9}{P_8} P_{r_8} = (14)(1.386) = 19.40 \longrightarrow h_9 = 635.5\ \text{kJ/kg}$$

$$T_{10} = 1400\ \text{K} \longrightarrow \begin{matrix} h_{10} = 1515.42\ \text{kJ/kg} \\ P_{r_{10}} = 450.5 \end{matrix}$$

$$P_{r_{11}} = \frac{P_{11}}{P_{10}} P_{r_{10}} = \left(\frac{1}{14}\right)(450.5) = 32.18 \longrightarrow h_{11} = 735.8\ \text{kJ/kg}$$

$$T_{12} = 460\ \text{K} \longrightarrow h_{12} = 462.02\ \text{kJ/kg}$$

From the steam tables (Tables A-4, A-5, A-6),

$$h_1 = h_{f\ @\ 20\ \text{kPa}} = 251.42\ \text{kJ/kg}$$
$$\upsilon_1 = \upsilon_{f\ @\ 20\ \text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \upsilon_1 (P_2 - P_1) \\ &= (0.001017\ \text{m}^3/\text{kg})(600 - 20\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 0.59\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 0.59 = 252.01\ \text{kJ/kg}$$

$$h_3 = h_{f\ @\ 0.6\ \text{MPa}} = 670.38\ \text{kJ/kg}$$
$$\upsilon_3 = \upsilon_{f\ @\ 0.6\ \text{MPa}} = 0.001101\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pII,in}} &= \upsilon_3 (P_4 - P_3) \\ &= (0.001101\ \text{m}^3/\text{kg})(8{,}000 - 600\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 8.15\ \text{kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pI,in}} = 670.38 + 8.15 = 678.53\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_5 = 8\ \text{MPa} \\ T_5 = 400°\text{C} \end{matrix}\right\} \begin{matrix} h_5 = 3139.4\ \text{kJ/kg} \\ s_5 = 6.3658\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_6 = 0.6\ \text{MPa} \\ s_6 = s_5 \end{matrix}\right\} \begin{matrix} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.3658 - 1.9308}{4.8285} = 0.9185 \\ h_6 = h_f + x_6 h_{fg} = 670.38 + (0.9185)(2085.8) = 2586.1\ \text{kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_7 = 20\ \text{kPa} \\ s_7 = s_5 \end{matrix}\right\} \begin{matrix} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.3658 - 0.8320}{7.0752} = 0.7821 \\ h_7 = h_f + x_7 h_{fg} = 251.42 + (0.7821)(2357.5) = 2095.2\ \text{kJ/kg} \end{matrix}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

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$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s (h_5 - h_4) = \dot{m}_{\text{air}} (h_{11} - h_{12})$$

$$\frac{\dot{m}_{\text{air}}}{\dot{m}_s} = \frac{h_5 - h_4}{h_{11} - h_{12}} = \frac{3139.4 - 678.53}{735.80 - 462.02} = \mathbf{8.99 \text{ kg air / kg steam}}$$

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$ for the open FWH, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = (1) h_3$$

Thus,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{670.38 - 252.01}{2586.1 - 252.01} = 0.1792 \quad (\text{the fraction of steam extracted})$$

$$w_T = h_5 - h_6 + (1 - y)(h_6 - h_7)$$
$$= 3139.4 - 2586.1 + (1 - 0.1792)(2586.1 - 2095.2) = 956.23 \text{ kJ/kg}$$

$$w_{\text{net,steam}} = w_T - w_{p,\text{in}} = w_T - (1 - y) w_{p,I} - w_{p,II}$$
$$= 956.23 - (1 - 0.1792)(0.59) - 8.15 = 948.56 \text{ kJ/kg}$$

$$w_{\text{net,gas}} = w_T - w_{C,\text{in}} = (h_{10} - h_{11}) - (h_9 - h_8)$$
$$= 1515.42 - 735.8 - (635.5 - 300.19) = 444.3 \text{ kJ/kg}$$

The net work output per unit mass of gas is

$$w_{\text{net}} = w_{\text{net,gas}} + \tfrac{1}{8.99} w_{\text{net,steam}} = 444.3 + \tfrac{1}{8.99}(948.56) = 549.8 \text{ kJ/kg}$$

$$\dot{m}_{\text{air}} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{450{,}000 \text{ kJ/s}}{549.7 \text{ kJ/kg}} = 818.7 \text{ kg/s}$$

and

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_{10} - h_9) = (818.5 \text{ kg/s})(1515.42 - 635.5) \text{ kJ/kg} = \mathbf{720{,}215 \text{ kW}}$$

(*c*) $$\eta_{th} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{450{,}000 \text{ kW}}{720{,}215 \text{ kW}} = \mathbf{62.5\%}$$

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10-84 Problem 10-83 is reconsidered. The effect of the gas cycle pressure ratio on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input data"
T[8] = 300 [K]              "Gas compressor inlet"
P[8] = 14.7 [kPa]           "Assumed air inlet pressure"
"Pratio = 14"               "Pressure ratio for gas compressor"
T[10] = 1400 [K]            "Gas turbine inlet"
T[12] = 460 [K]             "Gas exit temperature from Gas-to-steam heat exchanger "
P[12] = P[8]                "Assumed air exit pressure"
W_dot_net=450 [MW]
Eta_comp = 1.0
Eta_gas_turb = 1.0
Eta_pump = 1.0
Eta_steam_turb = 1.0
P[5] = 8000 [kPa]            "Steam turbine inlet"
T[5] =(400+273) "[K]"        "Steam turbine inlet"
P[6] = 600 [kPa]            "Extraction pressure for steam open feedwater heater"
P[7] = 20 [kPa]             "Steam condenser pressure"

"GAS POWER CYCLE ANALYSIS"

"Gas Compressor anaysis"
s[8]=ENTROPY(Air,T=T[8],P=P[8])
ss9=s[8] "For the ideal case the entropies are constant across the compressor"
P[9] = Pratio*P[8]
Ts9=temperature(Air,s=ss9,P=P[9])"Ts9 is the isentropic value of T[9] at compressor exit"
Eta_comp = w_gas_comp_isen/w_gas_comp "compressor adiabatic efficiency, w_comp > w_comp_isen"
h[8] + w_gas_comp_isen =hs9"SSSF conservation of energy for the isentropic compressor, assuming: adiabatic,
ke=pe=0 per unit gas mass flow rate in kg/s"
h[8]=ENTHALPY(Air,T=T[8])
hs9=ENTHALPY(Air,T=Ts9)
h[8] + w_gas_comp = h[9]"SSSF conservation of energy for the actual compressor, assuming: adiabatic,
ke=pe=0"
T[9]=temperature(Air,h=h[9])
s[9]=ENTROPY(Air,T=T[9],P=P[9])

"Gas Cycle External heat exchanger analysis"
h[9] + q_in = h[10]"SSSF conservation of energy for the external heat exchanger, assuming W=0, ke=pe=0"
h[10]=ENTHALPY(Air,T=T[10])
P[10]=P[9]                          "Assume process 9-10 is SSSF constant pressure"
Q_dot_in"MW"*1000"kW/MW"=m_dot_gas*q_in

"Gas Turbine analysis"
s[10]=ENTROPY(Air,T=T[10],P=P[10])
ss11=s[10] "For the ideal case the entropies are constant across the turbine"
P[11] = P[10] /Pratio
Ts11=temperature(Air,s=ss11,P=P[11])"Ts11 is the isentropic value of T[11] at gas turbine exit"
Eta_gas_turb = w_gas_turb /w_gas_turb_isen "gas turbine adiabatic efficiency, w_gas_turb_isen > w_gas_turb"
h[10] = w_gas_turb_isen + hs11"SSSF conservation of energy for the isentropic gas turbine, assuming:
adiabatic, ke=pe=0"
hs11=ENTHALPY(Air,T=Ts11)
h[10] = w_gas_turb + h[11]"SSSF conservation of energy for the actual gas turbine, assuming: adiabatic,
ke=pe=0"
T[11]=temperature(Air,h=h[11])
s[11]=ENTROPY(Air,T=T[11],P=P[11])
```

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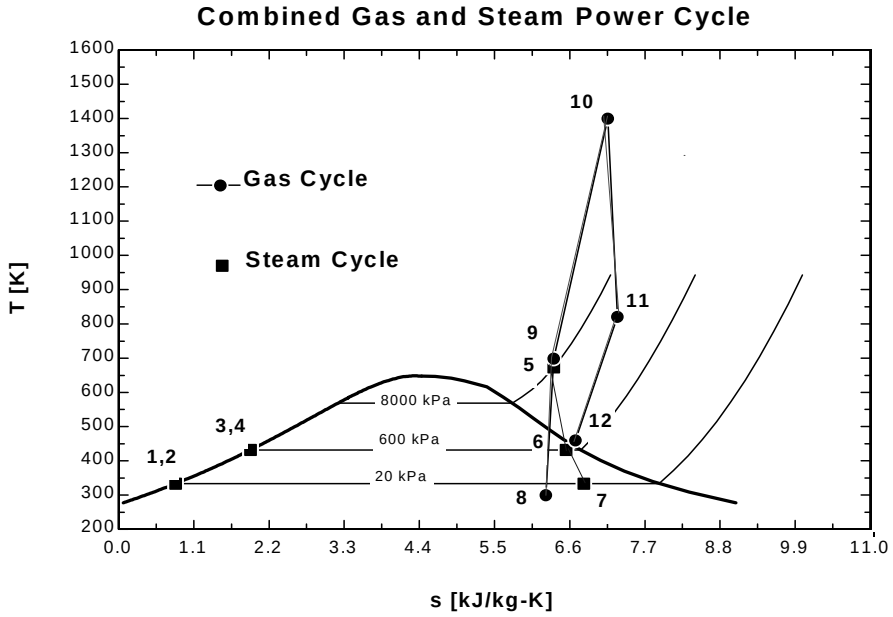
```
"Gas-to-Steam Heat Exchanger"
"SSSF conservation of energy for the gas-to-steam heat exchanger, assuming: adiabatic,
W=0, ke=pe=0"
m_dot_gas*h[11] + m_dot_steam*h[4] = m_dot_gas*h[12] + m_dot_steam*h[5]
h[12]=ENTHALPY(Air, T=T[12])
s[12]=ENTROPY(Air,T=T[12],P=P[12])

"STEAM CYCLE ANALYSIS"
"Steam Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
P[1] = P[7]
P[2]=P[6]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Saturated liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"Open Feedwater Heater analysis"
y*h[6] + (1-y)*h[2] = 1*h[3] "Steady-flow conservation of energy"
P[3]=P[6]
h[3]=enthalpy(Fluid$,P=P[3],x=0)      "Condensate leaves heater as sat. liquid at P[3]"
T[3]=temperature(Fluid$,P=P[3],x=0)
s[3]=entropy(Fluid$,P=P[3],x=0)
"Boiler condensate pump  or Pump 2 analysis"
P[4] = P[5]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])
w_steam_pumps = (1-y)*w_pump1+ w_pump2 "Total steam pump work input/ mass steam"
"Steam Turbine analysis"
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s[5]=entropy(Fluid$,P=P[5],T=T[5])
ss6=s[5]
hs6=enthalpy(Fluid$,s=ss6,P=P[6])
Ts6=temperature(Fluid$,s=ss6,P=P[6])
h[6]=h[5]-Eta_steam_turb*(h[5]-hs6)"Definition of steam turbine efficiency"
T[6]=temperature(Fluid$,P=P[6],h=h[6])
s[6]=entropy(Fluid$,P=P[6],h=h[6])
ss7=s[5]
hs7=enthalpy(Fluid$,s=ss7,P=P[7])
Ts7=temperature(Fluid$,s=ss7,P=P[7])
h[7]=h[5]-Eta_steam_turb*(h[5]-hs7)"Definition of steam turbine efficiency"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
"SSSF conservation of energy for the steam turbine: adiabatic, neglect ke and pe"
h[5] = w_steam_turb + y*h[6] +(1-y)*h[7]
"Steam Condenser analysis"
(1-y)*h[7]=q_out+(1-y)*h[1]"SSSF conservation of energy for the Condenser per unit mass"
Q_dot_out*Convert(MW, kW)=m_dot_steam*q_out
"Cycle Statistics"
MassRatio_gastosteam =m_dot_gas/m_dot_steam
W_dot_net*Convert(MW, kW)=m_dot_gas*(w_gas_turb-w_gas_comp)+ m_dot_steam*(w_steam_turb -
w_steam_pumps)"definition of the net cycle work"
Eta_th=W_dot_net/Q_dot_in*Convert(, %) "Cycle thermal efficiency, in percent"
```

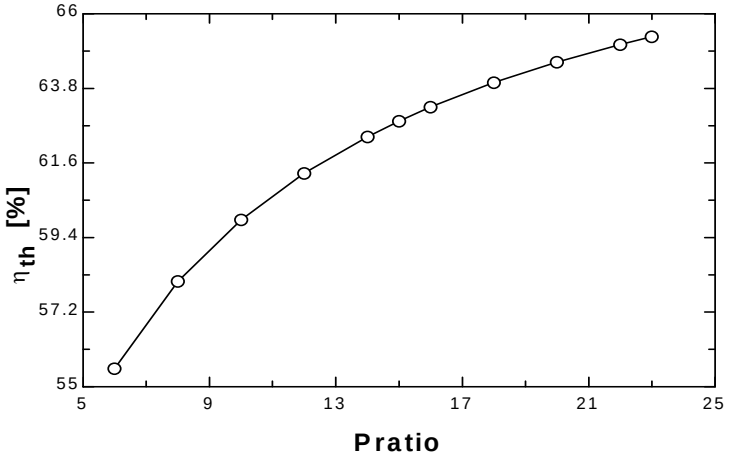
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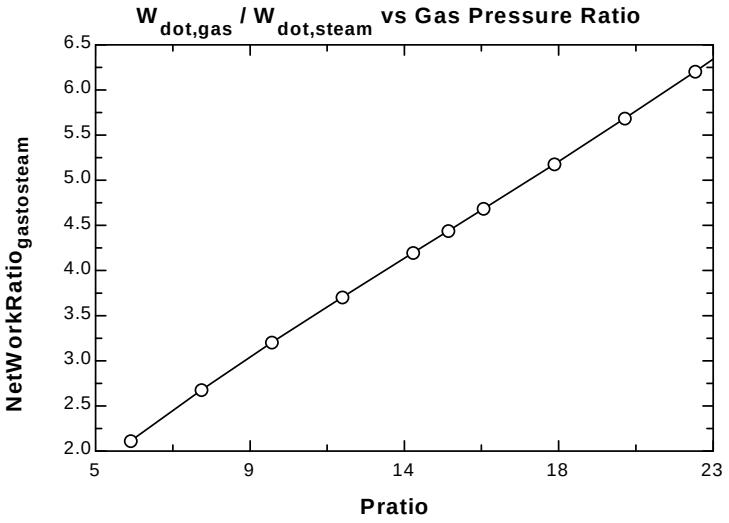
Bwr=(m_dot_gas*w_gas_comp + m_dot_steam*w_steam_pumps)/(m_dot_gas*w_gas_turb + m_dot_steam*w_steam_turb) "Back work ratio"
W_dot_net_steam = m_dot_steam*(w_steam_turb - w_steam_pumps)
W_dot_net_gas = m_dot_gas*(w_gas_turb - w_gas_comp)
NetWorkRatio_gastosteam = W_dot_net_gas/W_dot_net_steam

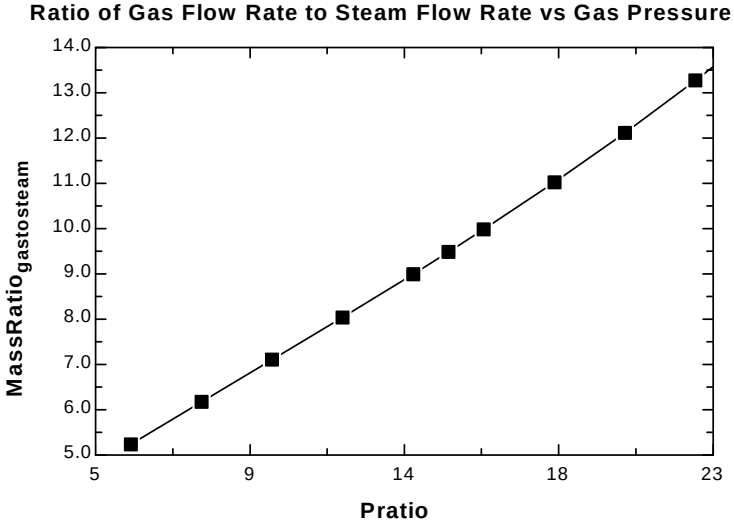
Pratio	MassRatio $_{gastosteam}$	W_{netgas} [kW]	$W_{netsteam}$ [kW]	η_{th} [%]	NetWorkRatio $_{gastosteam}$
10	7.108	342944	107056	59.92	3.203
11	7.574	349014	100986	60.65	3.456
12	8.043	354353	95647	61.29	3.705
13	8.519	359110	90890	61.86	3.951
14	9.001	363394	86606	62.37	4.196
15	9.492	367285	82715	62.83	4.44
16	9.993	370849	79151	63.24	4.685
17	10.51	374135	75865	63.62	4.932
18	11.03	377182	72818	63.97	5.18
19	11.57	380024	69976	64.28	5.431
20	12.12	382687	67313	64.57	5.685



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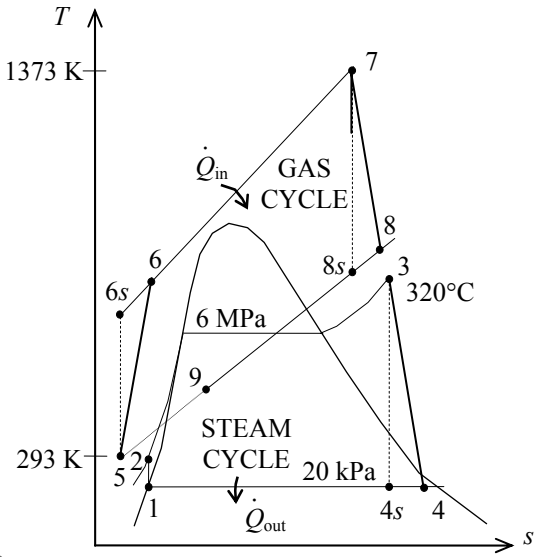
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10-85 A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The mass flow rate of air for a specified power output is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable fo Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).

Analysis Working around the topping cycle gives the following results:



$$T_{6s} = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (293\text{ K})(8)^{0.4/1.4} = 530.8\text{ K}$$

$$\eta_C = \frac{h_{6s} - h_5}{h_6 - h_5} = \frac{c_p(T_{6s} - T_5)}{c_p(T_6 - T_5)}$$

$$\longrightarrow T_6 = T_5 + \frac{T_{6s} - T_5}{\eta_C}$$

$$= 293 + \frac{530.8 - 293}{0.85} = 572.8\text{ K}$$

$$T_{8s} = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (1373\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 758.0\text{ K}$$

$$\eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} = \frac{c_p(T_7 - T_8)}{c_p(T_7 - T_{8s})} \longrightarrow T_8 = T_7 - \eta_T(T_7 - T_{8s})$$

$$= 1373 - (0.90)(1373 - 758.0)$$

$$= 819.5\text{ K}$$

$$T_9 = T_{\text{sat @ 6000 kPa}} = 275.6°\text{C} = 548.6\text{ K}$$

Fixing the states around the bottom steam cycle yields (Tables A-4, A-5, A-6):

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.42\text{ kJ/kg}$$

$$\nu_1 = \nu_{f\,@\,20\text{ kPa}} = 0.001017\text{ m}^3\text{/kg}$$

$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$

$$= (0.001017\text{ m}^3\text{/kg})(6000 - 20)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 6.08\text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 6.08 = 257.5\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 6000\text{ kPa} \\ T_3 = 320°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 2953.6\text{ kJ/kg} \\ s_3 = 6.1871\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 20\text{ kPa} \\ s_4 = s_3 \end{array}\right\} \quad h_{4s} = 2035.8\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s})$$

$$= 2953.6 - (0.90)(2953.6 - 2035.8)$$

$$= 2127.6\text{ kJ/kg}$$

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The net work outputs from each cycle are

$$\begin{aligned} w_{\text{net, gas cycle}} &= w_{\text{T,out}} - w_{\text{C,in}} \\ &= c_p(T_7 - T_8) - c_p(T_6 - T_5) \\ &= (1.005\ \text{kJ/kg}\cdot\text{K})(1373 - 819.5 - 572.7 + 293)\text{K} \\ &= 275.2\ \text{kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net, steam cycle}} &= w_{\text{T,out}} - w_{\text{P,in}} \\ &= (h_3 - h_4) - w_{\text{P,in}} \\ &= (2953.6 - 2127.6) - 6.08 \\ &= 819.9\ \text{kJ/kg} \end{aligned}$$

An energy balance on the heat exchanger gives

$$\dot{m}_a c_p(T_8 - T_9) = \dot{m}_w(h_3\text{-}h_2) \longrightarrow \dot{m}_w = \frac{c_p(T_8 - T_9)}{h_3\text{-}h_2}\dot{m}_a = \frac{(1.005)(819.5 - 548.6)}{2953.6 - 257.5} = 0.1010\dot{m}_a$$

That is, 1 kg of exhaust gases can heat only 0.1010 kg of water. Then, the mass flow rate of air is

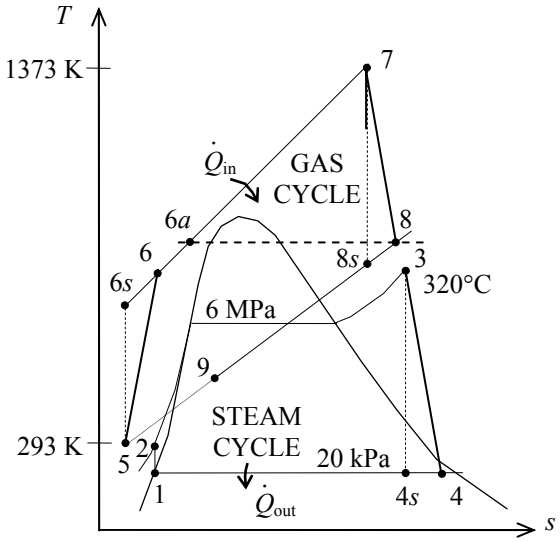
$$\dot{m}_a = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{100{,}000\ \text{kJ/s}}{(1\times 275.2 + 0.1010\times 819.9)\ \text{kJ/kg air}} = \mathbf{279.3\ kg/s}$$

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10-86 A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The mass flow rate of air for a specified power output is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable fo Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).



Analysis With an ideal regenerator, the temperature of the air at the compressor exit will be heated to the to the temperature at the turbine exit. Representing this state by "6a"

$$T_{6a} = T_8 = 819.5\,\text{K}$$

The rate of heat addition in the cycle is

$$\begin{aligned}\dot{Q}_{\text{in}} &= \dot{m}_a c_p (T_7 - T_{6a}) \\ &= (279.3\,\text{kg/s})(1.005\,\text{kJ/kg}\cdot°\text{C})(1373 - 819.5)\,\text{K} \\ &= 155{,}370\,\text{kW}\end{aligned}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{100{,}000\,\text{kW}}{155{,}370\,\text{kW}} = \mathbf{0.6436}$$

Without the regenerator, the rate of heat addition and the thermal efficiency are

$$\dot{Q}_{\text{in}} = \dot{m}_a c_p (T_7 - T_6) = (279.3\,\text{kg/s})(1.005\,\text{kJ/kg}\cdot°\text{C})(1373 - 572.7)\,\text{K} = 224{,}640\,\text{kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{100{,}000\,\text{kW}}{224{,}640\,\text{kW}} = \mathbf{0.4452}$$

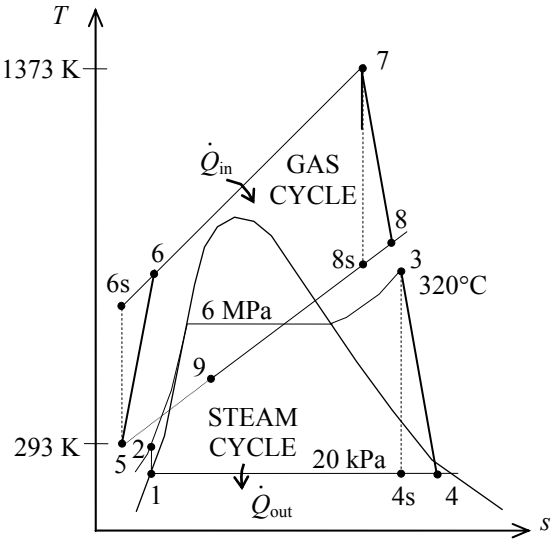
The change in the thermal efficiency due to using the ideal regenerator is

$$\Delta\eta_{\text{th}} = 0.6436 - 0.4452 = \mathbf{0.1984}$$

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10-87 The component of the combined cycle with the largest exergy destruction of the component of the combined cycle in Prob. 10-86 is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From Problem 10-86,

$$T_{\text{source, gas cycle}} = 1373\ \text{K}$$
$$T_{\text{source, steam cycle}} = T_8 = 819.5\ \text{K}$$
$$T_{\text{sink}} = 293\ \text{K}$$
$$s_1 = s_2 = s_{f\,@\,20\,\text{kPa}} = 0.8320\ \text{kJ/kg}\cdot\text{K}$$
$$s_3 = 6.1871\ \text{kJ/kg}\cdot\text{K}$$
$$s_4 = 6.4627\ \text{kJ/kg}\cdot\text{K}$$
$$q_{\text{in},67} = c_p(T_7 - T_6) = 804.3\ \text{kJ/kg}$$
$$q_{\text{in},23} = h_3 - h_2 = 2696.1\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 1876.2\ \text{kJ/kg}$$
$$\dot{m}_w = 0.1010\dot{m}_a = 0.1010(279.3) = 28.21\ \text{kg/s}$$

$$\dot{X}_{\text{destroyed},12} = 0 \quad \text{(isentropic process)}$$

$$\dot{X}_{\text{destroyed},34} = \dot{m}_w T_0 (s_4 - s_3) = (28.21\ \text{kg/s})(293\ \text{K})(6.4627 - 6.1871) = 2278\ \text{kW}$$

$$\dot{X}_{\text{destroyed},41} = \dot{m}_w T_0 \left(s_1 - s_4 + \frac{q_{\text{out}}}{T_{\text{sink}}} \right)$$
$$= (28.21\ \text{kg/s})(293\ \text{K})\left(0.8320 - 6.1871 + \frac{1876.2\ \text{kJ/kg}}{293\ \text{K}} \right) = 8665\ \text{kW}$$

$$\dot{X}_{\text{destroyed,heat exchanger}} = \dot{m}_a T_0 \Delta s_{89} + \dot{m}_w T_0 \Delta s_{23} = \dot{m}_a T_0 \left(c_p \ln \frac{T_9}{T_8} \right) + \dot{m}_a T_0 (s_3 - s_2)$$
$$= (279.3)(293)\left[(1.005) \ln \frac{548.6}{819.5} \right] + (28.21)(293)(6.1871 - 0.8320)$$
$$= 11260\ \text{kW}$$

$$\dot{X}_{\text{destroyed},56} = \dot{m}_a T_0 \left(c_p \ln \frac{T_6}{T_5} - R \ln \frac{P_6}{P_5} \right) = (279.3)(293)\left[(1.005) \ln \frac{572.7}{293} - (0.287) \ln(8) \right] = 6280\ \text{kW}$$

$$\dot{X}_{\text{destroyed},67} = \dot{m}_a T_0 \left(c_p \ln \frac{T_7}{T_6} - \frac{q_{\text{in}}}{T_{\text{source}}} \right) = (279.3)(293)\left[(1.005) \ln \frac{1373}{572.7} - \frac{804.3}{1373} \right] = \mathbf{23{,}970\ kW}$$

$$\dot{X}_{\text{destroyed},78} = \dot{m}_a T_0 \left(c_p \ln \frac{T_8}{T_7} - R \ln \frac{P_8}{P_7} \right) = (279.3)(293)\left[(1.005) \ln \frac{819.5}{1373} - (0.287) \ln\left(\frac{1}{8} \right) \right] = 6396\ \text{kW}$$

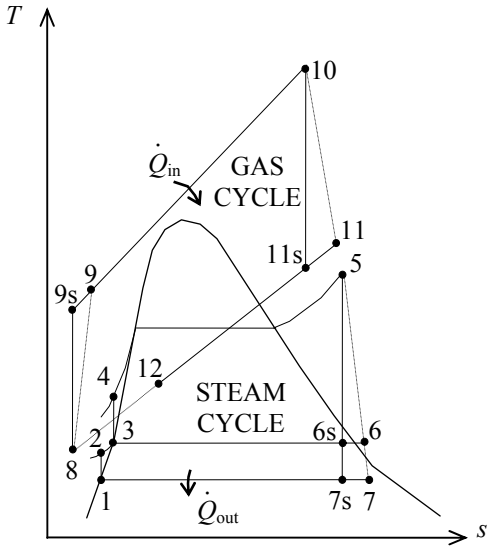
The largest exergy destruction occurs during the heat addition process in the combustor of the gas cycle.

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10-88 A 280-MW combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal Rankine cycle with an open feedwater heater. The mass flow rate of air to steam, the required rate of heat input in the combustion chamber, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) Using the properties of air from Table A-17, the analysis of gas cycle yields



$$T_8 = 300\ \text{K} \longrightarrow \begin{matrix} h_8 = 300.19\ \text{kJ/kg} \\ P_{r_8} = 1.386 \end{matrix}$$

$$P_{r_9} = \frac{P_9}{P_8} P_{r_8} = (11)(1.386) = 15.25 \longrightarrow h_{9s} = 595.84\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{9s} - h_8}{h_9 - h_8} \longrightarrow \begin{aligned} h_9 &= h_8 + (h_{9s} - h_8)/\eta_C \\ &= 300.19 + (595.84 - 300.19)/(0.82) \\ &= 660.74\ \text{kJ/kg} \end{aligned}$$

$$T_{10} = 1100\ \text{K} \longrightarrow \begin{matrix} h_{10} = 1161.07\ \text{kJ/kg} \\ P_{r_{10}} = 167.1 \end{matrix}$$

$$P_{r_{11}} = \frac{P_{11}}{P_{10}} P_{r_{10}} = \left(\frac{1}{11}\right)(167.1) = 15.19 \longrightarrow h_{11s} = 595.18\ \text{kJ/kg}$$

$$\eta_T = \frac{h_{10} - h_{11}}{h_{10} - h_{11s}} \longrightarrow \begin{aligned} h_{11} &= h_{10} - \eta_T (h_{10} - h_{11s}) \\ &= 1161.07 - (0.86)(1161.07 - 595.18) \\ &= 674.40\ \text{kJ/kg} \end{aligned}$$

$$T_{12} = 420\ \text{K} \longrightarrow h_{12} = 421.26\ \text{kJ/kg}$$

From the steam tables (Tables A-4, A-5, and A-6),

$$\begin{aligned} h_1 &= h_{f\,@\,10\ \text{kPa}} = 191.81\ \text{kJ/kg} \\ \nu_1 &= \nu_{f\,@\,10\ \text{kPa}} = 0.00101\ \text{m}^3/\text{kg} \end{aligned}$$

$$\begin{aligned} w_{\text{pI,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.00101\ \text{m}^3/\text{kg})(800 - 10\ \text{kPa}) \left(\frac{1\ \text{kJ}}{1\ \text{kPa} \cdot \text{m}^3}\right) \\ &= 0.80\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.80 = 192.60\ \text{kJ/kg}$$

$$\begin{aligned} h_3 &= h_{f\,@\,0.8\ \text{MPa}} = 720.87\ \text{kJ/kg} \\ \nu_3 &= \nu_{f\,@\,0.8\ \text{MPa}} = 0.001115\ \text{m}^3/\text{kg} \end{aligned}$$

$$\begin{aligned} w_{\text{pII,in}} &= \nu_3 (P_4 - P_3) \\ &= (0.001115\ \text{m}^3/\text{kg})(5000 - 800\ \text{kPa}) \left(\frac{1\ \text{kJ}}{1\ \text{kPa} \cdot \text{m}^3}\right) \\ &= 4.68\ \text{kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pI,in}} = 720.87 + 4.68 = 725.55\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_5 = 5\ \text{MPa} \\ T_5 = 350°\text{C} \end{matrix}\right\} \begin{matrix} h_5 = 3069.3\ \text{kJ/kg} \\ s_5 = 6.4516\ \text{kJ/kg} \cdot \text{K} \end{matrix}$$

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$$\left.\begin{array}{l} P_6 = 0.8 \text{ MPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.4516 - 2.0457}{4.6160} = 0.9545 \\ h_{6s} = h_f + x_{6s} h_{fg} = 720.87 + (0.9545)(2085.8) = 2675.1 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s}) = 3069.3 - (0.86)(3069.3 - 2675.1) = 2730.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 10 \text{ kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_{7s} = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.4516 - 0.6492}{7.4996} = 0.7737 \\ h_{7s} = h_f + x_7 h_{fg} = 191.81 + (0.7737)(2392.1) = 2042.5 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_7}{h_5 - h_{7s}} \longrightarrow h_7 = h_5 - \eta_T (h_5 - h_{7s}) = 3069.3 - (0.86)(3069.3 - 2042.5) = 2186.3 \text{ kJ/kg}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s (h_5 - h_4) = \dot{m}_{\text{air}} (h_{11} - h_{12})$$

$$\frac{\dot{m}_{\text{air}}}{\dot{m}_s} = \frac{h_5 - h_4}{h_{11} - h_{12}} = \frac{3069.3 - 725.55}{674.40 - 421.26} = \mathbf{9.259 \text{ kg air / kg steam}}$$

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the open FWH, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = (1) h_3$$

Thus,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{720.87 - 192.60}{2730.3 - 192.60} = 0.2082 \quad (\text{the fraction of steam extracted})$$

$$\begin{aligned} w_T &= \eta_T [h_5 - h_6 + (1 - y)(h_6 - h_7)] \\ &= (0.86)[3069.3 - 2730.3 + (1 - 0.2082)(2730.3 - 2186.3)] = 769.8 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net,steam}} &= w_T - w_{\text{p,in}} = w_T - (1 - y) w_{\text{p,I}} - w_{\text{p,II}} \\ &= 769.8 - (1 - 0.2082)(0.80) - 4.68 = 764.5 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net,gas}} &= w_T - w_{C,\text{in}} = (h_{10} - h_{11}) - (h_9 - h_8) \\ &= 1161.07 - 674.40 - (660.74 - 300.19) = 126.12 \text{ kJ/kg} \end{aligned}$$

The net work output per unit mass of gas is

$$w_{\text{net}} = w_{\text{net,gas}} + \frac{1}{6.425} w_{\text{net,steam}} = 126.12 + \frac{1}{9.259}(764.5) = 208.69 \text{ kJ/kg}$$

$$\dot{m}_{\text{air}} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{280{,}000 \text{ kJ/s}}{208.69 \text{ kJ/kg}} = 1341.7 \text{ kg/s}$$

and $$\dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_{10} - h_9) = (1341.7 \text{ kg/s})(1161.07 - 660.74) \text{ kJ/kg} = \mathbf{671{,}300 \text{ kW}}$$

(*c*) $$\eta_{th} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{280{,}000 \text{ kW}}{671{,}300 \text{ kW}} = 0.4171 = \mathbf{41.7\%}$$

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10-89 Problem 10-88 is reconsidered. The effect of the gas cycle pressure ratio on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input data"
T[8] = 300 [K]              "Gas compressor inlet"
P[8] = 100 [kPa]            "Assumed air inlet pressure"
"Pratio = 11"               "Pressure ratio for gas compressor"
T[10] = 1100 [K]            "Gas turbine inlet"
T[12] = 420 [K]             "Gas exit temperature from Gas-to-steam heat exchanger "
P[12] = P[8]                "Assumed air exit pressure"
W_dot_net=280 [MW]
Eta_comp = 0.82
Eta_gas_turb = 0.86
Eta_pump = 1.0
Eta_steam_turb = 0.86
P[5] = 5000 [kPa]            "Steam turbine inlet"
T[5] =(350+273.15) "K"       "Steam turbine inlet"
P[6] = 800 [kPa]            "Extraction pressure for steam open feedwater heater"
P[7] = 10 [kPa]             "Steam condenser pressure"

"GAS POWER CYCLE ANALYSIS"

"Gas Compressor anaysis"
s[8]=ENTROPY(Air,T=T[8],P=P[8])
ss9=s[8] "For the ideal case the entropies are constant across the compressor"
P[9] = Pratio*P[8]
Ts9=temperature(Air,s=ss9,P=P[9])"Ts9 is the isentropic value of T[9] at compressor exit"
Eta_comp = w_gas_comp_isen/w_gas_comp "compressor adiabatic efficiency, w_comp > w_comp_isen"
h[8] + w_gas_comp_isen =hs9"SSSF conservation of energy for the isentropic compressor, assuming: adiabatic,
ke=pe=0 per unit gas mass flow rate in kg/s"
h[8]=ENTHALPY(Air,T=T[8])
hs9=ENTHALPY(Air,T=Ts9)
h[8] + w_gas_comp = h[9]"SSSF conservation of energy for the actual compressor, assuming: adiabatic,
ke=pe=0"
T[9]=temperature(Air,h=h[9])
s[9]=ENTROPY(Air,T=T[9],P=P[9])

"Gas Cycle External heat exchanger analysis"
h[9] + q_in = h[10]"SSSF conservation of energy for the external heat exchanger, assuming W=0, ke=pe=0"
h[10]=ENTHALPY(Air,T=T[10])
P[10]=P[9]                          "Assume process 9-10 is SSSF constant pressure"
Q_dot_in"MW"*1000"kW/MW"=m_dot_gas*q_in

"Gas Turbine analysis"
s[10]=ENTROPY(Air,T=T[10],P=P[10])
ss11=s[10] "For the ideal case the entropies are constant across the turbine"
P[11] = P[10] /Pratio
Ts11=temperature(Air,s=ss11,P=P[11])"Ts11 is the isentropic value of T[11] at gas turbine exit"
Eta_gas_turb = w_gas_turb /w_gas_turb_isen "gas turbine adiabatic efficiency, w_gas_turb_isen > w_gas_turb"
h[10] = w_gas_turb_isen + hs11"SSSF conservation of energy for the isentropic gas turbine, assuming:
adiabatic, ke=pe=0"
hs11=ENTHALPY(Air,T=Ts11)
h[10] = w_gas_turb + h[11]"SSSF conservation of energy for the actual gas turbine, assuming: adiabatic,
ke=pe=0"
T[11]=temperature(Air,h=h[11])
s[11]=ENTROPY(Air,T=T[11],P=P[11])
```

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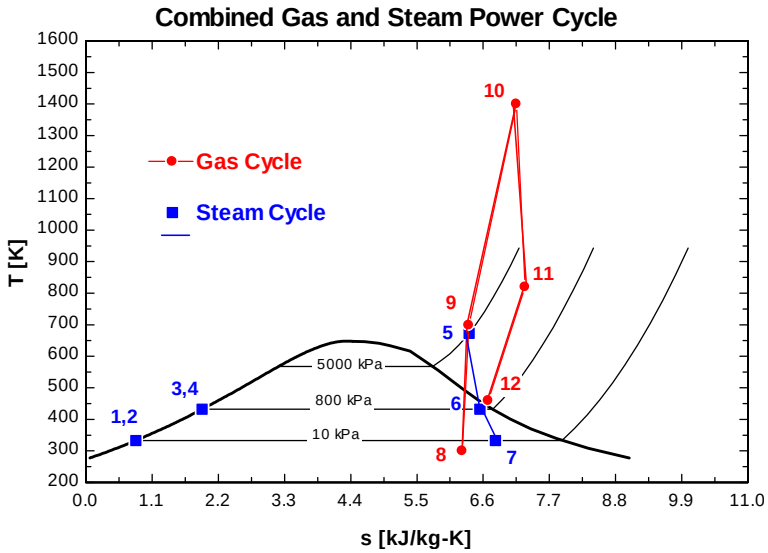
```
"Gas-to-Steam Heat Exchanger"
"SSSF conservation of energy for the gas-to-steam heat exchanger, assuming: adiabatic,
W=0, ke=pe=0"
m_dot_gas*h[11] + m_dot_steam*h[4] = m_dot_gas*h[12] + m_dot_steam*h[5]
h[12]=ENTHALPY(Air, T=T[12])
s[12]=ENTROPY(Air,T=T[12],P=P[12])

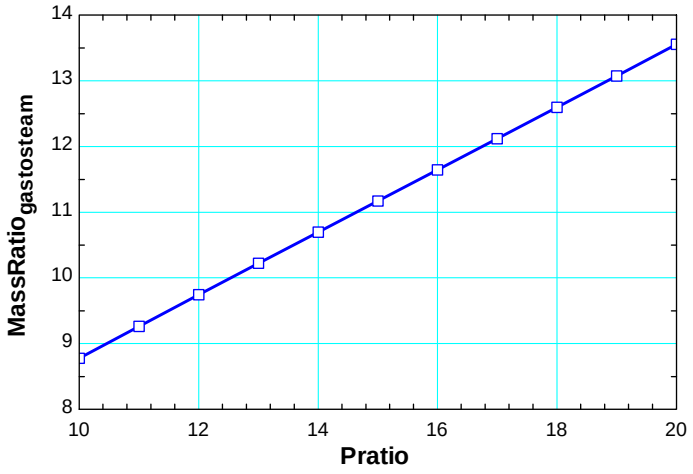
"STEAM CYCLE ANALYSIS"
"Steam Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
P[1] = P[7]
P[2]=P[6]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Saturated liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"Open Feedwater Heater analysis"
y*h[6] + (1-y)*h[2] = 1*h[3] "Steady-flow conservation of energy"
P[3]=P[6]
h[3]=enthalpy(Fluid$,P=P[3],x=0)      "Condensate leaves heater as sat. liquid at P[3]"
T[3]=temperature(Fluid$,P=P[3],x=0)
s[3]=entropy(Fluid$,P=P[3],x=0)
"Boiler condensate pump  or Pump 2 analysis"
P[4] = P[5]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])
w_steam_pumps = (1-y)*w_pump1+ w_pump2 "Total steam pump work input/ mass steam"
"Steam Turbine analysis"
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s[5]=entropy(Fluid$,P=P[5],T=T[5])
ss6=s[5]
hs6=enthalpy(Fluid$,s=ss6,P=P[6])
Ts6=temperature(Fluid$,s=ss6,P=P[6])
h[6]=h[5]-Eta_steam_turb*(h[5]-hs6)"Definition of steam turbine efficiency"
T[6]=temperature(Fluid$,P=P[6],h=h[6])
s[6]=entropy(Fluid$,P=P[6],h=h[6])
ss7=s[5]
hs7=enthalpy(Fluid$,s=ss7,P=P[7])
Ts7=temperature(Fluid$,s=ss7,P=P[7])
h[7]=h[5]-Eta_steam_turb*(h[5]-hs7)"Definition of steam turbine efficiency"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
"SSSF conservation of energy for the steam turbine: adiabatic, neglect ke and pe"
h[5] = w_steam_turb + y*h[6] +(1-y)*h[7]
"Steam Condenser analysis"
(1-y)*h[7]=q_out+(1-y)*h[1]"SSSF conservation of energy for the Condenser per unit mass"
Q_dot_out*Convert(MW, kW)=m_dot_steam*q_out
"Cycle Statistics"
MassRatio_gastosteam =m_dot_gas/m_dot_steam
W_dot_net*Convert(MW, kW)=m_dot_gas*(w_gas_turb-w_gas_comp)+ m_dot_steam*(w_steam_turb -
w_steam_pumps)"definition of the net cycle work"
Eta_th=W_dot_net/Q_dot_in*Convert(, %) "Cycle thermal efficiency, in percent"
```

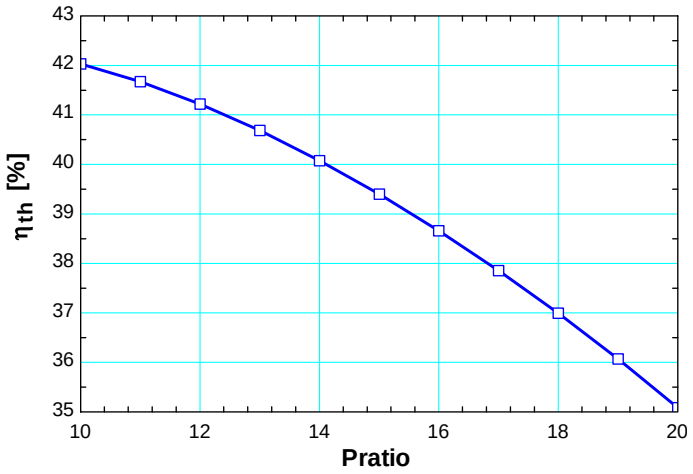
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Bwr=(m_dot_gas*w_gas_comp + m_dot_steam*w_steam_pumps)/(m_dot_gas*w_gas_turb + m_dot_steam*w_steam_turb) "Back work ratio"
W_dot_net_steam = m_dot_steam*(w_steam_turb - w_steam_pumps)
W_dot_net_gas = m_dot_gas*(w_gas_turb - w_gas_comp)
NetWorkRatio_gastosteam = W_dot_net_gas/W_dot_net_steam

Pratio	$MassRatio_{gastosteam}$	η_{th} [%]
10	8.775	42.03
11	9.262	41.67
12	9.743	41.22
13	10.22	40.68
14	10.7	40.08
15	11.17	39.4
16	11.64	38.66
17	12.12	37.86
18	12.59	36.99
19	13.07	36.07
20	13.55	35.08







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10-90 A combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal reheat Rankine cycle. The moisture percentage at the exit of the low-pressure turbine, the steam temperature at the inlet of the high-pressure turbine, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) We obtain the air properties from EES. The analysis of gas cycle is as follows

$$T_7 = 15°\text{C} \longrightarrow h_7 = 288.50 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_7 = 15°\text{C} \\ P_7 = 100 \text{ kPa} \end{matrix}\right\} s_7 = 5.6648 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_8 = 700 \text{ kPa} \\ s_8 = s_7 \end{matrix}\right\} h_{8s} = 503.47 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{8s} - h_7}{h_8 - h_7} \longrightarrow h_8 = h_7 + (h_{8s} - h_7)/\eta_C$$
$$= 290.16 + (503.47 - 290.16)/(0.80)$$
$$= 557.21 \text{ kJ/kg}$$

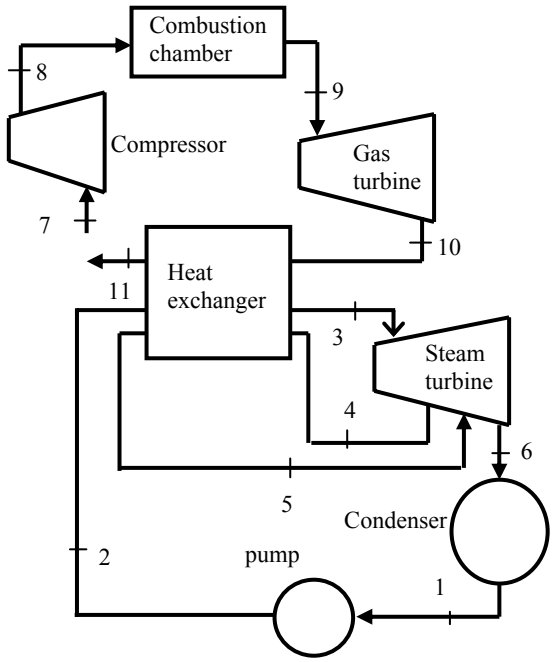
$$T_9 = 950°\text{C} \longrightarrow h_9 = 1304.8 \text{ kJ/kg}$$

$$\left.\begin{matrix} T_9 = 950°\text{C} \\ P_9 = 700 \text{ kPa} \end{matrix}\right\} s_9 = 6.6456 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_{10} = 100 \text{ kPa} \\ s_{10} = s_9 \end{matrix}\right\} h_{10s} = 763.79 \text{ kJ/kg}$$

$$\eta_T = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \longrightarrow h_{10} = h_9 - \eta_T (h_9 - h_{10s})$$
$$= 1304.8 - (0.80)(1304.8 - 763.79)$$
$$= 871.98 \text{ kJ/kg}$$

$$T_{11} = 200°\text{C} \longrightarrow h_{11} = 475.62 \text{ kJ/kg}$$

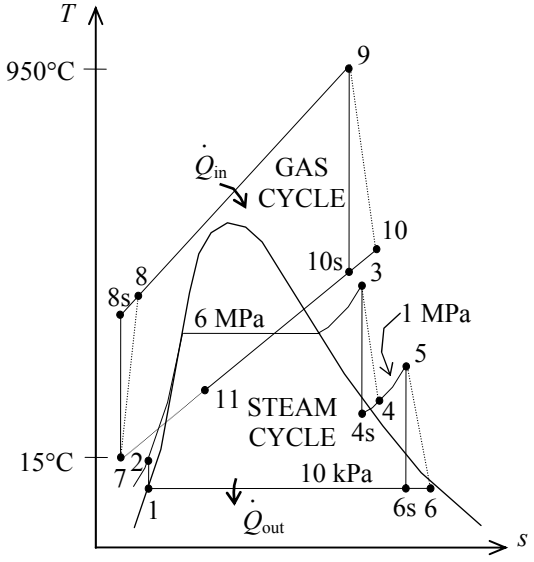


From the steam tables (Tables A-4, A-5, and A-6 or from EES),

$$h_1 = h_{f\,@\,10 \text{ kPa}} = 191.81 \text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,10 \text{ kPa}} = 0.00101 \text{ m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \nu_1 (P_2 - P_1)/\eta_p$$
$$= (0.00101 \text{ m}^3/\text{kg})(6000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)/0.80$$
$$= 7.56 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 7.65 = 199.37 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_5 = 1 \text{ MPa} \\ T_5 = 400°\text{C} \end{matrix}\right\} \begin{matrix} h_5 = 3264.5 \text{ kJ/kg} \\ s_5 = 7.4670 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$



$$\left.\begin{matrix} P_6 = 10 \text{ kPa} \\ s_{6s} = s_5 \end{matrix}\right\} \begin{matrix} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{7.4670 - 0.6492}{7.4996} = 0.9091 \\ h_{6s} = h_f + x_{6s} h_{fg} = 191.81 + (0.9091)(2392.1) = 2366.4 \text{ kJ/kg} \end{matrix}$$

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$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s})$$
$$= 3264.5 - (0.80)(3264.5 - 2366.4)$$
$$= 2546.0 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_6 = 10 \text{ kPa} \\ h_6 = 2546.5 \text{ kJ/kg} \end{matrix}\right\} x_6 = 0.9842$$

Moisture Percentage $= 1 - x_6 = 1 - 0.9842 = 0.0158 =$ **1.6%**

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_s (h_3 - h_2) + \dot{m}_s (h_5 - h_4) = \dot{m}_{\text{air}} (h_{10} - h_{11})$$
$$(1.15)[(3346.5 - 199.37) + (3264.5 - h_4)] = (10)(871.98 - 475.62) \longrightarrow h_4 = 2965.0 \text{ kJ/kg}$$

Also,

$$\left.\begin{matrix} P_3 = 6 \text{ MPa} \\ T_3 = ? \end{matrix}\right\} \begin{matrix} h_3 = \\ s_3 = \end{matrix} \qquad \left.\begin{matrix} P_4 = 1 \text{ MPa} \\ s_{4s} = s_3 \end{matrix}\right\} h_{4s} =$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

The temperature at the inlet of the high-pressure turbine may be obtained by a trial-error approach or using EES from the above relations. The answer is $T_3 =$ **468.0°C**. Then, the enthalpy at state 3 becomes: $h_3 = 3346.5$ kJ/kg

(*c*) $\dot{W}_{\text{T,gas}} = \dot{m}_{\text{air}} (h_9 - h_{10}) = (10 \text{ kg/s})(1304.8 - 871.98) \text{ kJ/kg} = 4328 \text{ kW}$

$\dot{W}_{\text{C,gas}} = \dot{m}_{\text{air}} (h_8 - h_7) = (10 \text{ kg/s})(557.21 - 288.50) \text{ kJ/kg} = 2687 \text{ kW}$

$\dot{W}_{\text{net,gas}} = \dot{W}_{\text{T,gas}} - \dot{W}_{\text{C,gas}} = 4328 - 2687 = 1641 \text{ kW}$

$\dot{W}_{\text{T,steam}} = \dot{m}_{\text{s}} (h_3 - h_4 + h_5 - h_6) = (1.15 \text{ kg/s})(3346.5 - 2965.0 + 3264.5 - 2546.0) \text{ kJ/kg} = 1265 \text{ kW}$

$\dot{W}_{\text{P,steam}} = \dot{m}_{\text{s}} w_{pump} = (1.15 \text{ kg/s})(7.564) \text{ kJ/kg} = 8.7 \text{ kW}$

$\dot{W}_{\text{net,steam}} = \dot{W}_{\text{T,steam}} - \dot{W}_{\text{P,steam}} = 1265 - 8.7 = 1256 \text{ kW}$

$\dot{W}_{\text{net,plant}} = \dot{W}_{\text{net,gas}} + \dot{W}_{\text{net,steam}} = 1641 + 1256 =$ **2897 kW**

(*d*) $\dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_9 - h_8) = (10 \text{ kg/s})(1304.8 - 557.21) \text{ kJ/kg} = 7476 \text{ kW}$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,plant}}}{\dot{Q}_{\text{in}}} = \frac{2897 \text{ kW}}{7476 \text{ kW}} = 0.388 = \textbf{38.8\%}$$

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Special Topic: Binary Vapor Cycles

10-91C In binary vapor power cycles, both cycles are vapor cycles. In the combined gas-steam power cycle, one of the cycles is a gas cycle.

10-92C Binary power cycle is a cycle which is actually a combination of two cycles; one in the high temperature region, and the other in the low temperature region. Its purpose is to increase thermal efficiency.

10-93C Steam is not an ideal fluid for vapor power cycles because its critical temperature is low, its saturation dome resembles an inverted V, and its condenser pressure is too low.

10-94C Because mercury has a high critical temperature, relatively low critical pressure, but a very low condenser pressure. It is also toxic, expensive, and has a low enthalpy of vaporization.

10-95 Consider the heat exchanger of a binary power cycle. The working fluid of the topping cycle (cycle A) enters the heat exchanger at state 1 and leaves at state 2. The working fluid of the bottoming cycle (cycle B) enters at state 3 and leaves at state 4. Neglecting any changes in kinetic and potential energies, and assuming the heat exchanger is well-insulated, the steady-flow energy balance relation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$

$$\dot{m}_A h_2 + \dot{m}_B h_4 = \dot{m}_A h_1 + \dot{m}_B h_3 \text{ or } \dot{m}_A (h_2 - h_1) = \dot{m}_B (h_3 - h_4)$$

Thus,

$$\frac{\dot{m}_A}{\dot{m}_B} = \frac{h_3 - h_4}{h_2 - h_1}$$

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Review Problems

10-96 A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The thermal efficiency of the cycle is to be compared when it is operated so that the liquid enters the pump as a saturated liquid against that when the liquid enters as a subcooled liquid.

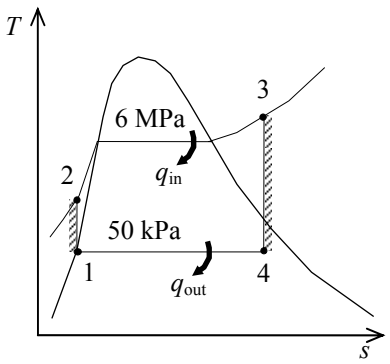
determined power produced by the turbine and consumed by the pump are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,50\,\text{kPa}} = 340.54\,\text{kJ/kg}$$
$$\nu_1 = \nu_{f@\,20\,\text{kPa}} = 0.001030\,\text{m}^3/\text{kg}$$

$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001030\,\text{m}^3/\text{kg})(6000-50)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 6.13\,\text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 340.54 + 6.13 = 346.67\,\text{kJ/kg}$$



$$\left.\begin{array}{l} P_3 = 6000\,\text{kPa} \\ T_3 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3658.8\,\text{kJ/kg} \\ s_3 = 7.1693\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 50\,\text{kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{7.1693 - 1.0912}{6.5019} = 0.9348 \\ h_4 = h_f + x_4 h_{fg} = 340.54 + (0.9348)(2304.7) = 2495.0\,\text{kJ/kg} \end{array}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3658.8 - 346.67 = 3312.1\,\text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2495.0 - 340.54 = 2154.5\,\text{kJ/kg}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2154.5}{3312.1} = \mathbf{0.3495}$$

When the liquid enters the pump 11.3°C cooler than a saturated liquid at the condenser pressure, the enthalpies become

$$\left.\begin{array}{l} P_1 = 50\,\text{kPa} \\ T_1 = T_{\text{sat@}\,50\,\text{kPa}} - 11.3 = 81.3 - 11.3 = 70°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@\,70°\text{C}} = 293.07\,\text{kJ/kg} \\ \nu_1 \cong \nu_{f@\,70°\text{C}} = 0.001023\,\text{m}^3/\text{kg} \end{array}$$

$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001023\,\text{m}^3/\text{kg})(6000-50)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 6.09\,\text{kJ/kg}$$

$$h_2 = h_1 + w_{\text{p,in}} = 293.07 + 6.09 = 299.16\,\text{kJ/kg}$$

Then,

$$q_{\text{in}} = h_3 - h_2 = 3658.8 - 299.16 = 3359.6\,\text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2495.0 - 293.09 = 2201.9\,\text{kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2201.9}{3359.6} = \mathbf{0.3446}$$

The thermal efficiency slightly decreases as a result of subcooling at the pump inlet.

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10-97E A geothermal power plant operating on the simple Rankine cycle using an organic fluid as the working fluid is considered. The exit temperature of the geothermal water from the vaporizer, the rate of heat rejection from the working fluid in the condenser, the mass flow rate of geothermal water at the preheater, and the thermal efficiency of the Level I cycle of this plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The exit temperature of geothermal water from the vaporizer is determined from the steady-flow energy balance on the geothermal water (brine),

$$\dot{Q}_{\text{brine}} = \dot{m}_{\text{brine}} c_p (T_2 - T_1)$$
$$-22{,}790{,}000 \text{ Btu/h} = (384{,}286 \text{ lbm/h})(1.03 \text{ Btu/lbm} \cdot °\text{F})(T_2 - 325°\text{F})$$
$$T_2 = \mathbf{267.4°F}$$

(*b*) The rate of heat rejection from the working fluid to the air in the condenser is determined from the steady-flow energy balance on air,

$$\dot{Q}_{\text{air}} = \dot{m}_{\text{air}} c_p (T_9 - T_8)$$
$$= (4{,}195{,}100 \text{ lbm/h})(0.24 \text{ Btu/lbm} \cdot °\text{F})(84.5 - 55°\text{F})$$
$$= \mathbf{29.7\ MBtu/h}$$

(*c*) The mass flow rate of geothermal water at the preheater is determined from the steady-flow energy balance on the geothermal water,

$$\dot{Q}_{\text{geo}} = \dot{m}_{\text{geo}} c_p (T_{\text{out}} - T_{\text{in}})$$
$$-11{,}140{,}000 \text{ Btu/h} = \dot{m}_{\text{geo}} (1.03 \text{ Btu/lbm} \cdot °\text{F})(154.0 - 211.8°\text{F})$$
$$\dot{m}_{\text{geo}} = \mathbf{187{,}120\ \ lbm/h}$$

(*d*) The rate of heat input is

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{vaporizer}} + \dot{Q}_{\text{reheater}} = 22{,}790{,}000 + 11{,}140{,}000$$
$$= 33{,}930{,}000 \text{ Btu/h}$$

and

$$\dot{W}_{\text{net}} = 1271 - 200 = 1071 \text{ kW}$$

Then,

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1071 \text{ kW}}{33{,}930{,}000 \text{ Btu/h}} \left(\frac{3412.14 \text{ Btu}}{1 \text{ kWh}} \right) = \mathbf{10.8\%}$$

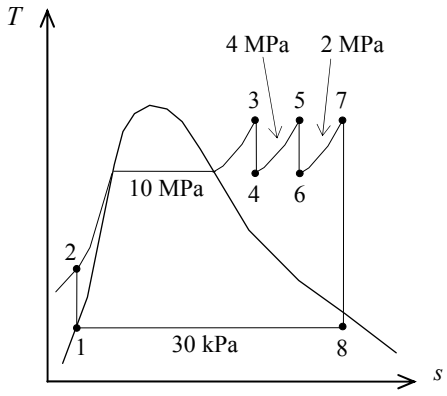
PROPRIETARY MATERIAL. © 2011 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-98 A steam power plant operating on an ideal Rankine cycle with two stages of reheat is considered. The thermal efficiency of the cycle and the mass flow rate of the steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,30\text{ kPa}} = 289.18\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,30\text{ kPa}} = 0.001022\ \text{m}^3/\text{kg}$$
$$w_{\text{p,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001022\ \text{m}^3/\text{kg})(10{,}000 - 30\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.19\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 289.18 + 10.19 = 299.37\text{ kJ/kg}$$



$$\left.\begin{matrix}P_3 = 10\text{ MPa}\\ T_3 = 550°\text{C}\end{matrix}\right\} \begin{matrix}h_3 = 3500.9\text{ kJ/kg}\\ s_3 = 6.7561\text{ kJ/kg}\cdot\text{K}\end{matrix}$$

$$\left.\begin{matrix}P_4 = 4\text{ MPa}\\ s_4 = s_3\end{matrix}\right\} h_4 = 3204.9\text{ kJ/kg}$$

$$\left.\begin{matrix}P_5 = 4\text{ MPa}\\ T_5 = 550°\text{C}\end{matrix}\right\} \begin{matrix}h_5 = 3559.7\text{ kJ/kg}\\ s_5 = 7.2335\text{ kJ/kg}\cdot\text{K}\end{matrix}$$

$$\left.\begin{matrix}P_6 = 2\text{ MPa}\\ s_6 = s_5\end{matrix}\right\} h_6 = 3321.1\text{ kJ/kg}$$

$$\left.\begin{matrix}P_7 = 2\text{ MPa}\\ T_7 = 550°\text{C}\end{matrix}\right\} \begin{matrix}h_7 = 3578.4\text{ kJ/kg}\\ s_7 = 7.5706\text{ kJ/kg}\cdot\text{K}\end{matrix}$$

$$\left.\begin{matrix}P_8 = 30\text{ kPa}\\ s_8 = s_7\end{matrix}\right\} \begin{matrix}x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{7.5706 - 0.9441}{6.8234} = 0.9711\\ h_8 = h_f + x_8 h_{fg} = 289.27 + (0.9711)(2335.3) = 2557.1\text{ kJ/kg}\end{matrix}$$

Then,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) + (h_7 - h_6)$$
$$= 3500.9 - 299.37 + 3559.7 - 3204.9 + 3578.4 - 3321.1 = 3813.7\text{ kJ/kg}$$
$$q_{\text{out}} = h_8 - h_1 = 2557.1 - 289.18 = 2267.9\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3813.7 - 2267.9 = 1545.8\text{ kJ/kg}$$

Thus,

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1545.8\text{ kJ/kg}}{3813.7\text{ kJ/kg}} = 0.4053 = \mathbf{40.5\%}$$

(*b*) The mass flow rate of the steam is then

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{75{,}000\text{ kJ/s}}{1545.8\text{ kJ/kg}} = \mathbf{48.5\ kg/s}$$

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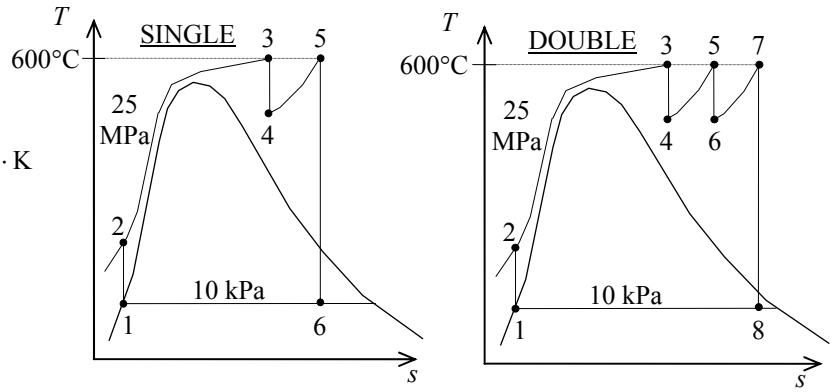
10-99 A steam power plant operating on the ideal Rankine cycle with reheating is considered. The reheat pressures of the cycle are to be determined for the cases of single and double reheat.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) Single Reheat: From the steam tables (Tables A-4, A-5, and A-6),

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ x_6 = 0.92 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6492 + (0.92)(7.4996) = 7.5488 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} T_5 = 600°\text{C} \\ s_5 = s_6 \end{array}\right\} P_5 = \mathbf{2780 \text{ kPa}}$$



(*b*) Double Reheat:

$$\left.\begin{array}{l} P_3 = 25 \text{ MPa} \\ T_3 = 600°\text{C} \end{array}\right\} s_3 = 6.3637 \text{ kJ/kg} \cdot \text{K}$$

$$\begin{array}{l} P_4 = P_x \\ s_4 = s_3 \end{array} \quad \text{and} \quad \begin{array}{l} P_5 = P_x \\ T_5 = 600°\text{C} \end{array}$$

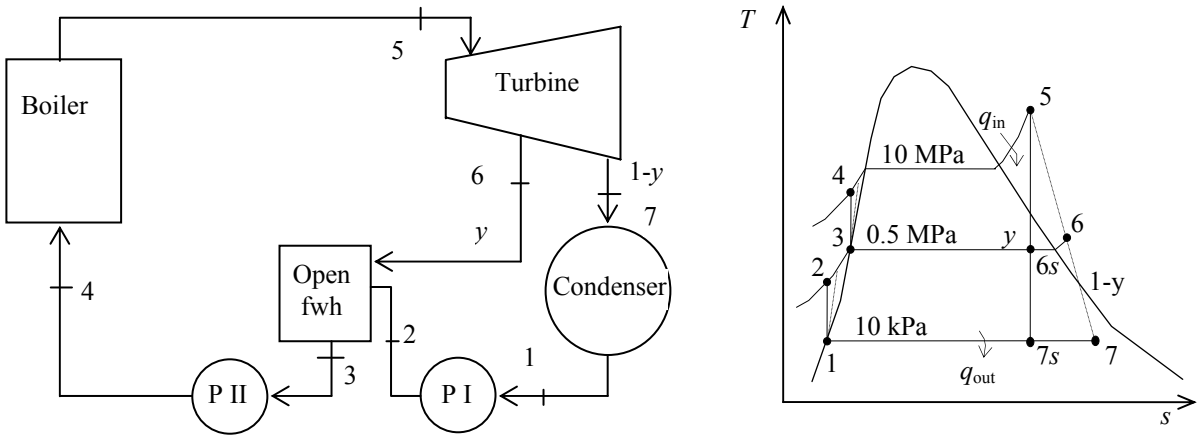
Any pressure P_x selected between the limits of 25 MPa and 2.78 MPa will satisfy the requirements, and can be used for the double reheat pressure.

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10-100 An 150-MW steam power plant operating on a regenerative Rankine cycle with an open feedwater heater is considered. The mass flow rate of steam through the boiler, the thermal efficiency of the cycle, and the irreversibility associated with the regeneration process are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis



(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\,\text{kPa}} = 191.81\,\text{kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,10\,\text{kPa}} = 0.00101\,\text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \nu_1(P_2 - P_1)/\eta_p \\ &= (0.00101\,\text{m}^3/\text{kg})(500-10\,\text{kPa})\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 0.52\,\text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.52 = 192.33\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.5\,\text{MPa} \\ \text{satliquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.5\,\text{MPa}} = 640.09\,\text{kJ/kg} \\ \nu_3 = \nu_{f\,@\,0.5\,\text{MPa}} = 0.001093\,\text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= \nu_3(P_4 - P_3)/\eta_p \\ &= (0.001093\ \text{m}^3/\text{kg})(10{,}000 - 500\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 10.93\ \text{kJ/kg} \end{aligned}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.93 = 651.02\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\ \text{MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\ \text{kJ/kg} \\ s_5 = 6.5995\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$x_{6s} = \frac{s_{6s} - s_f}{s_{fg}} = \frac{6.5995 - 1.8604}{4.9603} = 0.9554$$
$$\left.\begin{array}{l} P_{6s} = 0.5\ \text{MPa} \\ s_{6s} = s_5 \end{array}\right\} h_{6s} = h_f + x_{6s}h_{fg} = 640.09 + (0.9554)(2108.0) = 2654.1\ \text{kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow \begin{aligned} h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3375.1 - (0.80)(3375.1 - 2654.1) \\ &= 2798.3\ \text{kJ/kg} \end{aligned}$$

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$$x_{7s} = \frac{s_{7s} - s_f}{s_{fg}} = \frac{6.5995 - 0.6492}{7.4996} = 0.7934$$

$$\left.\begin{array}{l} P_{7s} = 10 \text{ kPa} \\ s_{7s} = s_5 \end{array}\right\} h_{7s} = h_f + x_{7s} h_{fg} = 191.81 + (0.7934)(2392.1) = 2089.7 \text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_7}{h_5 - h_{7s}} \longrightarrow h_7 = h_5 - \eta_T (h_5 - h_{7s}) = 3375.1 - (0.80)(3375.1 - 2089.7) = 2346.8 \text{ kJ/kg}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{640.09 - 192.33}{2798.3 - 192.33} = 0.1718$$

Then,
$$q_{\text{in}} = h_5 - h_4 = 3375.1 - 651.02 = 2724.1 \text{ kJ/kg}$$
$$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.1718)(2346.8 - 191.81) = 1784.7 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2724.1 - 1784.7 = 939.4 \text{ kJ/kg}$$

and

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150{,}000 \text{ kJ/s}}{939.4 \text{ kJ/kg}} = \mathbf{159.7 \text{ kg/s}}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1784.7 \text{ kJ/kg}}{2724.1 \text{ kJ/kg}} = \mathbf{34.5\%}$$

Also,

$$\left.\begin{array}{l} P_6 = 0.5 \text{ MPa} \\ h_6 = 2798.3 \text{ kJ/kg} \end{array}\right\} s_6 = 6.9453 \text{ kJ/kg} \cdot \text{K}$$

$$s_3 = s_{f\,@\,0.5 \text{ MPa}} = 1.8604 \text{ kJ/kg} \cdot \text{K}$$
$$s_2 = s_1 = s_{f\,@\,10 \text{ kPa}} = 0.6492 \text{ kJ/kg} \cdot \text{K}$$

Then the irreversibility (or exergy destruction) associated with this regeneration process is

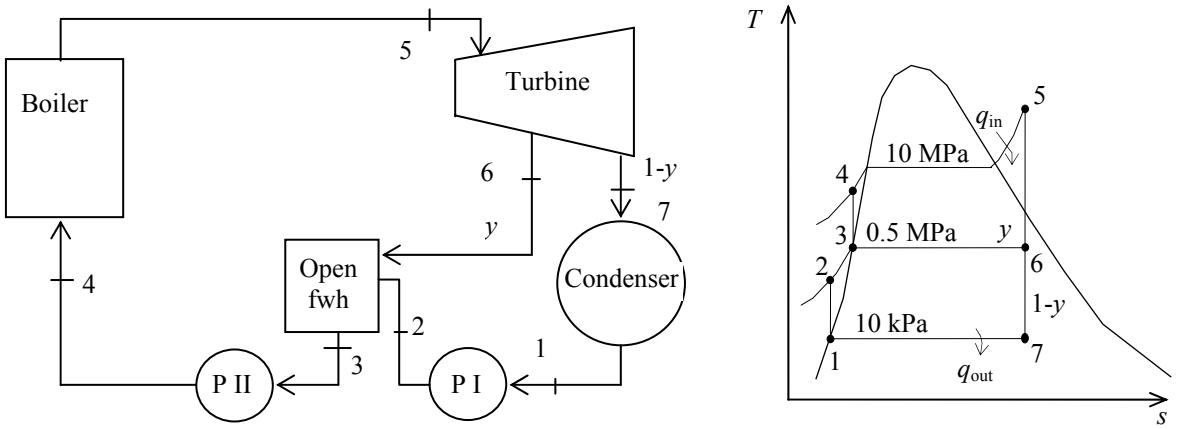
$$i_{\text{regen}} = T_0 s_{\text{gen}} = T_0 \left(\sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_L}^{\nearrow 0} \right) = T_0 \left[s_3 - y s_6 - (1 - y) s_2 \right]$$
$$= (303 \text{ K})\left[1.8604 - (0.1718)(6.9453) - (1 - 0.1718)(0.6492) \right]$$
$$= \mathbf{39.25 \text{ kJ/kg}}$$

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10-101 An 150-MW steam power plant operating on an ideal regenerative Rankine cycle with an open feedwater heater is considered. The mass flow rate of steam through the boiler, the thermal efficiency of the cycle, and the irreversibility associated with the regeneration process are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis



(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \nu_1(P_2 - P_1)$$
$$= (0.00101\ \text{m}^3/\text{kg})(500 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 0.50\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.50 = 192.30\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.5\text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.5\text{ MPa}} = 640.09\text{ kJ/kg} \\ \nu_3 = \nu_{f\,@\,0.5\text{ MPa}} = 0.001093\ \text{m}^3/\text{kg} \end{array}$$

$$w_{\text{pII,in}} = \nu_3(P_4 - P_3)$$
$$= (0.001093\ \text{m}^3/\text{kg})(10{,}000 - 500\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 10.38\text{ kJ/kg}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.38 = 650.47\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\text{ kJ/kg} \\ s_5 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.5\text{ MPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.5995 - 1.8604}{4.9603} = 0.9554 \\ h_6 = h_f + x_6 h_{fg} = 640.09 + (0.9554)(2108.0) = 2654.1\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_7 = 10\text{ kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.5995 - 0.6492}{7.4996} = 0.7934 \\ h_7 = h_f + x_7 h_{fg} = 191.81 + (0.7934)(2392.1) = 2089.7\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\,\nearrow 0\,(\text{steady})} = 0 \ \rightarrow\ \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for y,

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$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{640.09 - 192.31}{2654.1 - 192.31} = 0.1819$$

Then, $q_{\text{in}} = h_5 - h_4 = 3375.1 - 650.47 = 2724.6 \text{ kJ/kg}$

$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.1819)(2089.7 - 191.81) = 1552.7 \text{ kJ/kg}$

$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2724.6 - 1552.7 = 1172.0 \text{ kJ/kg}$

and $\dot{m} = \dfrac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \dfrac{150{,}000 \text{ kJ/s}}{1171.9 \text{ kJ/kg}} = \mathbf{128.0 \text{ kg/s}}$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1552.7 \text{ kJ/kg}}{2724.7 \text{ kJ/kg}} = \mathbf{43.0\%}$$

Also,

$s_6 = s_5 = 6.5995 \text{ kJ/kg} \cdot \text{K}$

$s_3 = s_{f\,@\,0.5 \text{ MPa}} = 1.8604 \text{ kJ/kg} \cdot \text{K}$

$s_2 = s_1 = s_{f\,@\,10 \text{ kPa}} = 0.6492 \text{ kJ/kg} \cdot \text{K}$

Then the irreversibility (or exergy destruction) associated with this regeneration process is

$$i_{\text{regen}} = T_0 s_{\text{gen}} = T_0 \left(\sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_L}^{\nearrow 0} \right) = T_0 \left[s_3 - y s_6 - (1 - y) s_2 \right]$$

$$= (303 \text{ K})\left[1.8604 - (0.1819)(6.5995) - (1 - 0.1819)(0.6492) \right]$$

$$= \mathbf{39.0 \text{ kJ/kg}}$$

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10-102 An ideal reheat-regenerative Rankine cycle with one open feedwater heater is considered. The fraction of steam extracted for regeneration and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

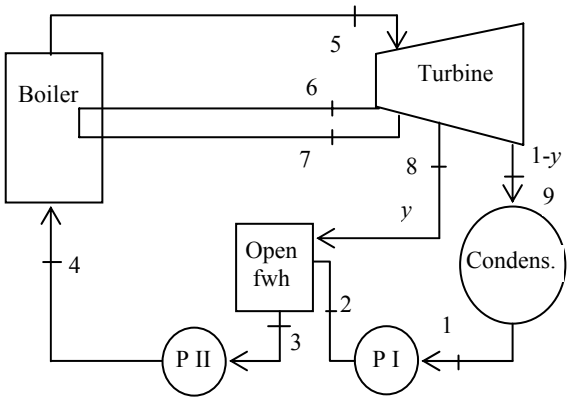
Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,15\text{ kPa}} = 225.94 \text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,15\text{ kPa}} = 0.001014 \text{ m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \nu_1(P_2 - P_1)$$
$$= (0.001014 \text{ m}^3/\text{kg})(600 - 15 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.59 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 225.94 + 0.59 = 226.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.6 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.6\text{ MPa}} = 670.38 \text{ kJ/kg} \\ \nu_3 = \nu_{f\,@\,0.6\text{ MPa}} = 0.001101 \text{ m}^3/\text{kg} \end{array}$$

$$w_{\text{pII,in}} = \nu_3(P_4 - P_3)$$
$$= (0.001101 \text{ m}^3/\text{kg})(10{,}000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.35 \text{ kJ/kg}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 670.38 + 10.35 = 680.73 \text{ kJ/kg}$$

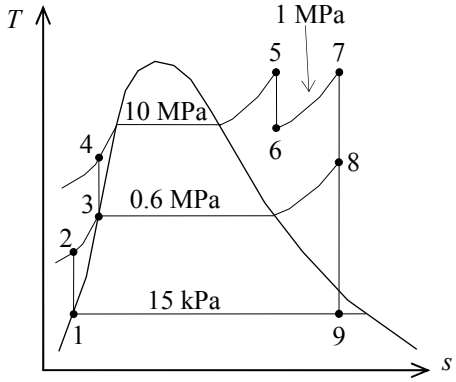


$$\left.\begin{array}{l} P_5 = 10 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1 \text{ kJ/kg} \\ s_5 = 6.5995 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1.0 \text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2783.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 1.0 \text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1 \text{ kJ/kg} \\ s_7 = 7.7642 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 0.6 \text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_8 = 3310.2 \text{ kJ/kg}$$



$$\left.\begin{array}{l} P_9 = 15 \text{ kPa} \\ s_9 = s_7 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{7.7642 - 0.7549}{7.2522} = 0.9665 \\ h_9 = h_f + x_9 h_{fg} = 225.94 + (0.9665)(2372.3) = 2518.8 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\,\nearrow 0\,\text{(steady)}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_8 + (1 - y) h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_8 - h_2} = \frac{670.38 - 226.53}{3310.2 - 226.53} = \mathbf{0.144}$$

(*b*) The thermal efficiency is determined from

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (3375.1 - 680.73) + (3479.1 - 2783.8) = 3389.7 \text{ kJ/kg}$$
$$q_{\text{out}} = (1 - y)(h_9 - h_1) = (1 - 0.1440)(2518.8 - 225.94) = 1962.7 \text{ kJ/kg}$$

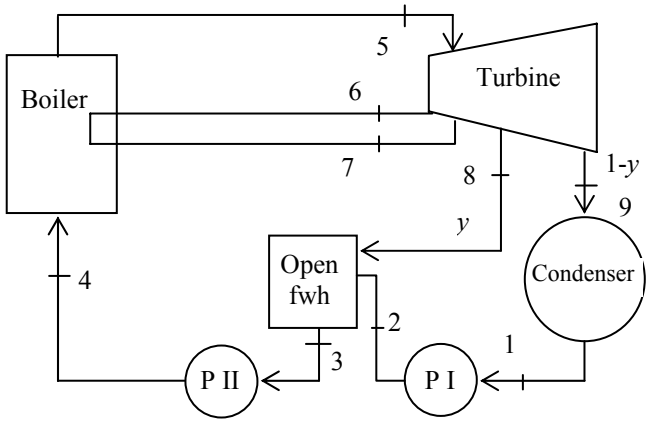
and
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1962.7 \text{ kJ/kg}}{3389.7 \text{ kJ/kg}} = \mathbf{42.1\%}$$

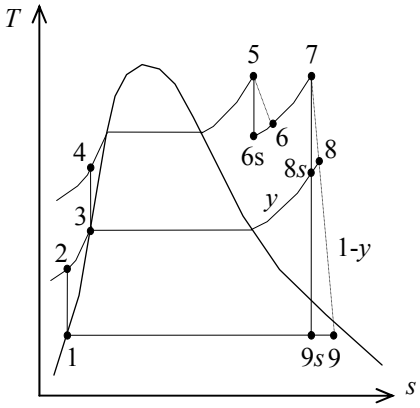
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10-103 A nonideal reheat-regenerative Rankine cycle with one open feedwater heater is considered. The fraction of steam extracted for regeneration and the thermal efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis





(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,15\text{ kPa}} = 225.94\text{ kJ/kg}$$
$$\nu_1 = \nu_{f@\,15\text{ kPa}} = 0.001014\text{ m}^3\text{/kg}$$

$$w_{pI,\text{in}} = \nu_1(P_2 - P_1)$$
$$= (0.001014\text{ m}^3\text{/kg})(600 - 15\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.59\text{ kJ/kg}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 225.94 + 0.59 = 226.54\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.6\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.6\text{ MPa}} = 670.38\text{ kJ/kg} \\ \nu_3 = \nu_{f@\,0.6\text{ MPa}} = 0.001101\text{ m}^3\text{/kg} \end{array}$$

$$w_{\text{pII,in}} = \nu_3(P_4 - P_3)$$
$$= (0.001101\text{ m}^3\text{/kg})(10{,}000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.35\text{ kJ/kg}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 670.38 + 10.35 = 680.73\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\text{ kJ/kg} \\ s_5 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{6s} = 1.0\text{ MPa} \\ s_{6s} = s_5 \end{array}\right\} h_{6s} = 2783.8\text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T(h_5 - h_{6s})$$
$$= 3375.1 - (0.84)(3375.1 - 2783.8)$$
$$= 2878.4\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 1.0\text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1\text{ kJ/kg} \\ s_7 = 7.7642\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{8s} = 0.6\text{ MPa} \\ s_{8s} = s_7 \end{array}\right\} h_{8s} = 3310.2\text{ kJ/kg}$$

$$\eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} \longrightarrow h_8 = h_7 - \eta_T(h_7 - h_{8s}) = 3479.1 - (0.84)(3479.1 - 3310.2)$$
$$= 3337.2\text{ kJ/kg}$$

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$$\left.\begin{array}{l} P_{9s} = 15 \text{ kPa} \\ s_{9s} = s_7 \end{array}\right\} \begin{array}{l} x_{9s} = \dfrac{s_{9s} - s_f}{s_{fg}} = \dfrac{7.7642 - 0.7549}{7.2522} = 0.9665 \\ h_{9s} = h_f + x_{9s} h_{fg} = 225.94 + (0.9665)(2372.3) = 2518.8 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_7 - h_9}{h_7 - h_{9s}} \longrightarrow \begin{array}{l} h_9 = h_7 - \eta_T (h_7 - h_{9s}) = 3479.1 - (0.84)(3479.1 - 2518.8) \\ \quad = 2672.5 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\begin{aligned} \dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \\ \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \sum \dot{m}_i h_i = \sum \dot{m}_e h_e &\longrightarrow \dot{m}_8 h_8 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_8 + (1 - y) h_2 = 1(h_3) \end{aligned}$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_8 - h_2} = \frac{670.38 - 226.53}{3335.3 - 226.53} = \mathbf{0.1427}$$

(*b*) The thermal efficiency is determined from

$$\begin{aligned} q_{\text{in}} &= (h_5 - h_4) + (h_7 - h_6) \\ &= (3375.1 - 680.73) + (3479.1 - 2878.4) = 3295.1 \text{ kJ/kg} \\ q_{\text{out}} &= (1 - y)(h_9 - h_1) = (1 - 0.1427)(2672.5 - 225.94) = 2097.2 \text{ kJ/kg} \end{aligned}$$

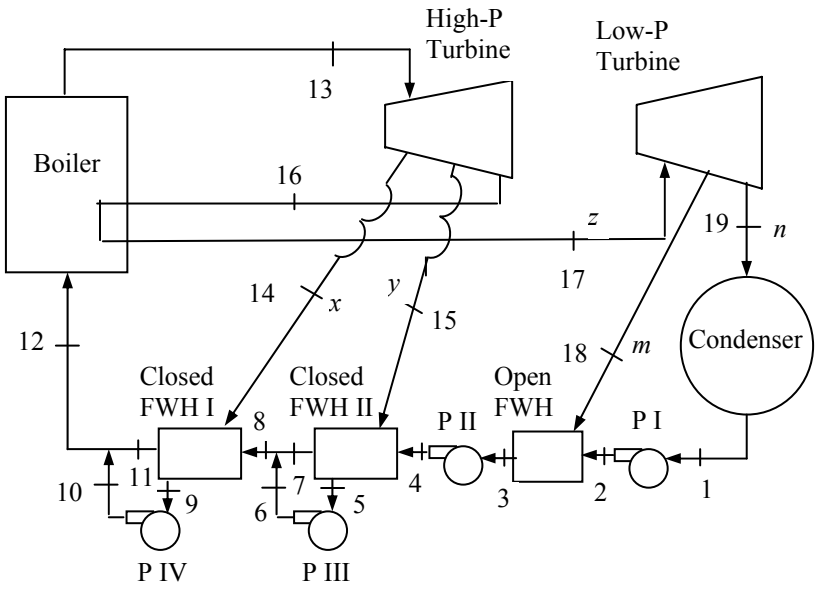
and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2097.2 \text{ kJ/kg}}{3295.1 \text{ kJ/kg}} = \mathbf{36.4\%}$$

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10-104 A steam power plant operating on the ideal reheat-regenerative Rankine cycle with three feedwater heaters is considered. Various items for this system per unit of mass flow rate through the boiler are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis The compression processes in the pumps and the expansion processes in the turbines are isentropic. Also, the state of water at the inlet of pumps is saturated liquid. Then, from the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = 168.75\ \text{kJ/kg}$$
$$h_2 = 168.84\ \text{kJ/kg}$$
$$h_3 = 417.51\ \text{kJ/kg}$$
$$h_4 = 419.28\ \text{kJ/kg}$$
$$h_5 = 884.46\ \text{kJ/kg}$$
$$h_6 = 885.86\ \text{kJ/kg}$$
$$h_9 = 1008.3\ \text{kJ/kg}$$
$$h_{10} = 1011.8\ \text{kJ/kg}$$

$$h_{13} = 3423.1\ \text{kJ/kg}$$
$$h_{14} = 3204.5\ \text{kJ/kg}$$
$$h_{15} = 3063.6\ \text{kJ/kg}$$
$$h_{16} = 2871.0\ \text{kJ/kg}$$
$$h_{17} = 3481.3\ \text{kJ/kg}$$
$$h_{18} = 2891.5\ \text{kJ/kg}$$
$$h_{19} = 2454.7\ \text{kJ/kg}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure. Then,

$$\left.\begin{array}{l} P_7 = 1800\ \text{kPa} \\ T_7 = T_5 = 207.1°\text{C} \end{array}\right\} \quad h_7 = 884.91\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_{11} = 3000\ \text{kPa} \\ T_{11} = T_9 = 233.9°\text{C} \end{array}\right\} \quad h_{11} = 1008.8\ \text{kJ/kg}$$

Enthalpies at other states and the fractions of steam extracted from the turbines can be determined from mass and energy balances on cycle components as follows:

Mass Balances:

$$x + y + z = 1$$
$$m + n = z$$

Open feedwater heater:

$$mh_{18} + nh_2 = zh_3$$

Closed feedwater heater-II:

$$zh_4 + yh_{15} = zh_7 + yh_5$$

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Closed feedwater heater-I:

$$(y+z)h_8 + xh_{14} = (y+z)h_{11} + xh_9$$

Mixing chamber after closed feedwater heater II:

$$zh_7 + yh_6 = (y+z)h_8$$

Mixing chamber after closed feedwater heater I:

$$xh_{10} + (y+z)h_{11} = 1h_{12}$$

Substituting the values and solving the above equations simultaneously using EES, we obtain

$$h_8 = 885.08 \text{ kJ/kg}$$
$$h_{12} = 1009.0 \text{ kJ/kg}$$
$$x = \mathbf{0.05334}$$
$$y = \mathbf{0.1667}$$
$$z = 0.78000$$
$$m = \mathbf{0.07124}$$
$$n = \mathbf{0.70882}$$

Note that these values may also be obtained by a hand solution by using the equations above with some rearrangements and substitutions. Other results of the cycle are

$$w_{\text{T,out,HP}} = x(h_{13} - h_{14}) + y(h_{13} - h_{15}) + z(h_{13} - h_{16}) = \mathbf{502.3 \text{ kJ/kg}}$$
$$w_{\text{T,out,LP}} = m(h_{17} - h_{18}) + n(h_{17} - h_{19}) = \mathbf{769.6 \text{ kJ/kg}}$$
$$q_{\text{in}} = h_{13} - h_{12} + z(h_{17} - h_{16}) = \mathbf{2890 \text{ kJ/kg}}$$
$$q_{\text{out}} = n(h_{19} - h_1) = \mathbf{1620 \text{ kJ/kg}}$$
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1620}{2890} = 0.4394 = \mathbf{43.9\%}$$

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10-105 The optimum bleed pressure for the open feedwater heater that maximizes the thermal efficiency of the cycle is to be determined using EES.

Analysis The EES program used to solve this problem as well as the solutions are given below.

```
"Given"
P_boiler=6000 [kPa]
P_cfwh1=3000 [kPa]
P_cfwh2=1800 [kPa]
P_reheat=800 [kPa]
"P_ofwh=100 [kPa]"
P_condenser=7.5 [kPa]
T_turbine=500 [C]

"Analysis"
Fluid$='steam_iapws'

"turbines"
h[13]=enthalpy(Fluid$, P=P_boiler, T=T_turbine)
s[13]=entropy(Fluid$, P=P_boiler, T=T_turbine)
h[14]=enthalpy(Fluid$, P=P_cfwh1, s=s[13])
h[15]=enthalpy(Fluid$, P=P_cfwh2, s=s[13])
h[16]=enthalpy(Fluid$, P=P_reheat, s=s[13])
h[17]=enthalpy(Fluid$, P=P_reheat, T=T_turbine)
s[17]=entropy(Fluid$, P=P_reheat, T=T_turbine)
h[18]=enthalpy(Fluid$, P=P_ofwh, s=s[17])
h[19]=enthalpy(Fluid$, P=P_condenser, s=s[17])

"pump I"
h[1]=enthalpy(Fluid$, P=P_condenser, x=0)
v[1]=volume(Fluid$, P=P_condenser, x=0)
w_pI_in=v[1]*(P_ofwh-P_condenser)
h[2]=h[1]+w_pI_in

"pump II"
h[3]=enthalpy(Fluid$, P=P_ofwh, x=0)
v[3]=volume(Fluid$, P=P_ofwh, x=0)
w_pII_in=v[3]*(P_cfwh2-P_ofwh)
h[4]=h[3]+w_pII_in

"pump III"
h[5]=enthalpy(Fluid$, P=P_cfwh2, x=0)
T[5]=temperature(Fluid$, P=P_cfwh2, x=0)
v[5]=volume(Fluid$, P=P_cfwh2, x=0)
w_pIII_in=v[5]*(P_cfwh1-P_cfwh2)
h[6]=h[5]+w_pIII_in

"pump IV"
h[9]=enthalpy(Fluid$, P=P_cfwh1, x=0)
T[9]=temperature(Fluid$, P=P_cfwh1, x=0)
v[9]=volume(Fluid$, P=P_cfwh1, x=0)
w_p4_in=v[5]*(P_boiler-P_cfwh1)
h[10]=h[9]+w_p4_in

"Mass balances"
x+y+z=1
m+n=z
```

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"Open feedwater heater"
m*h[18]+n*h[2]=z*h[3]

"closed feedwater heater 2"
T[7]=T[5]
h[7]=enthalpy(Fluid$, P=P_cfwh1, T=T[7])
z*h[4]+y*h[15]=z*h[7]+y*h[5]

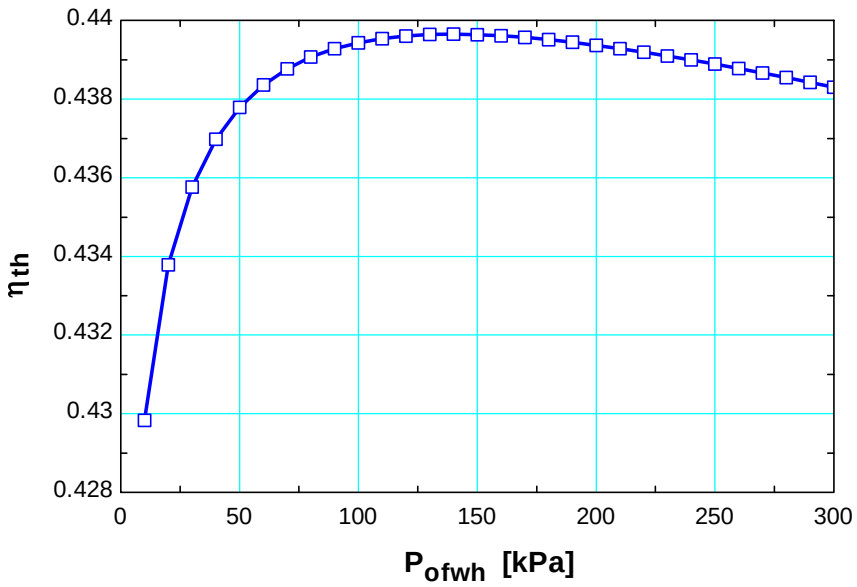
"closed feedwater heater 1"
T[11]=T[9]
h[11]=enthalpy(Fluid$, P=P_boiler, T=T[11])
(y+z)*h[8]+x*h[14]=(y+z)*h[11]+x*h[9]

"Mixing chamber after closed feedwater heater 2"
z*h[7]+y*h[6]=(y+z)*h[8]

"Mixing chamber after closed feedwater heater 1"
x*h[10]+(y+z)*h[11]=1*h[12]

"cycle"
w_T_out_high=x*(h[13]-h[14])+y*(h[13]-h[15])+z*(h[13]-h[16])
w_T_out_low=m*(h[17]-h[18])+n*(h[17]-h[19])
q_in=h[13]-h[12]+z*(h[17]-h[16])
q_out=n*(h[19]-h[1])
Eta_th=1-q_out/q_in

$P_{open\ fwh}$ [kPa]	η_{th}
10	0.429828
20	0.433780
30	0.435764
40	0.436978
50	0.437790
60	0.438359
70	0.438768
80	0.439065
90	0.439280
100	0.439432
110	0.439536
120	0.439602
130	0.439638
140	**0.439647**
150	0.439636
160	0.439608
170	0.439565
180	0.439509
190	0.439442
200	0.439367
210	0.439283
220	0.439192
230	0.439095
240	0.438993
250	0.438887
260	0.438776
270	0.438662
280	0.438544
290	0.438424
300	0.438301



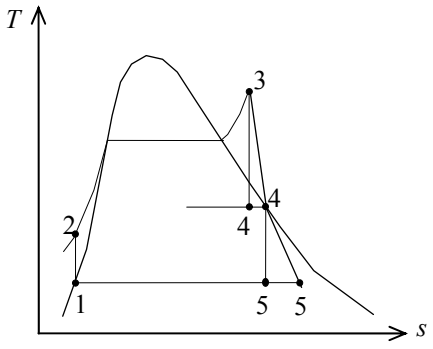
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10-106 A cogeneration plant is to produce power and process heat. There are two turbines in the cycle: a high-pressure turbine and a low-pressure turbine. The temperature, pressure, and mass flow rate of steam at the inlet of high-pressure turbine are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$\left.\begin{array}{l} P_4 = 1.4 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_4 = h_{g\,@\,1.4\text{ MPa}} = 2788.9 \text{ kJ/kg} \\ s_4 = s_{g\,@\,1.4\text{ MPa}} = 6.4675 \text{ kJ/kg}\cdot\text{K} \end{array}$$



$$\left.\begin{array}{l} P_5 = 10 \text{ kPa} \\ s_{5s} = s_4 \end{array}\right\} \begin{array}{l} x_{5s} = \dfrac{s_{4s} - s_f}{s_{fg}} = \dfrac{6.4675 - 0.6492}{7.4996} = 0.7758 \\ h_{5s} = h_f + x_{5s} h_{fg} \\ \quad = 191.81 + (0.7758)(2392.1) = 2047.6 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow \begin{array}{l} h_5 = h_4 - \eta_T (h_4 - h_{5s}) \\ \quad = 2788.9 - (0.60)(2788.9 - 2047.6) \\ \quad = 2344.1 \text{ kJ/kg} \end{array}$$

and

$$w_{\text{turb,low}} = h_4 - h_5 = 2788.9 - 2344.1 = 444.8 \text{ kJ/kg}$$

$$\dot{m}_{\text{low turb}} = \frac{\dot{W}_{\text{turb,II}}}{w_{\text{turb,low}}} = \frac{800 \text{ kJ/s}}{444.8 \text{ kJ/kg}} = 1.799 \text{ kg/s} = 107.9 \text{ kg/min}$$

Therefore ,

$$\dot{m}_{\text{total}} = 1000 + 108 = 1108 \text{ kg/min} = \mathbf{18.47 \text{ kg/s}}$$

$$w_{\text{turb,high}} = \frac{\dot{W}_{\text{turb},I}}{\dot{m}_{\text{high,turb}}} = \frac{1000 \text{ kJ/s}}{18.47 \text{ kg/s}} = 54.15 \text{ kJ/kg} = h_3 - h_4$$

$$h_3 = w_{\text{turb,high}} + h_4 = 54.15 + 2788.9 = 2843.0 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow \begin{array}{l} h_{4s} = h_3 - (h_3 - h_4)/\eta_T \\ \quad = 2843.0 - (2843.0 - 2788.9)/(0.75) \\ \quad = 2770.8 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{4s} = 1.4 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{h_{4s} - h_f}{h_{fg}} = \dfrac{2770.8 - 829.96}{1958.9} = 0.9908 \\ s_{4s} = s_f + x_{4s} s_{fg} = 2.2835 + (0.9908)(4.1840) = 6.4289 \text{ kJ/kg}\cdot\text{K} \end{array}$$

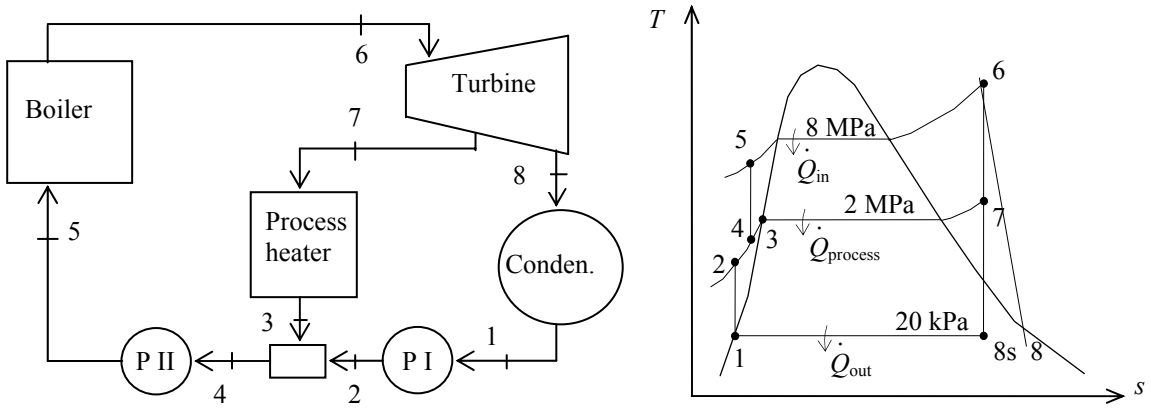
Then from the tables or the software, the turbine inlet temperature and pressure becomes

$$\left.\begin{array}{l} h_3 = 2843.0 \text{ kJ/kg} \\ s_3 = 6.4289 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} P_3 = \mathbf{2 \text{ MPa}} \\ T_3 = \mathbf{227.5°C} \end{array}$$

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10-107 A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The rate of process heat, the net power produced, and the utilization factor of the plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.42\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \nu_1(P_2 - P_1)/\eta_p \\ &= (0.001017\ \text{m}^3/\text{kg})(2000 - 20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/0.88 \\ &= 2.29\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 2.29 = 253.71\text{ kJ/kg}$$

$$h_3 = h_{f\,@\,2\text{ MPa}} = 908.47\text{ kJ/kg}$$

Mixing chamber:

$$\dot{m}_3 h_3 + \dot{m}_2 h_2 = \dot{m}_4 h_4$$
$$(4\text{ kg/s})(908.47\text{ kJ/kg}) + (11 - 4\text{ kg/s})(253.71\text{ kJ/kg})) = (11\text{ kg/s})h_4 \longrightarrow h_4 = 491.81\text{ kJ/kg}$$

$$\nu_4 \cong \nu_{f\,@\,h_f = 491.81\text{ kJ/kg}} = 0.001058\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{p\text{II,in}} &= \nu_4(P_5 - P_4)/\eta_p \\ &= (0.001058\ \text{m}^3/\text{kg})(8000 - 2000\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/0.88 \\ &= 7.21\text{ kJ/kg} \end{aligned}$$
$$h_5 = h_4 + w_{p\text{II,in}} = 491.81 + 7.21 = 499.02\text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 8\text{ MPa} \\ T_6 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3399.5\text{ kJ/kg} \\ s_6 &= 6.7266\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 2\text{ MPa} \\ s_7 &= s_6 \end{aligned}\right\} h_{7s} = 3000.4\text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} \longrightarrow h_7 = h_6 - \eta_T(h_6 - h_{7s}) = 3399.5 - (0.88)(3399.5 - 3000.4) = 3048.3\text{ kJ/kg}$$

$$\left.\begin{aligned} P_8 &= 20\text{ kPa} \\ s_8 &= s_6 \end{aligned}\right\} h_{8s} = 2215.5\text{ kJ/kg}$$

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$$\eta_T = \frac{h_6 - h_8}{h_6 - h_{8s}} \longrightarrow h_8 = h_6 - \eta_T (h_6 - h_{8s}) = 3399.5 - (0.88)(3399.5 - 2215.5) = 2357.6 \text{ kJ/kg}$$

Then,

$$\dot{Q}_{\text{process}} = \dot{m}_7 (h_7 - h_3) = (4 \text{ kg/s})(3048.3 - 908.47) \text{ kJ/kg} = \mathbf{8559\ kW}$$

(*b*) Cycle analysis:

$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_7 (h_6 - h_7) + \dot{m}_8 (h_6 - h_8) \\ &= (4 \text{ kg/s})(3399.5 - 3048.3)\text{kJ/kg} + (7 \text{ kg/s})(3399.5 - 2357.6)\text{kJ/kg} \\ &= 8698 \text{ kW} \end{aligned}$$

$$\dot{W}_{\text{p,in}} = \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{\text{pII,in}} = (7 \text{ kg/s})(2.29 \text{ kJ/kg}) + (11 \text{ kg/s})(7.21 \text{ kJ/kg}) = 95 \text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 8698 - 95 = \mathbf{8603\ kW}$$

(*c*) Then,

$$\dot{Q}_{\text{in}} = \dot{m}_5 (h_6 - h_5) = (11 \text{ kg/s})(3399.5 - 499.02) = 31{,}905 \text{ kW}$$

and

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{8603 + 8559}{31{,}905} = 0.538 = \mathbf{53.8\%}$$

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10-108E A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The thermal efficiency of the cycle is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable for Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 0.240$ Btu/lbm·R and $k = 1.4$ (Table A-2Ea).

Analysis Working around the topping cycle gives the following results:

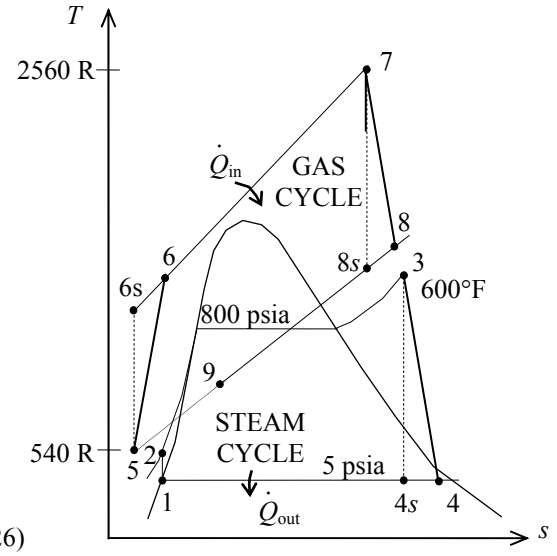
$$T_{6s} = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (540\text{ R})(10)^{0.4/1.4} = 1043\text{ R}$$

$$\eta_C = \frac{h_{6s}-h_5}{h_6-h_5} = \frac{c_p(T_{6s}-T_5)}{c_p(T_6-T_5)}$$

$$\longrightarrow T_6 = T_5 + \frac{T_{6s}-T_5}{\eta_C} = 540 + \frac{1043-540}{0.90} = 1099\text{ R}$$

$$T_{8s} = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (2560\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 1326\text{ R}$$

$$\eta_T = \frac{h_7-h_8}{h_7-h_{8s}} = \frac{c_p(T_7-T_8)}{c_p(T_7-T_{8s})} \longrightarrow T_8 = T_7 - \eta_T(T_7-T_{8s}) = 2560-(0.90)(2560-1326) = 1449\text{ R}$$



$$T_9 = T_{\text{sat @ 800 psia}} + 50 = 978.3\text{ R} + 50 = 1028\text{ R}$$

Fixing the states around the bottom steam cycle yields (Tables A-4E, A-5E, A-6E):

$$h_1 = h_{f@\,5\text{ psia}} = 130.18\text{ Btu/lbm}$$
$$\nu_1 = \nu_{f@\,5\text{ psia}} = 0.01641\text{ ft}^3\text{/lbm}$$

$$w_{\text{p,in}} = \nu_1(P_2-P_1) = (0.01641\text{ ft}^3\text{/lbm})(800-5)\text{psia}\left(\frac{1\text{ Btu}}{5.404\text{ psia}\cdot\text{ft}^3}\right) = 2.41\text{ Btu/lbm}$$
$$h_2 = h_1 + w_{\text{p,in}} = 130.18 + 2.41 = 132.59\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 800\text{ psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1270.9\text{ Btu/lbm} \\ s_3 = 1.4866\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 5\text{ psia} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 908.6\text{ Btu/lbm}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3-h_{4s}) = 1270.9-(0.95)(1270.9-908.6) = 926.7\text{ Btu/lbm}$$

The net work outputs from each cycle are

$$w_{\text{net, gas cycle}} = w_{\text{T,out}} - w_{\text{C,in}} = c_p(T_7-T_8) - c_p(T_6-T_5) = (0.240\text{ Btu/lbm}\cdot\text{R})(2560-1449-1099+540)\text{R} = 132.5\text{ Btu/lbm}$$

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$$\begin{aligned} w_{\text{net, steam cycle}} &= w_{\text{T,out}} - w_{\text{P,in}} \\ &= (h_3 - h_4) - w_{\text{P,in}} \\ &= (1270.9 - 926.7) - 2.41 \\ &= 341.8 \text{ Btu/lbm} \end{aligned}$$

An energy balance on the heat exchanger gives

$$\dot{m}_a c_p (T_8 - T_9) = \dot{m}_w (h_3 - h_2) \longrightarrow \dot{m}_w = \frac{c_p (T_8 - T_9)}{h_3 - h_2} \dot{m}_a = \frac{(0.240)(1449 - 1028)}{1270.9 - 132.59} = 0.08876 \dot{m}_a$$

That is, 1 lbm of exhaust gases can heat only 0.08876 lbm of water. Then the heat input, the heat output and the thermal efficiency are

$$q_{\text{in}} = \frac{\dot{m}_a}{\dot{m}_a} c_p (T_7 - T_6) = (0.240 \text{ Btu/lbm} \cdot \text{R})(2560 - 1099)\text{R} = 350.6 \text{ Btu/lbm}$$

$$\begin{aligned} q_{\text{out}} &= \frac{\dot{m}_a}{\dot{m}_a} c_p (T_9 - T_5) + \frac{\dot{m}_w}{\dot{m}_a} (h_4 - h_1) \\ &= 1 \times (0.240 \text{ Btu/lbm} \cdot \text{R})(1028 - 540)\text{R} + 0.08876 \times (926.7 - 130.18) \text{ Btu/lbm} \\ &= 187.8 \text{ Btu/lbm} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{187.8}{350.6} = \mathbf{0.4643}$$

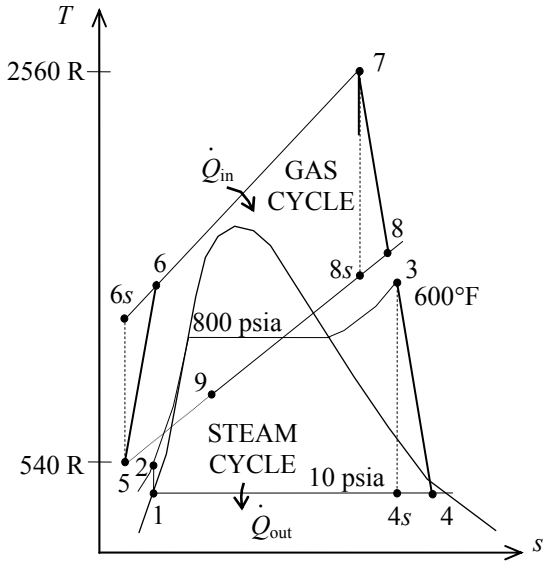
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10-109E A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The thermal efficiency of the cycle is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable fo Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis Working around the topping cycle gives the following results:



$$T_{6s} = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (540\text{ R})(10)^{0.4/1.4} = 1043\text{ R}$$

$$\eta_C = \frac{h_{6s}-h_5}{h_6-h_5} = \frac{c_p(T_{6s}-T_5)}{c_p(T_6-T_5)}$$

$$\longrightarrow T_6 = T_5 + \frac{T_{6s}-T_5}{\eta_C} = 540 + \frac{1043-540}{0.90} = 1099\text{ R}$$

$$T_{8s} = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (2560\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 1326\text{ R}$$

$$\eta_T = \frac{h_7-h_8}{h_7-h_{8s}} = \frac{c_p(T_7-T_8)}{c_p(T_7-T_{8s})} \longrightarrow T_8 = T_7 - \eta_T(T_7-T_{8s}) = 2560-(0.90)(2560-1326) = 1449\text{ R}$$

$$T_9 = T_{\text{sat @ 800 psia}} + 50 = 978.3\text{ R} + 50 = 1028\text{ R}$$

Fixing the states around the bottom steam cycle yields (Tables A-4E, A-5E, A-6E):

$$h_1 = h_{f@10\text{ psia}} = 161.25\text{ Btu/lbm}$$
$$\nu_1 = \nu_{f@10\text{ psia}} = 0.01659\text{ ft}^3\text{/lbm}$$

$$w_{\text{p,in}} = \nu_1(P_2-P_1) = (0.01659\text{ ft}^3\text{/lbm})(800-10)\text{psia}\left(\frac{1\text{ Btu}}{5.404\text{ psia}\cdot\text{ft}^3}\right) = 2.43\text{ Btu/lbm}$$
$$h_2 = h_1 + w_{\text{p,in}} = 161.25 + 2.43 = 163.7\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 800\text{ psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1270.9\text{ Btu/lbm} \\ s_3 = 1.4866\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ psia} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 946.6\text{ Btu/lbm}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3-h_{4s}) = 1270.9-(0.95)(1270.9-946.6) = 962.8\text{ Btu/lbm}$$

The net work outputs from each cycle are

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$$\begin{aligned} w_{\text{net, gas cycle}} &= w_{\text{T,out}} - w_{\text{C,in}} \\ &= c_p(T_7 - T_8) - c_p(T_6 - T_5) \\ &= (0.240 \text{ Btu/lbm} \cdot \text{R})(2560 - 1449 - 1099 + 540)\text{R} \\ &= 132.5 \text{ Btu/lbm} \end{aligned}$$

$$\begin{aligned} w_{\text{net, steam cycle}} &= w_{\text{T,out}} - w_{\text{P,in}} \\ &= (h_3 - h_4) - w_{\text{P,in}} \\ &= (1270.9 - 962.8) - 2.43 \\ &= 305.7 \text{ Btu/lbm} \end{aligned}$$

An energy balance on the heat exchanger gives

$$\dot{m}_a c_p(T_8 - T_9) = \dot{m}_w(h_3 - h_2) \longrightarrow \dot{m}_w = \frac{c_p(T_8 - T_9)}{h_3 - h_2}\dot{m}_a = \frac{(0.240)(1449 - 1028)}{1270.9 - 163.7} = 0.09126\dot{m}_a$$

That is, 1 lbm of exhaust gases can heat only 0.09126 lbm of water. Then the heat input, the heat output and the thermal efficiency are

$$q_{\text{in}} = \frac{\dot{m}_a}{\dot{m}_a} c_p(T_7 - T_6) = (0.240 \text{ Btu/lbm} \cdot \text{R})(2560 - 1099)\text{R} = 350.6 \text{ Btu/lbm}$$

$$\begin{aligned} q_{\text{out}} &= \frac{\dot{m}_a}{\dot{m}_a} c_p(T_9 - T_5) + \frac{\dot{m}_w}{\dot{m}_a}(h_4 - h_1) \\ &= 1 \times (0.240 \text{ Btu/lbm} \cdot \text{R})(1028 - 540)\text{R} + 0.09126 \times (962.8 - 161.25) \text{ Btu/lbm} \\ &= 190.3 \text{ Btu/lbm} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{190.3}{350.6} = \mathbf{0.4573}$$

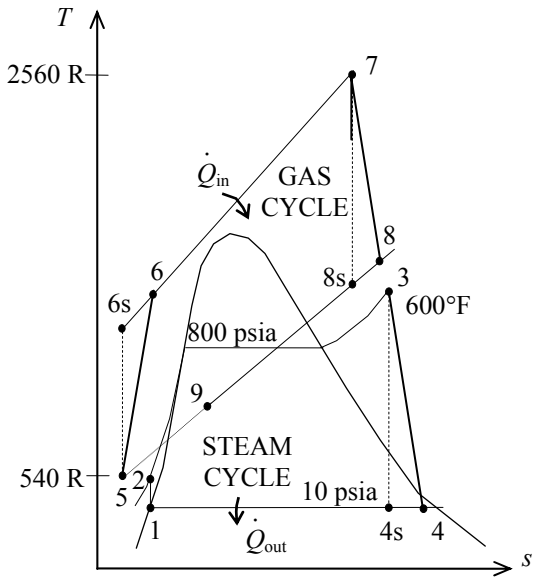
When the condenser pressure is increased from 5 psia to 10 psia, the thermal efficiency is decreased from 0.4643 to 0.4573.

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10-110E A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The cycle supplies a specified rate of heat to the buildings during winter. The mass flow rate of air and the net power output from the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable to Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 0.240$ Btu/lbm·R and $k = 1.4$ (Table A-2Ea).



Analysis The mass flow rate of water is

$$\dot{m}_w = \frac{\dot{Q}_{\text{buildings}}}{h_4 - h_1} = \frac{2\times10^6 \text{ Btu/h}}{(962.8 - 161.25)\text{ Btu/lbm}} = 2495 \text{ lbm/h}$$

The mass flow rate of air is then

$$\dot{m}_a = \frac{\dot{m}_w}{0.09126} = \frac{2495}{0.09126} = \mathbf{27{,}340 \text{ lbm/h}}$$

The power outputs from each cycle are

$$\begin{aligned}\dot{W}_{\text{net, gas cycle}} &= \dot{m}_a (w_{\text{T,out}} - w_{\text{C,in}}) \\ &= \dot{m}_a c_p (T_7 - T_8) - \dot{m}_a c_p (T_6 - T_5) \\ &= (27{,}340 \text{ lbm/h})(0.240 \text{ Btu/lbm}\cdot\text{R})(2560 - 1449 - 1099 + 540)\text{R}\left(\frac{1\text{ kW}}{3412.14 \text{ Btu/h}}\right) \\ &= 1062 \text{ kW}\end{aligned}$$

$$\begin{aligned}\dot{W}_{\text{net, steam cycle}} &= \dot{m}_a (w_{\text{T,out}} - w_{\text{P,in}}) \\ &= \dot{m}_a (h_3 - h_4 - w_{\text{P,in}}) \\ &= (2495 \text{ lbm/h})(1270.9 - 962.8 - 2.43)\left(\frac{1\text{ kW}}{3412.14 \text{ Btu/h}}\right) \\ &= 224 \text{ kW}\end{aligned}$$

The net electricity production by this cycle is then

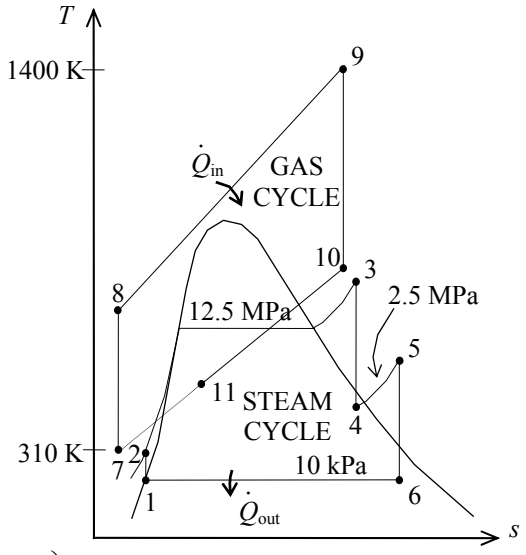
$$\dot{W}_{\text{net}} = 1062 + 224 = \mathbf{1286 \text{ kW}}$$

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10-111 A combined gas-steam power plant is considered. The topping cycle is an ideal gas-turbine cycle and the bottoming cycle is an ideal reheat Rankine cycle. The mass flow rate of air in the gas-turbine cycle, the rate of total heat input, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) The analysis of gas cycle yields



$$T_7 = 310\ \text{K} \longrightarrow \begin{matrix} h_7 = 310.24\ \text{kJ/kg} \\ P_{r_7} = 1.5546 \end{matrix}$$

$$P_{r_8} = \frac{P_8}{P_7} P_{r_7} = (12)(1.5546) = 18.66 \longrightarrow h_8 = 630.18\ \text{kJ/kg}$$

$$T_9 = 1400\ \text{K} \longrightarrow \begin{matrix} h_9 = 1515.42\ \text{kJ/kg} \\ P_{r_9} = 450.5 \end{matrix}$$

$$P_{r_{10}} = \frac{P_{10}}{P_9} P_{r_9} = \left(\frac{1}{12}\right)(450.5) = 37.54 \longrightarrow h_{10} = 768.38\ \text{kJ/kg}$$

$$T_{11} = 520\ \text{K} \longrightarrow h_{11} = 523.63\ \text{kJ/kg}$$

From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\ \text{kPa}} = 191.81\ \text{kJ/kg}$$

$$\nu_1 = \nu_{f\,@\,10\ \text{kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \nu_1 (P_2 - P_1) = (0.00101\ \text{m}^3/\text{kg})(12{,}500 - 10\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 12.62\ \text{kJ/kg}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 12.62 = 204.42\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 12.5\ \text{MPa} \\ T_3 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3343.6\ \text{kJ/kg} \\ s_3 = 6.4651\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 2.5\ \text{MPa} \\ s_4 = s_3 \end{matrix}\right\} h_4 = 2909.6\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_5 = 2.5\ \text{MPa} \\ T_5 = 550°\text{C} \end{matrix}\right\} \begin{matrix} h_5 = 3574.4\ \text{kJ/kg} \\ s_5 = 7.4653\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_6 = 10\ \text{kPa} \\ s_6 = s_5 \end{matrix}\right\} \begin{matrix} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.4653 - 0.6492}{7.4996} = 0.9089 \\ h_6 = h_f + x_6 h_{fg} = 191.81 + (0.9089)(2392.1) = 2365.8\ \text{kJ/kg} \end{matrix}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow \sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s (h_3 - h_2) = \dot{m}_{\text{air}} (h_{10} - h_{11})$$

$$\dot{m}_{\text{air}} = \frac{h_3 - h_2}{h_{10} - h_{11}} \dot{m}_s = \frac{3343.6 - 204.42}{768.38 - 523.63}(12\ \text{kg/s}) = \mathbf{153.9\ kg/s}$$

(*b*) The rate of total heat input is

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{Q}_{\text{air}} + \dot{Q}_{\text{reheat}} = \dot{m}_{\text{air}}(h_9 - h_8) + \dot{m}_{\text{reheat}}(h_5 - h_4) \\ &= (153.9\ \text{kg/s})(1515.42 - 630.18)\ \text{kJ/kg} + (12\ \text{kg/s})(3574.4 - 2909.6)\text{kJ/kg} \\ &= 144{,}200\ \text{kW} \\ &\cong \mathbf{1.44 \times 10^5\ \ kW} \end{aligned}$$

(*c*) The rate of heat rejection and the thermal efficiency are then

$$\begin{aligned} \dot{Q}_{\text{out}} &= \dot{Q}_{\text{out,air}} + \dot{Q}_{\text{out,steam}} = \dot{m}_{\text{air}}(h_{11} - h_7) + \dot{m}_s(h_6 - h_1) \\ &= (153.9\ \text{kg/s})(523.63 - 310.24)\ \text{kJ/kg} + (12\ \text{kg/s})(2365.8 - 191.81)\ \text{kJ/kg} \\ &= 58{,}930\ \text{kW} \end{aligned}$$

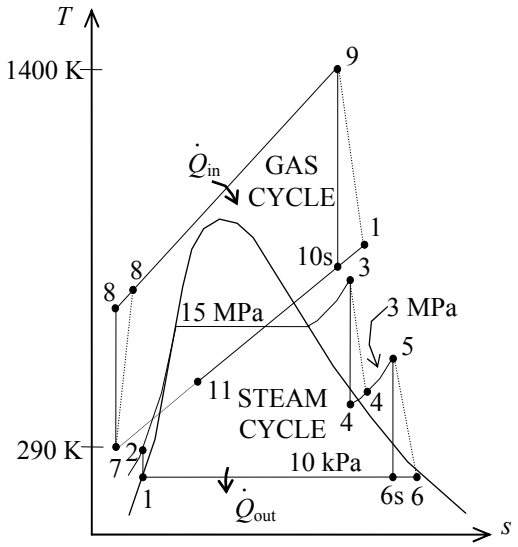
$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{58{,}930\ \text{kW}}{144{,}200\ \text{kW}} = 0.5913 = \mathbf{59.1\%}$$

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10-112 A combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal reheat Rankine cycle. The mass flow rate of air in the gas-turbine cycle, the rate of total heat input, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) The analysis of gas cycle yields (Table A-17)



$$T_7 = 290\ \text{K} \longrightarrow \begin{matrix} h_7 = 290.16\ \text{kJ/kg} \\ P_{r_7} = 1.2311 \end{matrix}$$

$$P_{r_{8s}} = \frac{P_{8s}}{P_7} P_{r_7} = (8)(1.2311) = 9.849 \longrightarrow h_{8s} = 526.12\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{8s} - h_7}{h_8 - h_7} \longrightarrow \begin{aligned} h_8 &= h_7 + (h_{8s} - h_7)/\eta_C \\ &= 290.16 + (526.12 - 290.16)/(0.80) \\ &= 585.1\ \text{kJ/kg} \end{aligned}$$

$$T_9 = 1400\ \text{K} \longrightarrow \begin{matrix} h_9 = 1515.42\ \text{kJ/kg} \\ P_{r_9} = 450.5 \end{matrix}$$

$$P_{r_{10s}} = \frac{P_{10s}}{P_9} P_{r_9} = \left(\frac{1}{8}\right)(450.5) = 56.3 \longrightarrow h_{10s} = 860.35\ \text{kJ/kg}$$

$$\eta_T = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \longrightarrow \begin{aligned} h_{10} &= h_9 - \eta_T (h_9 - h_{10s}) \\ &= 1515.42 - (0.85)(1515.42 - 860.35) \\ &= 958.4\ \text{kJ/kg} \end{aligned}$$

$$T_{11} = 520\ \text{K} \longrightarrow h_{11} = 523.63\ \text{kJ/kg}$$

From the steam tables (Tables A-4, A-5, and A-6),

$$\begin{aligned} h_1 &= h_{f\ @\ 10\ \text{kPa}} = 191.81\ \text{kJ/kg} \\ \nu_1 &= \nu_{f\ @\ 10\ \text{kPa}} = 0.00101\ \text{m}^3/\text{kg} \end{aligned}$$

$$\begin{aligned} w_{\text{pI,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.00101\ \text{m}^3/\text{kg})(15{,}000 - 10\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa} \cdot \text{m}^3}\right) \\ &= 15.14\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 15.14 = 206.95\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 15\ \text{MPa} \\ T_3 = 450°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3157.9\ \text{kJ/kg} \\ s_3 = 6.1428\ \text{kJ/kg} \cdot \text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 3\ \text{MPa} \\ s_{4s} = s_3 \end{matrix}\right\} \begin{aligned} x_{4s} &= \frac{s_{4s} - s_f}{s_{fg}} = \frac{6.1434 - 2.6454}{3.5402} = 0.9880 \\ h_{4s} &= h_f + x_{4s} h_{fg} = 1008.3 + (0.9879)(1794.9) = 2781.7\ \text{kJ/kg} \end{aligned}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow \begin{aligned} h_4 &= h_3 - \eta_T (h_3 - h_{4s}) \\ &= 3157.9 - (0.85)(3157.9 - 2781.7) \\ &= 2838.1\ \text{kJ/kg} \end{aligned}$$

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$$\left.\begin{array}{l} P_5 = 3\ \text{MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3457.2\ \text{kJ/kg} \\ s_5 = 7.2359\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10\ \text{kPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{7.2359 - 0.6492}{7.4996} = 0.8783 \\ h_{6s} = h_f + x_{6s}h_{fg} = 191.81 + (0.8782)(2392.1) = 2292.8\ \text{kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow \begin{aligned} h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3457.2 - (0.85)(3457.2 - 2292.8) \\ &= 2467.5\ \text{kJ/kg} \end{aligned}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s(h_3 - h_2) = \dot{m}_{\text{air}}(h_{10} - h_{11})$$

$$\dot{m}_{\text{air}} = \frac{h_3 - h_2}{h_{10} - h_{11}}\dot{m}_s = \frac{3157.9 - 206.95}{958.4 - 523.63}(30\ \text{kg/s}) = \mathbf{203.6\ kg/s}$$

(*b*) $$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{Q}_{\text{air}} + \dot{Q}_{\text{reheat}} = \dot{m}_{\text{air}}(h_9 - h_8) + \dot{m}_{\text{reheat}}(h_5 - h_4) \\ &= (203.6\ \text{kg/s})(1515.42 - 585.1)\ \text{kJ/kg} + (30\ \text{kg/s})(3457.2 - 2838.1)\ \text{kJ/kg} \\ &= \mathbf{207{,}986\ kW} \end{aligned}$$

(*c*) $$\begin{aligned} \dot{Q}_{\text{out}} &= \dot{Q}_{\text{out,air}} + \dot{Q}_{\text{out,steam}} = \dot{m}_{\text{air}}(h_{11} - h_7) + \dot{m}_s(h_6 - h_1) \\ &= (203.6\ \text{kg/s})(523.63 - 290.16)\ \text{kJ/kg} + (30\ \text{kg/s})(2467.5 - 191.81)\ \text{kJ/kg} \\ &= 115{,}805\ \text{kW} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{115{,}805\ \text{kW}}{207{,}986\ \text{kW}} = \mathbf{44.3\%}$$

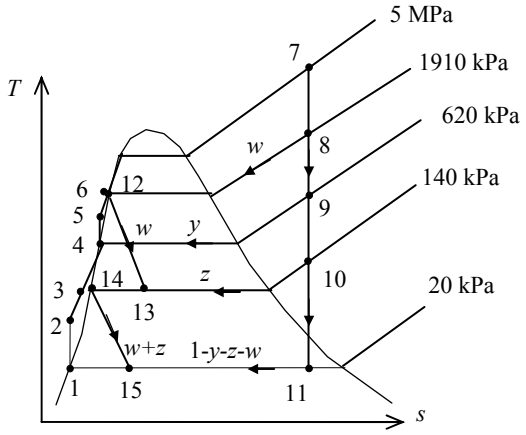
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10-113 A Rankine steam cycle modified with two closed feedwater heaters and one open feed water heater is considered. The *T-s* diagram for the ideal cycle is to be sketched. The fraction of mass extracted for the open feedwater heater *y* and the cooling water flow temperature rise are to be determined. Also, the rate of heat rejected in the condenser and the thermal efficiency of the plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*b*) Using the data from the problem statement, the enthalpies at various states are

$$h_1 = h_{f\,@20\text{ kPa}} = 251\text{ kJ/kg}$$
$$h_{15} = h_3 = h_{14} = h_{f\,@140\text{ kPa}} = 458\text{ kJ/kg}$$
$$h_4 = h_{f\,@620\text{ kPa}} = 676\text{ kJ/kg}$$
$$h_6 = h_{12} = h_{f\,@1910\text{ kPa}} = 898\text{ kJ/kg}$$



An energy balance on the open feedwater heater gives

$$yh_9 + (1-y)h_3 = 1h_4$$

where *z* is the fraction of steam extracted from the low-pressure turbine. Solving for *z*,

$$y = \frac{h_4 - h_3}{h_9 - h_3} = \frac{676 - 458}{3154 - 458} = \mathbf{0.08086}$$

(c) An energy balance on the condenser gives

$$\dot{m}_7\left[(1-w-y-z)h_{11} + (w+z)h_{15} - (1-y)h_1\right] = \dot{m}_w(h_{w2} - h_{w2}) = \dot{m}_w c_{pw}\Delta T_w$$

Solving for the temperature rise of cooling water, and substituting with correct units,

$$\Delta T_w = \frac{\dot{m}_7\left[(1-w-y-z)h_{11} + (w+z)h_{15} - (1-y)h_1\right]}{\dot{m}_w c_{pw}}$$
$$= \frac{(100)\left[(1-0.0830-0.08086-0.0655)(2478) + (0.0830+0.0655)(458) - (1-0.08086)(251)\right]}{(4200)(4.18)}$$
$$= \mathbf{9.95°C}$$

(d) The rate of heat rejected in the condenser is

$$\dot{Q}_{\text{out}} = \dot{m}_w c_{pw}\Delta T_w = (4200\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})(9.95°\text{C}) = \mathbf{174{,}700\text{ kW}}$$

The rate of heat input in the boiler is

$$\dot{Q}_{\text{in}} = \dot{m}(h_7 - h_6) = (100\text{ kg/s})(3900 - 898)\text{ kJ/kg} = 300{,}200\text{ kW}$$

The thermal efficiency is then

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{174{,}700\text{ kW}}{300{,}200\text{ kW}} = 0.418 = \mathbf{41.8\%}$$

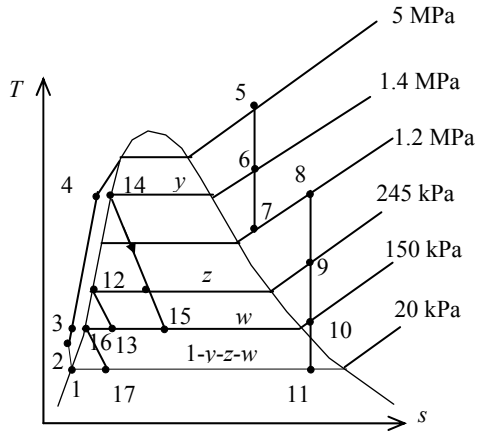
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10-114 A Rankine steam cycle modified for reheat, two closed feedwater heaters and a process heater is considered. The *T*-*s* diagram for the ideal cycle is to be sketched. The fraction of mass, *w*, that is extracted for the closed feedwater heater is to be determined. Also, the mass flow rate through the boiler, the rate of process heat supplied, and the utilization efficiency of this cogeneration plant are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*b*) Using the data from the problem statement, the enthalpies at various states are

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.4\text{ kJ/kg}$$
$$\nu_1 = \nu_{f\,@\,20\text{ kPa}} = 0.00102\ \text{m}^3/\text{kg}$$
$$w_{\text{pI,in}} = \nu_1(P_2 - P_1)$$
$$= (0.00102\ \text{m}^3/\text{kg})(5000 - 20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 5.1\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 251.4 + 5.1 = 256.5\text{ kJ/kg}$$
$$h_{13} = h_{12} = h_{f\,@\,245\text{ kPa}} = 533\text{ kJ/kg}$$
$$h_4 = h_{15} = h_{14} = h_{f\,@\,1400\text{ kPa}} = 830\text{ kJ/kg}$$
$$h_3 = h_{16} = h_{f\,@\,150\text{ kPa}} = 467\text{ kJ/kg}$$



An energy balance on the closed feedwater heater gives

$$1h_2 + wh_{10} + zh_{13} + yh_{15} = 1h_3 + (y + z + w)h_{16}$$

where *w* is the fraction of steam extracted from the low-pressure turbine. Solving for *z*,

$$w = \frac{(h_3 - h_2) + (y + z)h_{16} - zh_{13} - yh_{15}}{h_{10} - h_{16}}$$
$$= \frac{(467 - 256.5) + (0.1160 + 0.15)(467) - (0.15)(533) - (0.1160)(830)}{3023 - 467}$$
$$= \mathbf{0.0620}$$

(c) The work output from the turbines is

$$w_{\text{T,out}} = h_5 - yh_6 - (1 - y)h_7 + (1 - y)h_8 - zh_9 - wh_{10} - (1 - y - z - w)h_{11}$$
$$= 3894 - (0.1160)(3400) - (1 - 0.1160)(3349)$$
$$+ (1 - 0.1160)(3692) - (0.15)(3154) - (0.0620)(3023) - (1 - 0.1160 - 0.15 - 0.0620)(2620)$$
$$= 1381.6\text{ kJ/kg}$$

The net work output from the cycle is

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 1381.6 - 5.1 = 1376.5\text{ kJ/kg}$$

The mass flow rate through the boiler is

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{300{,}000\text{ kW}}{1376.5\text{ kJ/kg}} = \mathbf{217.9\text{ kg/s}}$$

The rate of heat input in the boiler is

$$\dot{Q}_{\text{in}} = \dot{m}(h_5 - h_4) + (1 - y)\dot{m}(h_8 - h_7)$$
$$= (217.9\text{ kg/s})(3894 - 830)\text{ kJ/kg} + (1 - 0.1160)(217.9\text{ kg/s})(3692 - 3349)\text{ kJ/kg}$$
$$= 733{,}700\text{ kW}$$

The rate of process heat and the utilization efficiency of this cogeneration plant are

$$\dot{Q}_{\text{process}} = z\dot{m}(h_9 - h_{12}) = (0.15)(217.9\text{ kg/s})(3154 - 533)\text{ kJ/kg} = \mathbf{85{,}670\text{ kW}}$$

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{(300{,}000 + 85{,}670)\text{ kW}}{733{,}700\text{ kW}} = 0.526 = \mathbf{52.6\%}$$

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10-115 The effect of the condenser pressure on the performance a simple ideal Rankine cycle is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
        x4$=''
        if (x4>1) then x4$='(superheated)'
        if (x4<0) then x4$='(compressed)'
end

P[3] = 10000 [kPa]
T[3] = 550 [C]
"P[4] = 5 [kPa]"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"
x4s$=x4$(x[4])

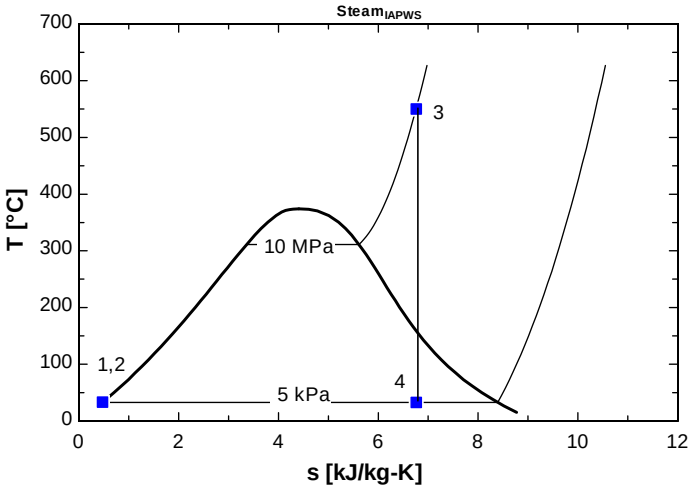
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"

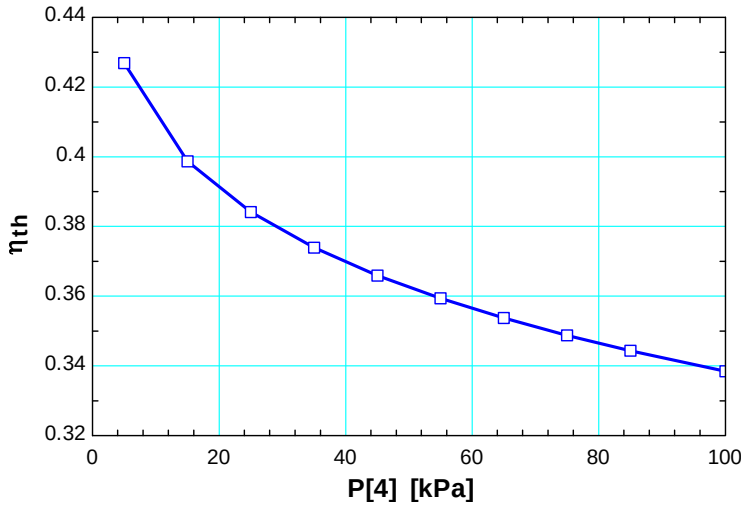
"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"

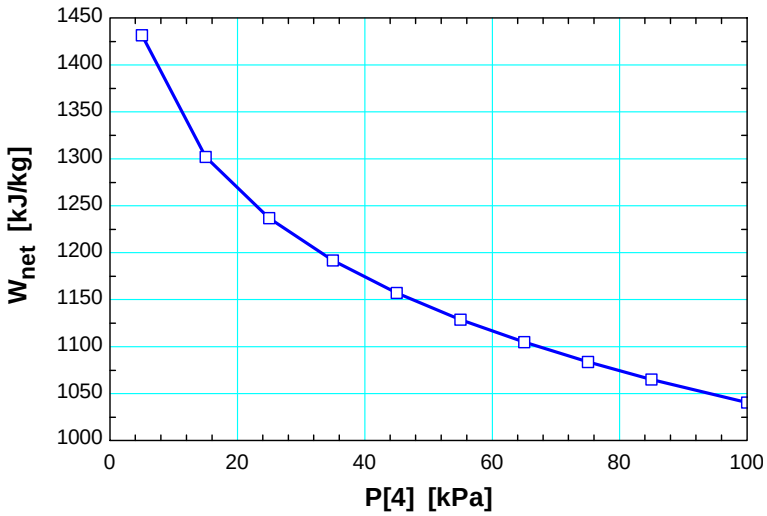
"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

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P_4 [kPa]	η_{th}	W_{net} [kJ/kg]
5	0.4268	1432
15	0.3987	1302
25	0.3841	1237
35	0.3739	1192
45	0.3659	1157
55	0.3594	1129
65	0.3537	1105
75	0.3488	1084
85	0.3443	1065
100	0.3385	1040







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10-116  The effect of superheating the steam on the performance a simple ideal Rankine cycle is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
        x4$=''
        if (x4>1) then x4$='(superheated)'
        if (x4<0) then x4$='(compressed)'
end

P[3] = 3000 [kPa]
{T[3] = 600 [C]}
P[4] = 10 [kPa]
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0 "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"
x4s$=x4$(x[4])

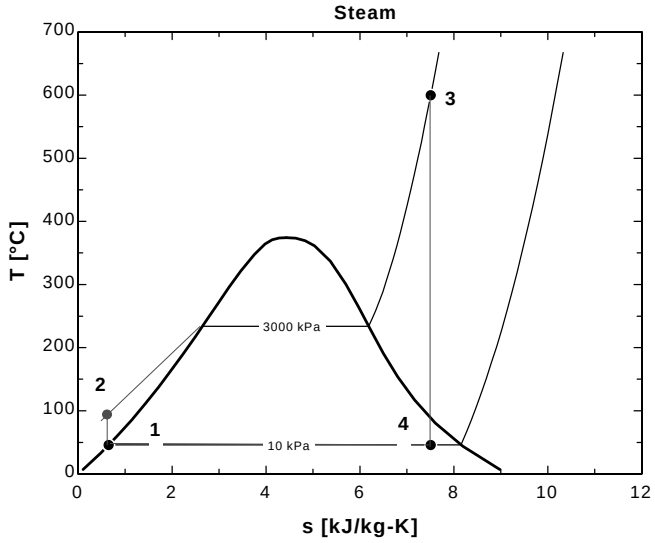
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"

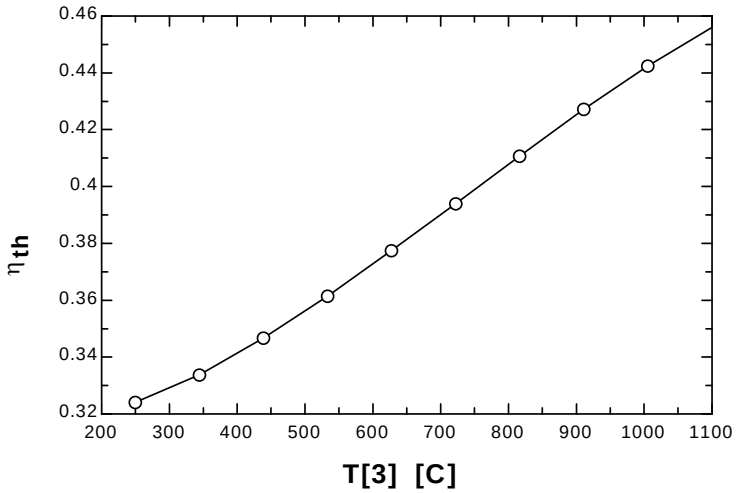
"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"

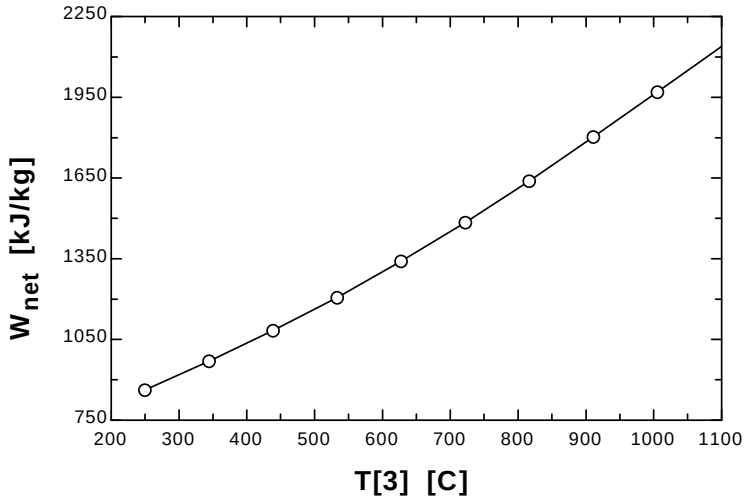
"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

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T_3 [C]	η_{th}	W_{net} [kJ/kg]	x_4
250	0.3241	862.8	0.752
344.4	0.3338	970.6	0.81
438.9	0.3466	1083	0.8536
533.3	0.3614	1206	0.8909
627.8	0.3774	1340	0.9244
722.2	0.3939	1485	0.955
816.7	0.4106	1639	0.9835
911.1	0.4272	1803	100
1006	0.4424	1970	100
1100	0.456	2139	100







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10-117 The effect of number of reheat stages on the performance an ideal Rankine cycle is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end

Procedure Reheat(P[3],T[3],T[5],h[4],NoRHStages,Pratio,Eta_t:Q_in_reheat,W_t_lp,h6)
P3=P[3]
T5=T[5]
h4=h[4]
Q_in_reheat =0
W_t_lp = 0
R_P=(1/Pratio)^(1/(NoRHStages+1))

 imax:=NoRHStages - 1
 i:=0

REPEAT
i:=i+1

P4 = P3*R_P

P5=P4
P6=P5*R_P

Fluid$='Steam_IAPWS'
s5=entropy(Fluid$,T=T5,P=P5)
h5=enthalpy(Fluid$,T=T5,P=P5)
s_s6=s5
hs6=enthalpy(Fluid$,s=s_s6,P=P6)
Ts6=temperature(Fluid$,s=s_s6,P=P6)
vs6=volume(Fluid$,s=s_s6,P=P6)
"Eta_t=(h5-h6)/(h5-hs6)""Definition of turbine efficiency"
h6=h5-Eta_t*(h5-hs6)
W_t_lp=W_t_lp+h5-h6"SSSF First Law for the low pressure turbine"
x6=QUALITY(Fluid$,h=h6,P=P6)
Q_in_reheat =Q_in_reheat + (h5 - h4)
P3=P4

UNTIL (i>imax)

END

"NoRHStages = 2"
P[6] = 10"kPa"
P[3] = 15000"kPa"
P_extract = P[6]  "Select a lower limit on the reheat pressure"
T[3] = 500"C"
T[5] = 500"C"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"
Pratio = P[3]/P_extract
P[4] = P[3]*(1/Pratio)^(1/(NoRHStages+1))"kPa"

Fluid$='Steam_IAPWS'
```

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```
"Pump analysis"
P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"

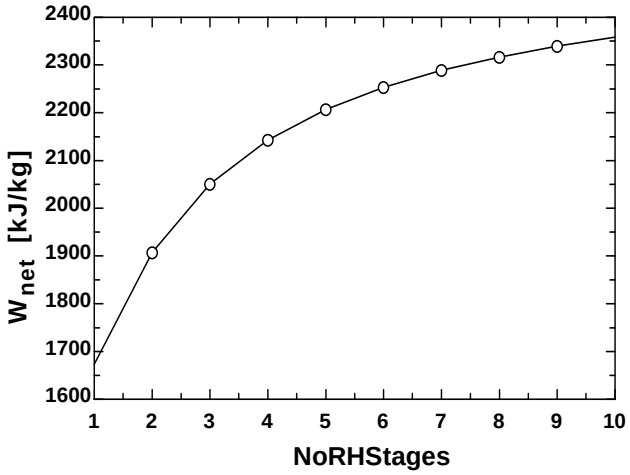
"Low Pressure Turbine analysis"
Call Reheat(P[3],T[3],T[5],h[4],NoRHStages,Pratio,Eta_t:Q_in_reheat,W_t_lp,h6)
h[6]=h6

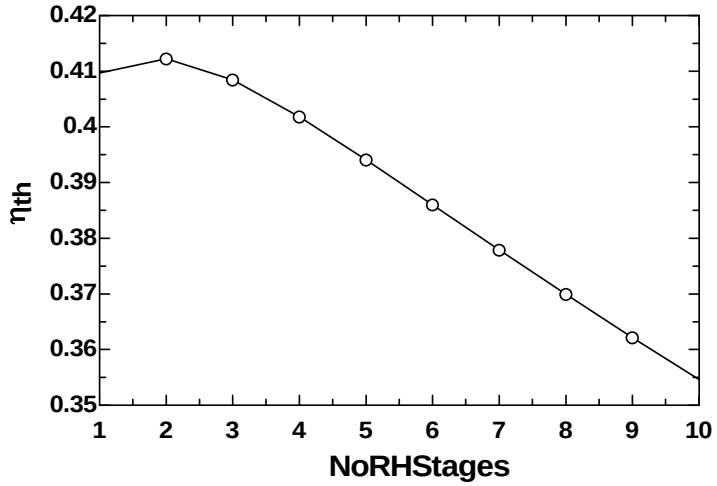
{P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
W_t_lp_total = NoRHStages*W_t_lp
Q_in_reheat = NoRHStages*(h[5] - h[4])}

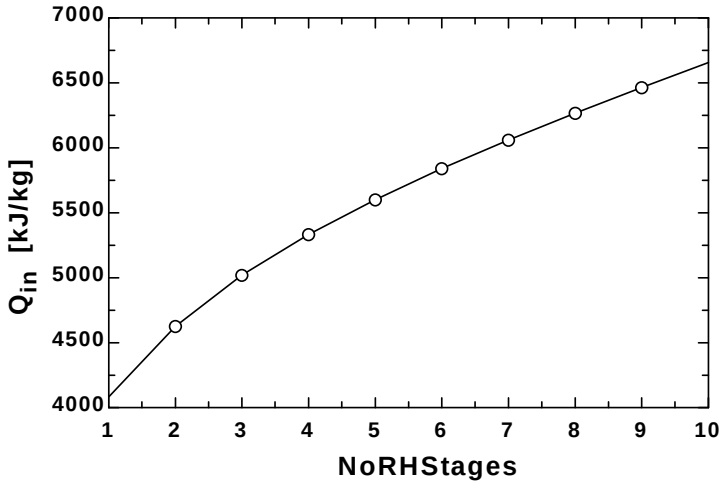
"Boiler analysis"
Q_in_boiler + h[2]=h[3]"SSSF First Law for the Boiler"
Q_in = Q_in_boiler+Q_in_reheat
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])
"Cycle Statistics"
W_net=W_t_hp+W_t_lp - W_p
Eta_th=W_net/Q_in
```

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η_{th}	NoRH Stages	Q_{in} [kJ/kg]	W_{net} [kJ/kg]
0.4097	1	4085	1674
0.4122	2	4628	1908
0.4085	3	5020	2051
0.4018	4	5333	2143
0.3941	5	5600	2207
0.386	6	5838	2253
0.3779	7	6058	2289
0.3699	8	6264	2317
0.3621	9	6461	2340
0.3546	10	6651	2358







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10-118 The effect of number of regeneration stages on the performance an ideal regenerative Rankine cycle with one open feedwater heater is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
Procedure Reheat(NoFwh,T[5],P[5],P_cond,Eta_turb,Eta_pump:q_in,w_net)

Fluid$='Steam_IAPWS'
Tcond = temperature(Fluid$,P=P_cond,x=0)
Tboiler = temperature(Fluid$,P=P[5],x=0)
P[7] = P_cond
s[5]=entropy(Fluid$, T=T[5], P=P[5])
h[5]=enthalpy(Fluid$, T=T[5], P=P[5])
h[1]=enthalpy(Fluid$, P=P[7],x=0)

P4[1] = P[5]  "NOTICE THIS IS P4[i] WITH i = 1"

DELTAT_cond_boiler = Tboiler - Tcond

If NoFWH = 0 Then

    "the following are h7, h2, w_net, and q_in for zero feedwater heaters, NoFWH = 0"
    h7=enthalpy(Fluid$, s=s[5],P=P[7])
    h2=h[1]+volume(Fluid$, P=P[7],x=0)*(P[5] - P[7])/Eta_pump
    w_net = Eta_turb*(h[5]-h7)-(h2-h[1])
    q_in = h[5] - h2

else

i=0
REPEAT
i=i+1
"The following maintains the same temperature difference between any two regeneration stages."
T_FWH[i] = (NoFWH +1 - i)*DELTAT_cond_boiler/(NoFWH + 1)+Tcond"[C]"
P_extract[i] = pressure(Fluid$,T=T_FWH[i],x=0)"[kPa]"
P3[i]=P_extract[i]
P6[i]=P_extract[i]
If i > 1 then P4[i] = P6[i - 1]

UNTIL i=NoFWH

P4[NoFWH+1]=P6[NoFWH]
h4[NoFWH+1]=h[1]+volume(Fluid$, P=P[7],x=0)*(P4[NoFWH+1] - P[7])/Eta_pump

i=0
REPEAT
i=i+1

"Boiler condensate pump  or the Pumps 2 between feedwater heaters analysis"
h3[i]=enthalpy(Fluid$,P=P3[i],x=0)
v3[i]=volume(Fluid$,P=P3[i],x=0)
w_pump2_s=v3[i]*(P4[i]-P3[i])"SSSF isentropic pump work assuming constant specific volume"
w_pump2[i]=w_pump2_s/Eta_pump "Definition of pump efficiency"
h4[i]= w_pump2[i] +h3[i]  "Steady-flow conservation of energy"
s4[i]=entropy(Fluid$,P=P4[i],h=h4[i])
T4[i]=temperature(Fluid$,P=P4[i],h=h4[i])
Until i = NoFWH
i=0
REPEAT
```

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```
i=i+1
"Open Feedwater Heater analysis:"
{h2[i] = h6[i]}
s5[i] = s[5]
ss6[i]=s5[i]
hs6[i]=enthalpy(Fluid$,s=ss6[i],P=P6[i])
Ts6[i]=temperature(Fluid$,s=ss6[i],P=P6[i])
h6[i]=h[5]-Eta_turb*(h[5]-hs6[i])"Definition of turbine efficiency for high pressure stages"
If i=1 then y[1]=(h3[1] - h4[2])/(h6[1]  - h4[2])  "Steady-flow conservation of energy for the FWH"
If i > 1 then
   js = i -1
   j = 0
   sumyj = 0
   REPEAT
   j = j+1
   sumyj = sumyj + y[ j ]
   UNTIL j = js
y[i] =(1- sumyj)*(h3[i] - h4[i+1])/(h6[i]  - h4[i+1])

ENDIF
T3[i]=temperature(Fluid$,P=P3[i],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s3[i]=entropy(Fluid$,P=P3[i],x=0)

"Turbine analysis"
T6[i]=temperature(Fluid$,P=P6[i],h=h6[i])
s6[i]=entropy(Fluid$,P=P6[i],h=h6[i])
yh6[i] = y[i]*h6[i]
UNTIL i=NoFWH
ss[7]=s6[i]
hs[7]=enthalpy(Fluid$,s=ss[7],P=P[7])
Ts[7]=temperature(Fluid$,s=ss[7],P=P[7])
h[7]=h6[i]-Eta_turb*(h6[i]-hs[7])"Definition of turbine efficiency for low pressure stages"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])

sumyi = 0
sumyh6i = 0
wp2i =  W_pump2[1]
i=0
REPEAT
i=i+1
sumyi = sumyi + y[i]
sumyh6i = sumyh6i + yh6[i]
If NoFWH > 1 then wp2i = wp2i + (1- sumyi)*W_pump2[i]
UNTIL i = NoFWH

"Condenser Pump---Pump_1 Analysis:"
P[2] = P6 [ NoFWH]
P[1] = P_cond
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[2]=w_pump1+ h[1]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Boiler analysis"
q_in = h[5] - h4[1]"SSSF conservation of energy for the Boiler"
w_turb = h[5] - sumyh6i - (1- sumyi)*h[7]  "SSSF conservation of energy for turbine"
```

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"Condenser analysis"
q_out=(1- sumyi)*(h[7] - h[1])"SSSF First Law for the Condenser"

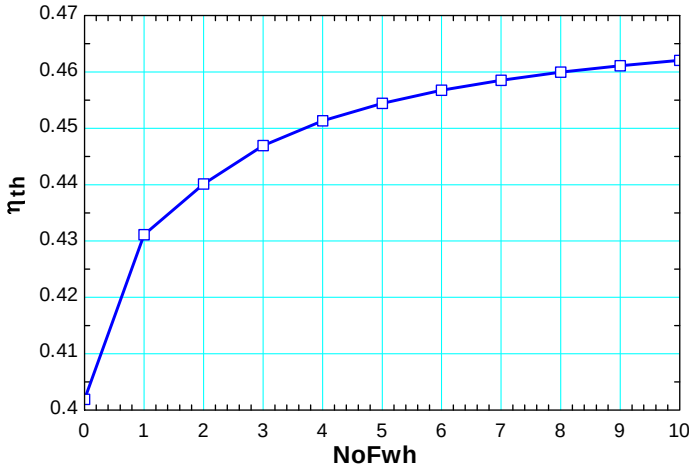
"Cycle Statistics"
w_net=w_turb - ((1- sumyi)*w_pump1+ wp2i)

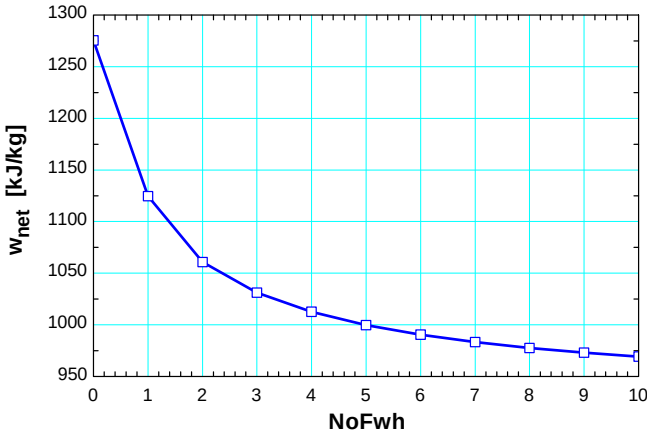
endif
END

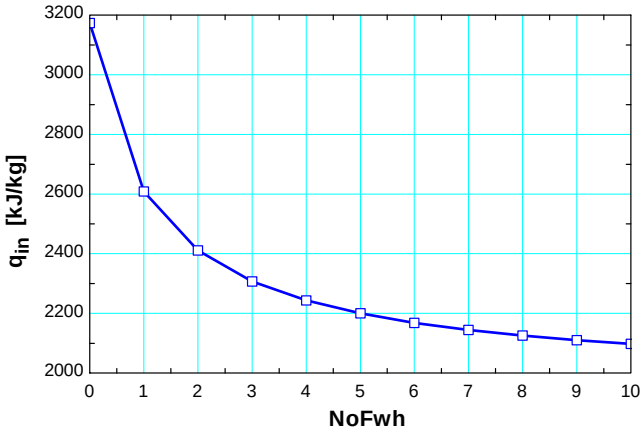
"Input Data"
NoFWH = 2
P[5] = 10000 [kPa]
T[5] = 500 [C]
P_cond=10 [kPa]
Eta_turb= 1.0 "Turbine isentropic efficiency"
Eta_pump = 1.0 "Pump isentropic efficiency"
P[1] = P_cond
P[4] = P[5]

"Condenser exit pump or Pump 1 analysis"
Call Reheat(NoFwh,T[5],P[5],P_cond,Eta_turb,Eta_pump:q_in,w_net)
Eta_th=w_net/q_in

No FWH	η_{th}	w_{net} [kJ/kg]	q_{in} [kJ/kg]
0	0.4019	1275	3173
1	0.4311	1125	2609
2	0.4401	1061	2411
3	0.4469	1031	2307
4	0.4513	1013	2243
5	0.4544	1000	2200
6	0.4567	990.5	2169
7	0.4585	983.3	2145
8	0.4599	977.7	2126
9	0.4611	973.1	2111
10	0.462	969.4	2098







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10-119 It is to be demonstrated that the thermal efficiency of a combined gas-steam power plant η_{cc} can be expressed as $\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s$ where $\eta_g = W_g / Q_{in}$ and $\eta_s = W_s / Q_{g,out}$ are the thermal efficiencies of the gas and steam cycles, respectively, and the efficiency of a combined cycle is to be obtained.

Analysis The thermal efficiencies of gas, steam, and combined cycles can be expressed as

$$\eta_{cc} = \frac{W_{total}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$
$$\eta_g = \frac{W_g}{Q_{in}} = 1 - \frac{Q_{g,out}}{Q_{in}}$$
$$\eta_s = \frac{W_s}{Q_{g,out}} = 1 - \frac{Q_{out}}{Q_{g,out}}$$

where Q_{in} is the heat supplied to the gas cycle, where Q_{out} is the heat rejected by the steam cycle, and where $Q_{g,out}$ is the heat rejected from the gas cycle and supplied to the steam cycle.

Using the relations above, the expression $\eta_g + \eta_s - \eta_g\eta_s$ can be expressed as

$$\begin{aligned}\eta_g + \eta_s - \eta_g\eta_s &= \left(1 - \frac{Q_{g,out}}{Q_{in}}\right) + \left(1 - \frac{Q_{out}}{Q_{g,out}}\right) - \left(1 - \frac{Q_{g,out}}{Q_{in}}\right)\left(1 - \frac{Q_{out}}{Q_{g,out}}\right) \\ &= 1 - \frac{Q_{g,out}}{Q_{in}} + 1 - \frac{Q_{out}}{Q_{g,out}} - 1 + \frac{Q_{g,out}}{Q_{in}} + \frac{Q_{out}}{Q_{g,out}} - \frac{Q_{out}}{Q_{in}} \\ &= 1 - \frac{Q_{out}}{Q_{in}} \\ &= \eta_{cc}\end{aligned}$$

Therefore, the proof is complete. Using the relation above, the thermal efficiency of the given combined cycle is determined to be

$$\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s = 0.4 + 0.30 - 0.40 \times 0.30 = \mathbf{0.58}$$

10-120 The thermal efficiency of a combined gas-steam power plant η_{cc} can be expressed in terms of the thermal efficiencies of the gas and the steam turbine cycles as $\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s$. It is to be shown that the value of η_{cc} is greater than either of η_g or η_s.

Analysis By factoring out terms, the relation $\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s$ can be expressed as

$$\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s = \eta_g + \underbrace{\eta_s(1 - \eta_g)}_{\substack{\text{Positive since} \\ \eta_g < 1}} > \eta_g$$

or
$$\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s = \eta_s + \underbrace{\eta_g(1 - \eta_s)}_{\substack{\text{Positive since} \\ \eta_s < 1}} > \eta_s$$

Thus we conclude that the combined cycle is more efficient than either of the gas turbine or steam turbine cycles alone.

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10-121 It is to be shown that the exergy destruction associated with a simple ideal Rankine cycle can be expressed as $x_{\text{destroyed}} = q_{in}\left(\eta_{\text{th,Carnot}} - \eta_{th}\right)$, where η_{th} is efficiency of the Rankine cycle and $\eta_{\text{th, Carnot}}$ is the efficiency of the Carnot cycle operating between the same temperature limits.

Analysis The exergy destruction associated with a cycle is given on a unit mass basis as

$$x_{\text{destroyed}} = T_0 \sum \frac{q_R}{T_R}$$

where the direction of q_{in} is determined with respect to the reservoir (positive if to the reservoir and negative if from the reservoir). For a cycle that involves heat transfer only with a source at T_{H} and a sink at T_0, the irreversibility becomes

$$\begin{aligned} x_{\text{destroyed}} &= T_0\left(\frac{q_{\text{out}}}{T_0} - \frac{q_{\text{in}}}{T_H}\right) = q_{\text{out}} - \frac{T_0}{T_H} q_{\text{in}} = q_{\text{in}}\left(\frac{q_{\text{out}}}{q_{\text{in}}} - \frac{T_0}{T_H}\right) \\ &= q_{\text{in}}\left[\left(1-\eta_{th}\right) - \left(1-\eta_{th,C}\right)\right] = q_{\text{in}}\left(\eta_{th,C} - \eta_{th}\right) \end{aligned}$$

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Fundamentals of Engineering (FE) Exam Problems

10-122 Consider a simple ideal Rankine cycle. If the condenser pressure is lowered while keeping turbine inlet state the same, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the cycle efficiency will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the pump work input will decrease.

Answer (b) the amount of heat rejected will decrease.

10-123 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. If the steam is superheated to a higher temperature, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the cycle efficiency will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the amount of heat input will decrease.

Answer (d) the moisture content at turbine exit will decrease.

10-124 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures . If the cycle is modified with reheating, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the pump work input will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the amount of heat input will decrease.

Answer (d) the moisture content at turbine exit will decrease.

10-125 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures . If the cycle is modified with regeneration that involves one open feed water heater, (select the correct statement per unit mass of steam flowing through the boiler)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will increase.
(c) the cycle thermal efficiency will decrease.
(d) the quality of steam at turbine exit will decrease.
(e) the amount of heat input will increase.

Answer (a) the turbine work output will decrease.

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10-126 Consider a steady-flow Carnot cycle with water as the working fluid executed under the saturation dome between the pressure limits of 3 MPa and 10 kPa. Water changes from saturated liquid to saturated vapor during the heat addition process. The net work output of this cycle is

(a) 666 kJ/kg (b) 888 kJ/kg (c) 1040 kJ/kg (d) 1130 kJ/kg (e) 1440 kJ/kg

Answer (a) 666 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=33000 "kPa"
P2=10 "kPa"
h_fg=ENTHALPY(Steam_IAPWS,x=1,P=P1)-ENTHALPY(Steam_IAPWS,x=0,P=P1)
T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1)+273
T2=TEMPERATURE(Steam_IAPWS,x=0,P=P2)+273
q_in=h_fg
Eta_Carnot=1-T2/T1
w_net=Eta_Carnot*q_in

"Some Wrong Solutions with Common Mistakes:"
W1_work = Eta1*q_in; Eta1=T2/T1 "Taking Carnot efficiency to be T2/T1"
W2_work = Eta2*q_in; Eta2=1-(T2-273)/(T1-273)  "Using C instead of K"
W3_work = Eta_Carnot*ENTHALPY(Steam_IAPWS,x=1,P=P1)  "Using h_g instead of h_fg"
W4_work = Eta_Carnot*q2; q2=ENTHALPY(Steam_IAPWS,x=1,P=P2)-ENTHALPY(Steam_IAPWS,x=0,P=P2)
"Using h_fg at P2"
```

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10-127 A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 3 MPa, with a turbine inlet temperature of 600°C. Disregarding the pump work, the cycle efficiency is

(a) 24% (b) 37% (c) 52% (d) 63% (e) 71%

Answer (b) 37%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=3000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1) "kJ/kg"
h2=h1+w_pump
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
q_in=h3-h2
q_out=h4-h1
Eta_th=1-q_out/q_in

"Some Wrong Solutions with Common Mistakes:"
W1_Eff = q_out/q_in "Using wrong relation"
W2_Eff = 1-(h44-h1)/(h3-h2); h44 = ENTHALPY(Steam_IAPWS,x=1,P=P4) "Using h_g for h4"
W3_Eff = 1-(T1+273)/(T3+273); T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_Eff = (h3-h4)/q_in  "Disregarding pump work"
```

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10-128 A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 5 MPa, with a turbine inlet temperature of 600°C. The mass fraction of steam that condenses at the turbine exit is

(a) 6% (b) 9% (c) 12% (d) 15% (e) 18%

Answer (c) 12%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=5000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
x4=QUALITY(Steam_IAPWS,s=s4,P=P4)
moisture=1-x4

"Some Wrong Solutions with Common Mistakes:"
W1_moisture = x4 "Taking quality as moisture"
W2_moisture = 0 "Assuming superheated vapor"
```

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10-129 A steam power plant operates on the simple ideal Rankine cycle between the pressure limits of 5 kPa and 10 MPa, with a turbine inlet temperature of 600°C. The rate of heat transfer in the boiler is 300 kJ/s. Disregarding the pump work, the power output of this plant is

(a) 93 kW (b) 118 kW (c) 190 kW (d) 216 kW (e) 300 kW

Answer (b) 118 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=5000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
Q_rate=300 "kJ/s"
m=Q_rate/q_in
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
h2=h1  "pump work is neglected"
"v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1)
h2=h1+w_pump"
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
q_in=h3-h2
W_turb=m*(h3-h4)

"Some Wrong Solutions with Common Mistakes:"
W1_power = Q_rate "Assuming all heat is converted to power"
W3_power = Q_rate*Carnot; Carnot = 1-(T1+273)/(T3+273); T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1)
"Using Carnot efficiency"
W4_power = m*(h3-h44); h44 = ENTHALPY(Steam_IAPWS,x=1,P=P4) "Taking h4=h_g"
```

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10-130 Consider a combined gas-steam power plant. Water for the steam cycle is heated in a well-insulated heat exchanger by the exhaust gases that enter at 800 K at a rate of 60 kg/s and leave at 400 K. Water enters the heat exchanger at 200°C and 8 MPa and leaves at 350°C and 8 MPa. If the exhaust gases are treated as air with constant specific heats at room temperature, the mass flow rate of water through the heat exchanger becomes

(a) 11 kg/s (b) 24 kg/s (c) 46 kg/s (d) 53 kg/s (e) 60 kg/s

Answer (a) 11 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_gas=60 "kg/s"
Cp=1.005 "kJ/kg.K"
T3=800 "K"
T4=400 "K"
Q_gas=m_gas*Cp*(T3-T4)
P1=8000 "kPa"
T1=200 "C"
P2=8000 "kPa"
T2=350 "C"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
Q_steam=m_steam*(h2-h1)
Q_gas=Q_steam

"Some Wrong Solutions with Common Mistakes:"
m_gas*Cp*(T3 -T4)=W1_msteam*4.18*(T2-T1) "Assuming no evaporation of liquid water"
m_gas*Cv*(T3 -T4)=W2_msteam*(h2-h1); Cv=0.718 "Using Cv for air instead of Cp"
W3_msteam = m_gas "Taking the mass flow rates of two fluids to be equal"
m_gas*Cp*(T3 -T4)=W4_msteam*(h2-h11); h11=ENTHALPY(Steam_IAPWS,x=0,P=P1) "Taking h1=hf@P1"
```

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10-131 An ideal reheat Rankine cycle operates between the pressure limits of 10 kPa and 8 MPa, with reheat occurring at 4 MPa. The temperature of steam at the inlets of both turbines is 500°C, and the enthalpy of steam is 3185 kJ/kg at the exit of the high-pressure turbine, and 2247 kJ/kg at the exit of the low-pressure turbine. Disregarding the pump work, the cycle efficiency is

(a) 29% (b) 32% (c) 36% (d) 41% (e) 49%

Answer (d) 41%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=8000 "kPa"
P3=P2
P4=4000 "kPa"
P5=P4
P6=P1
T3=500 "C"
T5=500 "C"
s4=s3
s6=s5
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
h2=h1
h44=3185 "kJ/kg - for checking given data"
h66=2247 "kJ/kg - for checking given data"
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
h5=ENTHALPY(Steam_IAPWS,T=T5,P=P5)
s5=ENTROPY(Steam_IAPWS,T=T5,P=P5)
h6=ENTHALPY(Steam_IAPWS,s=s6,P=P6)
q_in=(h3-h2)+(h5-h4)
q_out=h6-h1
Eta_th=1-q_out/q_in

"Some Wrong Solutions with Common Mistakes:"
W1_Eff = q_out/q_in "Using wrong relation"
W2_Eff = 1-q_out/(h3-h2) "Disregarding heat input during reheat"
W3_Eff = 1-(T1+273)/(T3+273); T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_Eff = 1-q_out/(h5-h2)  "Using wrong relation for q_in"
```

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10-132 Pressurized feedwater in a steam power plant is to be heated in an ideal open feedwater heater that operates at a pressure of 2 MPa with steam extracted from the turbine. If the enthalpy of feedwater is 252 kJ/kg and the enthalpy of extracted steam is 2810 kJ/kg, the mass fraction of steam extracted from the turbine is

(a) 10% (b) 14% (c) 26% (d) 36% (e) 50%

Answer (c) 26%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_feed=252 "kJ/kg"
h_extracted=2810 "kJ/kg"
P3=2000 "kPa"
h3=ENTHALPY(Steam_IAPWS,x=0,P=P3)
"Energy balance on the FWH"
h3=x_ext*h_extracted+(1-x_ext)*h_feed

"Some Wrong Solutions with Common Mistakes:"
W1_ext = h_feed/h_extracted "Using wrong relation"
W2_ext = h3/(h_extracted-h_feed) "Using wrong relation"
W3_ext = h_feed/(h_extracted-h_feed) "Using wrong relation"
```

10-133 Consider a steam power plant that operates on the regenerative Rankine cycle with one open feedwater heater. The enthalpy of the steam is 3374 kJ/kg at the turbine inlet, 2797 kJ/kg at the location of bleeding, and 2346 kJ/kg at the turbine exit. The net power output of the plant is 120 MW, and the fraction of steam bled off the turbine for regeneration is 0.172. If the pump work is negligible, the mass flow rate of steam at the turbine inlet is

(a) 117 kg/s (b) 126 kg/s (c) 219 kg/s (d) 288 kg/s (e) 679 kg/s

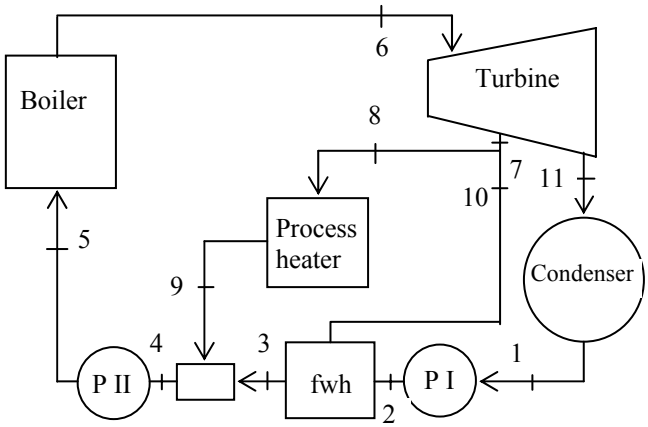
Answer (b) 126 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_in=3374 "kJ/kg"
h_out=2346 "kJ/kg"
h_extracted=2797 "kJ/kg"
Wnet_out=120000 "kW"
x_bleed=0.172
w_turb=(h_in-h_extracted)+(1-x_bleed)*(h_extracted-h_out)
m=Wnet_out/w_turb
"Some Wrong Solutions with Common Mistakes:"
W1_mass = Wnet_out/(h_in-h_out) "Disregarding extraction of steam"
W2_mass = Wnet_out/(x_bleed*(h_in-h_out)) "Assuming steam is extracted at trubine inlet"
W3_mass = Wnet_out/(h_in-h_out-x_bleed*h_extracted) "Using wrong relation"
```

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10-134 Consider a cogeneration power plant modified with regeneration. Steam enters the turbine at 6 MPa and 450°C at a rate of 20 kg/s and expands to a pressure of 0.4 MPa. At this pressure, 60% of the steam is extracted from the turbine, and the remainder expands to a pressure of 10 kPa. Part of the extracted steam is used to heat feedwater in an open feedwater heater. The rest of the extracted steam is used for process heating and leaves the process heater as a saturated liquid at 0.4 MPa. It is subsequently mixed with the feedwater leaving the feedwater heater, and the mixture is pumped to the boiler pressure. The steam in the condenser is cooled and condensed by the cooling water from a nearby river, which enters the adiabatic condenser at a rate of 463 kg/s.



$h_1 = 191.81$
$h_2 = 192.20$
$h_3 = h_4 = h_9 = 604.66$
$h_5 = 610.73$
$h_6 = 3302.9$
$h_7 = h_8 = h_{10} = 2665.6$
$h_{11} = 2128.8$

1. The total power output of the turbine is

(*a*) 17.0 MW (*b*) 8.4 MW (*c*) 12.2 MW (*d*) 20.0 MW (*e*) 3.4 MW

Answer (a) 17.0 MW

2. The temperature rise of the cooling water from the river in the condenser is

(*a*) 8.0°C (*b*) 5.2°C (*c*) 9.6°C (*d*) 12.9°C (*e*) 16.2°C

Answer (*a*) 8.0°C

3. The mass flow rate of steam through the process heater is

(*a*) 1.6 kg/s (*b*) 3.8 kg/s (*c*) 5.2 kg/s (*d*) 7.6 kg/s (*e*) 10.4 kg/s

Answer (*e*) 10.4 kg/s

4. The rate of heat supply from the process heater per unit mass of steam passing through it is

(*a*) 246 kJ/kg (*b*) 893 kJ/kg (*c*) 1344 kJ/kg (*d*) 1891 kJ/kg (*e*) 2060 kJ/kg

Answer (*e*) 2060 kJ/kg

5. The rate of heat transfer to the steam in the boiler is

(*a*) 26.0 MJ/s (*b*) 53.8 MJ/s (*c*) 39.5 MJ/s (*d*) 62.8 MJ/s (*e*) 125.4 MJ/s

Answer (*b*) 53.8 MJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

Note: The solution given below also evaluates all enthalpies given on the figure.

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```
P1=10 "kPa"
P11=P1
P2=400 "kPa"
P3=P2; P4=P2; P7=P2; P8=P2; P9=P2; P10=P2
P5=6000 "kPa"
P6=P5
T6=450 "C"
m_total=20 "kg/s"
m7=0.6*m_total
m_cond=0.4*m_total
C=4.18 "kJ/kg.K"
m_cooling=463 "kg/s"
s7=s6
s11=s6
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1)
h2=h1+w_pump
h3=ENTHALPY(Steam_IAPWS,x=0,P=P3)
h4=h3; h9=h3
v4=VOLUME(Steam_IAPWS,x=0,P=P4)
w_pump2=v4*(P5-P4)
h5=h4+w_pump2
h6=ENTHALPY(Steam_IAPWS,T=T6,P=P6)
s6=ENTROPY(Steam_IAPWS,T=T6,P=P6)
h7=ENTHALPY(Steam_IAPWS,s=s7,P=P7)
h8=h7; h10=h7
h11=ENTHALPY(Steam_IAPWS,s=s11,P=P11)
W_turb=m_total*(h6-h7)+m_cond*(h7-h11)
m_cooling*C*T_rise=m_cond*(h11-h1)
m_cond*h2+m_feed*h10=(m_cond+m_feed)*h3
m_process=m7-m_feed
q_process=h8-h9
Q_in=m_total*(h6-h5)
```

10-135 ··· 10-142 Design and Essay Problems



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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 11
REFRIGERATION CYCLES

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The Reversed Carnot Cycle

11-1C The reversed Carnot cycle serves as a standard against which actual refrigeration cycles can be compared. Also, the COP of the reversed Carnot cycle provides the upper limit for the COP of a refrigeration cycle operating between the specified temperature limits.

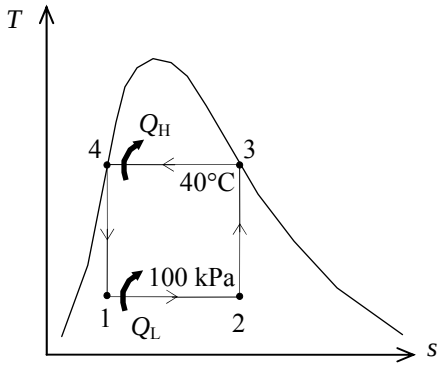
11-2C Because the compression process involves the compression of a liquid-vapor mixture which requires a compressor that will handle two phases, and the expansion process involves the expansion of high-moisture content refrigerant.

11-3 A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The coefficient of performance, the amount of heat absorbed from the refrigerated space, and the net work input are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) Noting that $T_H = 40°C = 313$ K and $T_L = T_{\text{sat @ 100 kPa}} = -26.37°C = 246.6$ K, the COP of this Carnot refrigerator is determined from

$$\text{COP}_{\text{R,C}} = \frac{1}{T_H / T_L - 1} = \frac{1}{(313\text{ K})/(246.6\text{ K}) - 1} = \mathbf{3.72}$$



(*b*) From the refrigerant tables (Table A-11),

$$h_3 = h_{g@40°C} = 271.27 \text{ kJ/kg}$$
$$h_4 = h_{f@40°C} = 108.26 \text{ kJ/kg}$$

Thus,

$$q_H = h_3 - h_4 = 271.27 - 108.26 = 163.0 \text{ kJ/kg}$$

and

$$\frac{q_H}{q_L} = \frac{T_H}{T_L} \longrightarrow q_L = \frac{T_L}{T_H} q_H = \left(\frac{246.6\text{ K}}{313\text{ K}}\right)(163.0\text{ kJ/kg}) = \mathbf{128.4\ kJ/kg}$$

(*c*) The net work input is determined from

$$w_{\text{net}} = q_H - q_L = 163.0 - 128.4 = \mathbf{34.6\ kJ/kg}$$

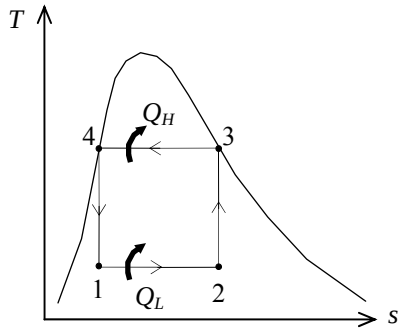
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11-4E A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The coefficient of performance, the quality at the beginning of the heat-absorption process, and the net work input are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) Noting that $T_H = T_{\text{sat @ 90 psia}} = 72.78°\text{F} = 532.8$ R and $T_L = T_{\text{sat @ 30 psia}} = 15.37°\text{F} = 475.4$ R.

$$\text{COP}_{\text{R,C}} = \frac{1}{T_H / T_L - 1} = \frac{1}{(532.8\ \text{R})/(475.4\ \text{R}) - 1} = \mathbf{8.28}$$



(*b*) Process 4-1 is isentropic, and thus

$$s_1 = s_4 = (s_f + x_4 s_{fg})_{@\ 90\ \text{psia}} = 0.07481 + (0.05)(0.14525)$$
$$= 0.08207\ \text{Btu/lbm} \cdot \text{R}$$

$$x_1 = \left(\frac{s_1 - s_f}{s_{fg}} \right)_{@\ 30\ \text{psia}} = \frac{0.08207 - 0.03793}{0.18589} = \mathbf{0.2374}$$

(*c*) Remembering that on a *T*-*s* diagram the area enclosed represents the net work, and $s_3 = s_{g\ @\ 90\ \text{psia}} = 0.22006$ Btu/lbm·R,

$$w_{\text{net,in}} = (T_H - T_L)(s_3 - s_4) = (72.78 - 15.37)(0.22006 - 0.08207)\ \text{Btu/lbm} \cdot \text{R} = \mathbf{7.92\ Btu/lbm}$$

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Ideal and Actual Vapor-Compression Refrigeration Cycles

11-5C Yes; the throttling process is an internally irreversible process.

11-6C To make the ideal vapor-compression refrigeration cycle more closely approximate the actual cycle.

11-7C No. Assuming the water is maintained at 10°C in the evaporator, the evaporator pressure will be the saturation pressure corresponding to this pressure, which is 1.2 kPa. It is not practical to design refrigeration or air-conditioning devices that involve such extremely low pressures.

11-8C Allowing a temperature difference of 10°C for effective heat transfer, the condensation temperature of the refrigerant should be 25°C. The saturation pressure corresponding to 25°C is 0.67 MPa. Therefore, the recommended pressure would be 0.7 MPa.

11-9C The area enclosed by the cyclic curve on a *T*-*s* diagram represents the net work input for the reversed Carnot cycle, but not so for the ideal vapor-compression refrigeration cycle. This is because the latter cycle involves an irreversible process for which the process path is not known.

11-10C The cycle that involves saturated liquid at 30°C will have a higher COP because, judging from the *T*-*s* diagram, it will require a smaller work input for the same refrigeration capacity.

11-11C The minimum temperature that the refrigerant can be cooled to before throttling is the temperature of the sink (the cooling medium) since heat is transferred from the refrigerant to the cooling medium.

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11-12E A refrigerator operating on the ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The increase in the COP if the throttling process were replaced by an isentropic expansion is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-11E, A-12E, and A-13E),

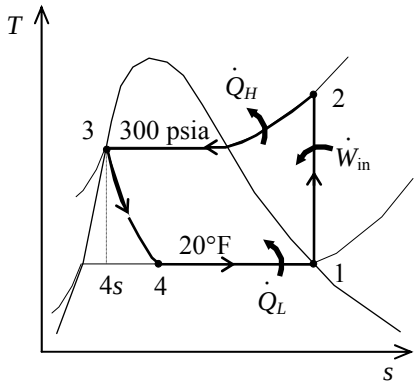
$$\left.\begin{array}{l} T_1 = 20°\text{F} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,20°\text{F}} = 105.98\ \text{Btu/lbm} \\ s_1 = s_{g\,@\,20°\text{F}} = 0.22341\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 300\ \text{psia} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 125.68\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 300\ \text{psia} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,300\,\text{psia}} = 66.339\ \text{Btu/lbm} \\ s_3 = s_{f\,@\,300\,\text{psia}} = 0.12715\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$h_4 \cong h_3 = 66.339\ \text{Btu/lbm} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_4 = 20°\text{F} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} h_{4s} = 59.80\ \text{Btu/lbm} \\ x_{4s} = 0.4723 \end{array} \quad \text{(isentropic expansion)}$$



The COP of the refrigerator for the throttling case is

$$\text{COP}_R = \frac{q_L}{w_{\text{in}}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{105.98 - 66.339}{125.68 - 105.98} = \mathbf{2.012}$$

The COP of the refrigerator for the isentropic expansion case is

$$\text{COP}_R = \frac{q_L}{w_{\text{in}}} = \frac{h_1 - h_{4s}}{h_2 - h_1} = \frac{105.98 - 59.80}{125.68 - 105.98} = \mathbf{2.344}$$

The increase in the COP by isentropic expansion is **16.5%**.

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11-13 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The COP and the power requirement are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

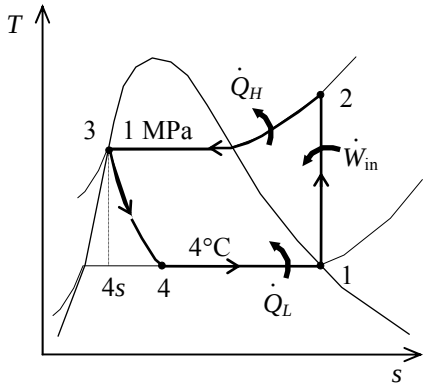
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-11, A-12, and A-13),

$$\left.\begin{array}{l} T_1 = 4°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,4°\text{C}} = 252.77\ \text{kJ/kg} \\ s_1 = s_{g\,@\,4°\text{C}} = 0.92927\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 275.29\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\,@\,1\,\text{MPa}} = 107.32\ \text{kJ/kg}$$

$$h_4 \cong h_3 = 107.32\ \text{kJ/kg} \quad \text{(throttling)}$$



The mass flow rate of the refrigerant is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) \longrightarrow \dot{m} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{400\ \text{kJ/s}}{(252.77 - 107.32)\ \text{kJ/kg}} = 2.750\ \text{kg/s}$$

The power requirement is

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (2.750\ \text{kg/s})(275.29 - 252.77)\ \text{kJ/kg} = \mathbf{61.93\ kW}$$

The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{400\ \text{kW}}{61.93\ \text{kW}} = \mathbf{6.46}$$

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11-14 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

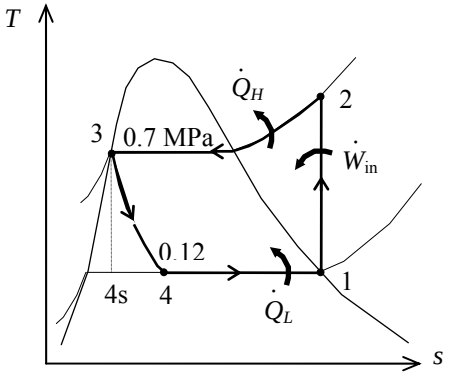
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,120\text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g\,@\,120\text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.50 \text{ kJ/kg} \ (T_2 = 34.95°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.7\text{ MPa}} = 88.82 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 88.82 \text{ kJ/kg} \ (\text{throttling})$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 88.82) \text{ kJ/kg} = \mathbf{7.41\ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(273.50 - 236.97) \text{ kJ/kg} = \mathbf{1.83\ kW}$$

(*b*) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.41 + 1.83 = \mathbf{9.23\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.41 \text{ kW}}{1.83 \text{ kW}} = \mathbf{4.06}$$

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11-15 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

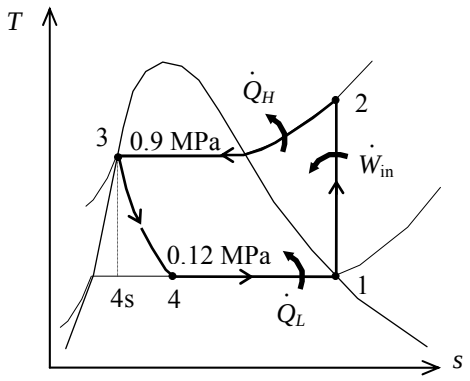
Analysis (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,120\text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g\,@\,120\text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.9 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 278.93 \text{ kJ/kg} \; (T_2 = 44.45°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.9 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.9\text{ MPa}} = 101.61 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 101.61 \text{ kJ/kg} \; (\text{throttling})$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 101.61) \text{ kJ/kg} = \mathbf{6.77\ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(278.93 - 236.97) \text{ kJ/kg} = \mathbf{2.10\ kW}$$

(b) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 6.77 + 2.10 = \mathbf{8.87\ kW}$$

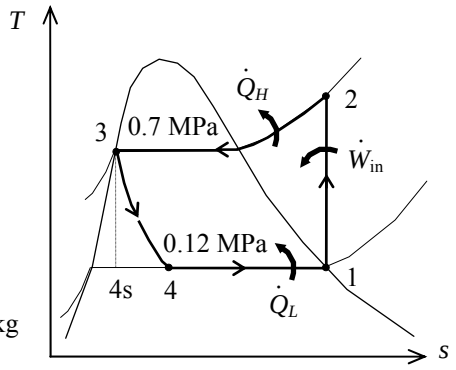
(c) The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{6.77 \text{ kW}}{2.10 \text{ kW}} = \mathbf{3.23}$$

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11-16 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The throttling valve in the cycle is replaced by an isentropic turbine. The percentage increase in the COP and in the rate of heat removal from the refrigerated space due to this replacement are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis If the throttling valve in the previous problem is replaced by an isentropic turbine, we would have

$$s_{4s} = s_3 = s_{f\,@\,0.7\text{ MPa}} = 0.33230 \text{ kJ/kg·K}$$

and the enthalpy at the turbine exit would be

$$x_{4s} = \left(\frac{s_3 - s_f}{s_{fg}} \right)_{@\,120\text{ kPa}} = \frac{0.33230 - 0.09275}{0.85503} = 0.2802$$

$$h_{4s} = \left(h_f + x_{4s} h_{fg} \right)_{@\,120\text{ kPa}} = 22.49 + (0.2802)(214.48) = 82.58 \text{ kJ/kg}$$

Then,

$$\dot{Q}_L = \dot{m}(h_1 - h_{4s}) = (0.05 \text{ kg/s})(236.97 - 82.58) \text{ kJ/kg} = 7.72 \text{ kW}$$

and

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.72 \text{ kW}}{1.83 \text{ kW}} = 4.23$$

Then the percentage increase in $\dot{Q}$ and COP becomes

$$\text{Increase in } \dot{Q}_L = \frac{\Delta \dot{Q}_L}{\dot{Q}_L} = \frac{7.72 - 7.41}{7.41} = \mathbf{4.2\%}$$

$$\text{Increase in } \text{COP}_\text{R} = \frac{\Delta \text{COP}_\text{R}}{\text{COP}_\text{R}} = \frac{4.23 - 4.06}{4.06} = \mathbf{4.2\%}$$

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11-17 A refrigerator with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the isentropic efficiency of the compressor, and the COP of the refrigerator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the refrigerant tables (Tables A-12 and A-13),

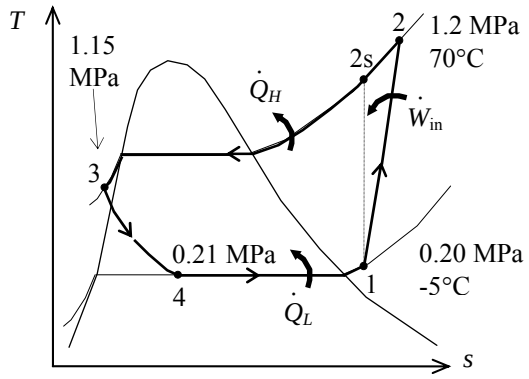
$$\left.\begin{matrix} P_1 = 0.20 \text{ MPa} \\ T_1 = -5°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 248.80 \text{ kJ/kg} \\ s_1 = 0.95407 \text{ kJ/kg} \cdot \text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 1.2 \text{ MPa} \\ T_2 = 70°\text{C} \end{matrix}\right\} h_2 = 300.61 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_{2s} = 1.2 \text{ MPa} \\ s_{2s} = s_1 \end{matrix}\right\} h_{2s} = 287.21 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_3 = 1.15 \text{ MPa} \\ T_3 = 44°\text{C} \end{matrix}\right\} h_3 = h_{f\ @\ 44°\text{C}} = 114.28 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 114.28 \text{ kJ/kg} \ (\text{throttling})$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.07 \text{ kg/s})(248.80 - 114.28) \text{ kJ/kg} = \mathbf{9.42\ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.07 \text{ kg/s})(300.61 - 248.80) \text{ kJ/kg} = \mathbf{3.63\ kW}$$

(*b*) The isentropic efficiency of the compressor is determined from

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{287.21 - 248.80}{300.61 - 248.80} = 0.741 = \mathbf{74.1\%}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{9.42 \text{ kW}}{3.63 \text{ kW}} = \mathbf{2.60}$$

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11-18E An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The mass flow rate of the refrigerant and the power requirement are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

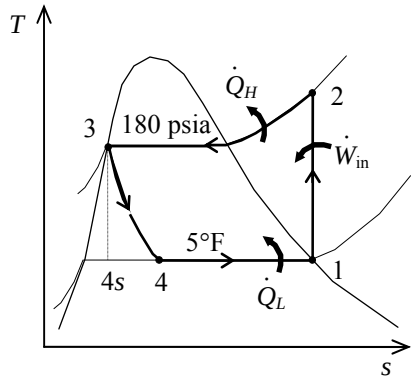
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-11E, A-12E, and A-13E),

$$\left.\begin{array}{l} T_1 = 5°\text{F} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,50°\text{F}} = 103.82\ \text{Btu/lbm} \\ s_1 = s_{g\,@\,5°\text{F}} = 0.22485\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 180\ \text{psia} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 121.99\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 180\ \text{psia} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\,@\,180\ \text{psia}} = 51.50\ \text{Btu/lbm}$$

$$h_4 \cong h_3 = 51.50\ \text{Btu/lbm} \quad \text{(throttling)}$$



The mass flow rate of the refrigerant is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) \longrightarrow \dot{m} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{45{,}000\ \text{Btu/h}}{(103.82 - 51.50)\ \text{Btu/lbm}} = \mathbf{860.1\ lbm/h}$$

The power requirement is

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (860.1\ \text{lbm/h})(121.99 - 103.82)\ \text{Btu/lbm}\left(\frac{1\ \text{kW}}{3412.14\ \text{Btu/h}}\right) = \mathbf{4.582\ kW}$$

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11-19E Problem 11-18E is to be repeated if ammonia is used as the refrigerant.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
x[1]=1
T[1]=5 [F]
x[3]=0
P[3]=180 [psia]
Q_dot_L=45000 [Btu/h]

"Analysis"
Fluid$='ammonia'
"compressor"
h[1]=enthalpy(Fluid$, T=T[1], x=x[1])
s[1]=entropy(Fluid$, T=T[1], x=x[1])
s[2]=s[1]
P[2]=P[3]
h[2]=enthalpy(Fluid$, P=P[2], s=s[2])
"expansion valve"
h[3]=enthalpy(Fluid$, P=P[3], x=x[3])
h[4]=h[3]
"cycle"
m_dot_R=Q_dot_L/(h[1]-h[4])
W_dot_in=m_dot_R*(h[2]-h[1])*Convert(Btu/h, kW)
```

Solution for ammonia

```
COP_R=4.515
Fluid$='ammonia'
m_dot_R=95.8 [lbm/h]
Q_dot_L=45000 [Btu/h]
W_dot_in=2.921 [kW]
```

Solution for R-134a

```
COP_R=2.878
Fluid$='R134a'
m_dot_R=860.1 [lbm/h]
Q_dot_L=45000 [Btu/h]
W_dot_in=4.582 [kW]
```

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11-20 A commercial refrigerator with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the evaporator inlet, the refrigeration load, the COP of the refrigerator, and the theoretical maximum refrigeration load for the same power input to the compressor are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From refrigerant-134a tables (Tables A-11 through A-13)

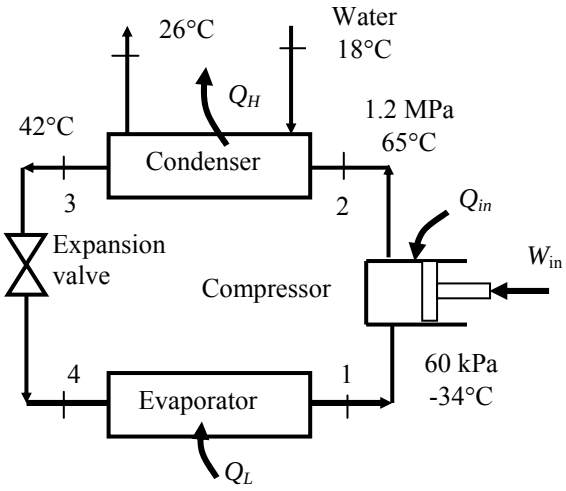
$$\left.\begin{matrix} P_1 = 60\ \text{kPa} \\ T_1 = -34°\text{C} \end{matrix}\right\} h_1 = 230.03\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_2 = 1200\ \text{kPa} \\ T_2 = 65°\text{C} \end{matrix}\right\} h_2 = 295.16\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 1200\ \text{kPa} \\ T_3 = 42°\text{C} \end{matrix}\right\} h_3 = 111.23\ \text{kJ/kg}$$

$$h_4 = h_3 = 111.23\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_4 = 60\ \text{kPa} \\ h_4 = 111.23\ \text{kJ/kg} \end{matrix}\right\} x_4 = \mathbf{0.4795}$$



Using saturated liquid enthalpy at the given temperature, for water we have (Table A-4)

$$h_{w1} = h_{f@18°\text{C}} = 75.47\ \text{kJ/kg}$$

$$h_{w2} = h_{f@26°\text{C}} = 108.94\ \text{kJ/kg}$$

(*b*) The mass flow rate of the refrigerant may be determined from an energy balance on the compressor

$$\dot{m}_R(h_2 - h_3) = \dot{m}_w(h_{w2} - h_{w1})$$

$$\dot{m}_R(295.16 - 111.23)\text{kJ/kg} = (0.25\ \text{kg/s})(108.94 - 75.47)\text{kJ/kg}$$

$$\longrightarrow \dot{m}_R = 0.0455\ \text{kg/s}$$

The waste heat transferred from the refrigerant, the compressor power input, and the refrigeration load are

$$\dot{Q}_H = \dot{m}_R(h_2 - h_3) = (0.0455\ \text{kg/s})(295.16 - 111.23)\text{kJ/kg} = 8.367\ \text{kW}$$

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) - \dot{Q}_{\text{in}} = (0.0455\ \text{kg/s})(295.16 - 230.03)\text{kJ/kg} - 0.450\ \text{kW} = 2.513\ \text{kW}$$

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} - \dot{Q}_{\text{in}} = 8.367 - 2.513 - 0.450 = \mathbf{5.404\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition

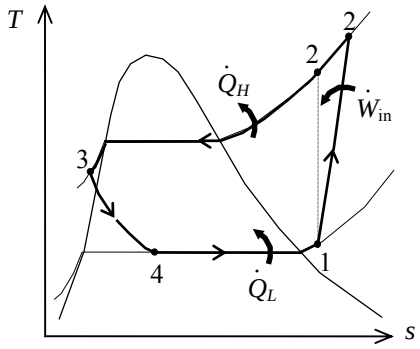
$$\text{COP} = \frac{\dot{Q}_{\text{L}}}{\dot{W}_{\text{in}}} = \frac{5.404}{2.513} = \mathbf{2.15}$$

(*d*) The reversible COP of the refrigerator for the same temperature limits is

$$\text{COP}_{\text{max}} = \frac{1}{T_H / T_L - 1} = \frac{1}{(18 + 273)/(-30 + 273) - 1} = 5.063$$

Then, the maximum refrigeration load becomes

$$\dot{Q}_{\text{L,max}} = COP_{\text{max}}\dot{W}_{\text{in}} = (5.063)(2.513\ \text{kW}) = \mathbf{12.72\ kW}$$



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11-21 A refrigerator with refrigerant-134a as the working fluid is considered. The power input to the compressor, the rate of heat removal from the refrigerated space, and the pressure drop and the rate of heat gain in the line between the evaporator and the compressor are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the refrigerant tables (Tables A-12 and A-13),

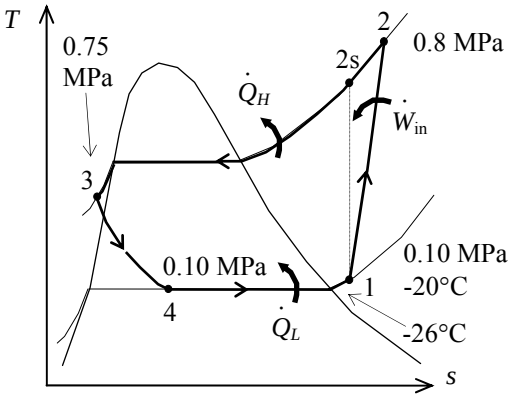
$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ T_1 = -20°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 239.50 \text{ kJ/kg} \\ s_1 = 0.97207 \text{ kJ/kg} \cdot \text{K} \\ v_1 = 0.19841 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 284.07 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.75 \text{ MPa} \\ T_3 = 26°\text{C} \end{array}\right\} h_3 \cong h_{f\,@\,26\,°\text{C}} = 87.83 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 87.83 \text{ kJ/kg} \ (\text{throttling})$$

$$\left.\begin{array}{l} T_5 = -26°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} P_5 = 0.10173 \text{ MPa} \\ h_5 = 234.68 \text{ kJ/kg} \end{array}$$



Then the mass flow rate of the refrigerant and the power input becomes

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.5/60 \text{ m}^3/\text{s}}{0.19841 \text{ m}^3/\text{kg}} = 0.0420 \text{ kg/s}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_{2s} - h_1)/\eta_C = (0.0420 \text{ kg/s})[(284.07 - 239.50) \text{ kJ/kg}]/(0.78) = \mathbf{2.40\ kW}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}(h_5 - h_4) = (0.0420 \text{ kg/s})(234.68 - 87.83) \text{ kJ/kg} = \mathbf{6.17\ kW}$$

(*c*) The pressure drop and the heat gain in the line between the evaporator and the compressor are

$$\Delta P = P_5 - P_1 = 101.73 - 100 = \mathbf{1.73}$$

and

$$\dot{Q}_{\text{gain}} = \dot{m}(h_1 - h_5) = (0.0420 \text{ kg/s})(239.50 - 234.68) \text{ kJ/kg} = \mathbf{0.203\ kW}$$

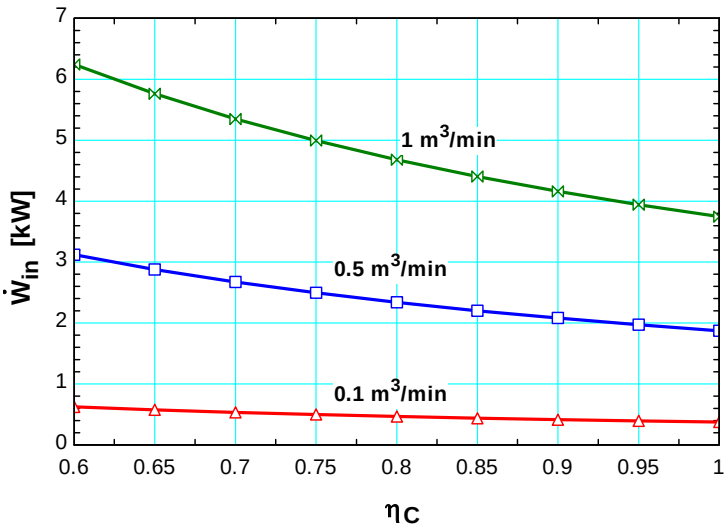
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11-22 Problem 11-21 is reconsidered. The effects of the compressor isentropic efficiency and the compressor inlet volume flow rate on the power input and the rate of refrigeration are to be investigated.

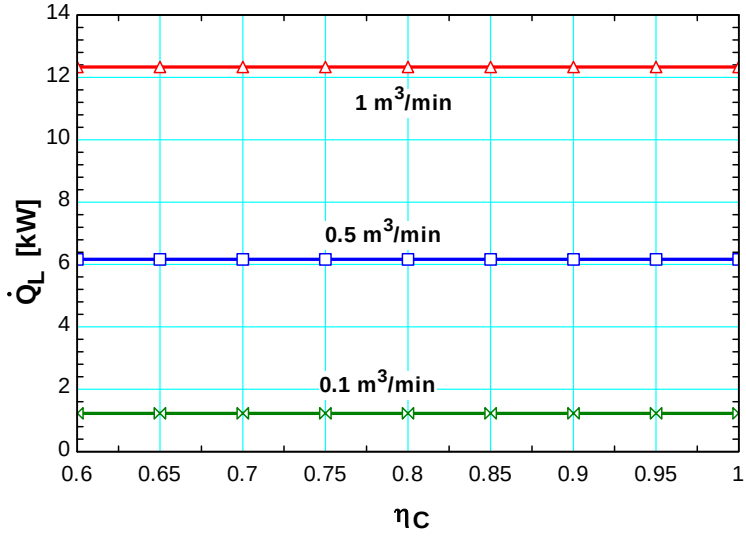
Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
P[1]=100 [kPa]
T[1]=-20 [C]
V_dot=0.5 [m^3/min]
P[2]=800 [kPa]
"Eta_C=0.78"
P[3]=750 [kPa]
T[3]=26  [C]
T[5]=-26 [C]
x[5]=1

"Analysis"
Fluid$='R134a'
"compressor"
h[1]=enthalpy(Fluid$, P=P[1], T=T[1])
s[1]=entropy(Fluid$, P=P[1], T=T[1])
v[1]=volume(Fluid$, P=P[1], T=T[1])
s_s[2]=s[1]
h_s[2]=enthalpy(Fluid$, P=P[2], s=s_s[2])
"expansion valve"
x[3]=0  "assumed saturated liquid"
h[3]=enthalpy(Fluid$, T=T[3], x=x[3])
h[4]=h[3]
"evaporator exit"
h[5]=enthalpy(Fluid$, T=T[5], x=x[5])
P[5]=pressure(Fluid$, T=T[5], x=x[5])
"cycle"
m_dot=V_dot/v[1]*Convert(kg/min, kg/s)
W_dot_in=m_dot*(h_s[2]-h[1])/Eta_C
Q_dot_L=m_dot*(h[5]-h[4])
DELTAP=P[5]-P[1]
Q_dot_gain=m_dot*(h[1]-h[5])
```



η_c	W_{in} [kW]	Q_L [kW]
0.6	3.12	6.168
0.65	2.88	6.168
0.7	2.674	6.168
0.75	2.496	6.168
0.8	2.34	6.168
0.85	2.202	6.168
0.9	2.08	6.168
0.95	1.971	6.168
1	1.872	6.168



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11-23 A refrigerator uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle except for the compression process. The mass flow rate of the refrigerant, the condenser pressure, and the COP of the refrigerator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) (*b*) From the refrigerant-134a tables (Tables A-11 through A-13)

$$\left.\begin{matrix} P_4 = 120\ \text{kPa} \\ x_4 = 0.30 \end{matrix}\right\} h_4 = 86.83\ \text{kJ/kg}$$

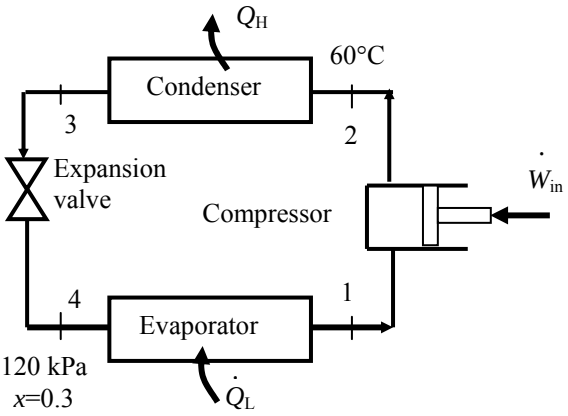
$$h_3 = h_4$$

$$\left.\begin{matrix} h_3 = 86.83\ \text{kJ/kg} \\ x_3 = 0\ (\text{sat. liq.}) \end{matrix}\right\} P_3 = \mathbf{671.8\ kPa}$$

$$P_2 = P_3$$

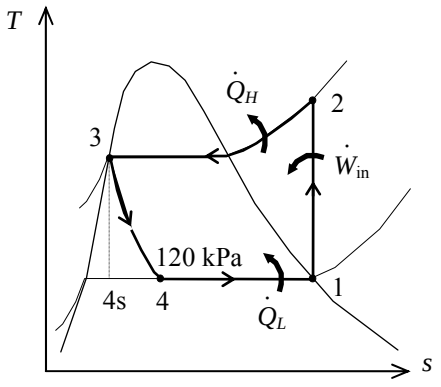
$$\left.\begin{matrix} P_2 = 671.8\ \text{kPa} \\ T_2 = 60°\text{C} \end{matrix}\right\} h_2 = 298.87\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_1 = P_4 = 120\ \text{kPa} \\ x_1 = 1\ (\text{sat. vap.}) \end{matrix}\right\} h_1 = 236.97\ \text{kJ/kg}$$



The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{W}_{\text{in}}}{h_2 - h_1} = \frac{0.45\ \text{kW}}{(298.87 - 236.97)\text{kJ/kg}} = \mathbf{0.00727\ kg/s}$$



(*c*) The refrigeration load and the COP are

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$
$$= (0.0727\ \text{kg/s})(236.97 - 86.83)\text{kJ/kg}$$
$$= 1.091\ \text{kW}$$

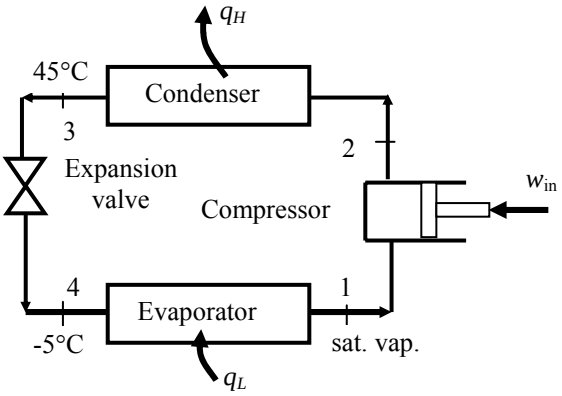
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{1.091\ \text{kW}}{0.45\ \text{kW}} = \mathbf{2.43}$$

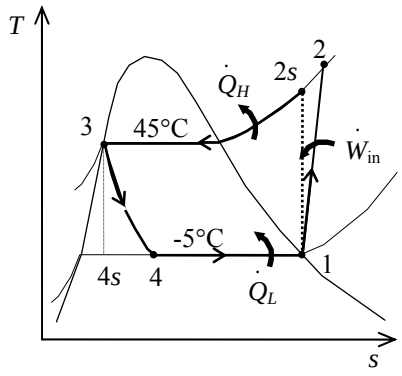
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11-24 A vapor-compression refrigeration cycle with refrigerant-22 as the working fluid is considered. The hardware and the *T-s* diagram for this air conditioner are to be sketched. The heat absorbed by the refrigerant, the work input to the compressor and the heat rejected in the condenser are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) In this normal vapor-compression refrigeration cycle, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure.





(b) The properties as given in the problem statement are

$$h_4 = h_3 = h_{f\,@\,45°C} = 101 \text{ kJ/kg}$$

$$h_1 = h_{g\,@\,-5°C} = 248.1 \text{ kJ/kg.}$$

The heat absorbed by the refrigerant in the evaporator is

$$q_L = h_1 - h_4 = 248.1 - 101 = \mathbf{147.1 kJ/kg}$$

(c) The COP of the air conditioner is

$$\text{COP}_R = \text{SEER}\left(\frac{1\text{ W}}{3.412\text{ Btu/h}}\right) = \left(16\frac{\text{Btu/h}}{\text{W}}\right)\left(\frac{1\text{ W}}{3.412\text{ Btu/h}}\right) = 4.689$$

The work input to the compressor is

$$\text{COP}_R = \frac{q_L}{w_{in}} \longrightarrow w_{in} = \frac{q_L}{\text{COP}_R} = \frac{147.1\text{ kJ/kg}}{4.689} = \mathbf{31.4\ kJ/kg}$$

The enthalpy at the compressor exit is

$$w_{in} = h_2 - h_1 \longrightarrow h_2 = h_1 + w_{in} = 248.1\text{ kJ/kg} + 31.4\text{ kJ/kg} = 279.5\text{ kJ/kg}$$

The heat rejected from the refrigerant in the condenser is then

$$q_H = h_2 - h_3 = 279.5 - 101 = \mathbf{178.5\ kJ/kg}$$

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11-25 A vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The amount of cooling, the work input, and the COP are to be determined. Also, the same parameters are to be determined if the cycle operated on the ideal vapor-compression refrigeration cycle between the same temperature limits.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

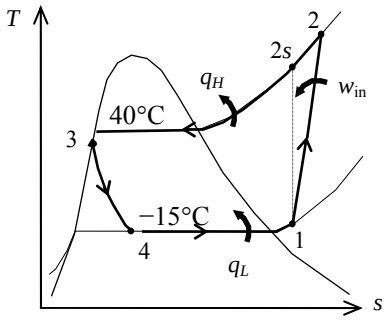
Analysis (a) The expansion process through the expansion valve is isenthalpic: $h_4 = h_3$. Then,

$$q_L = h_1 - h_4 = 402.49 - 243.19 = \mathbf{159.3\ kJ/kg}$$

$$q_H = h_2 - h_3 = 454.00 - 243.19 = 210.8\ \text{kJ/kg}$$

$$w_{\text{in}} = h_2 - h_1 = 454.00 - 402.49 = \mathbf{51.51\ kJ/kg}$$

$$\text{COP} = \frac{q_L}{w_{\text{in}}} = \frac{159.3\ \text{kJ/kg}}{51.51\ \text{kJ/kg}} = \mathbf{3.093}$$



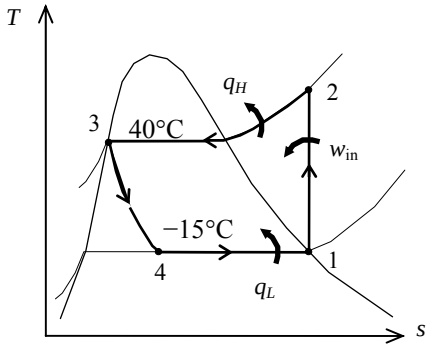
(c) Ideal vapor-compression refrigeration cycle solution:

$$q_L = h_1 - h_4 = 399.04 - 249.80 = \mathbf{149.2\ kJ/kg}$$

$$q_H = h_2 - h_3 = 440.71 - 249.80 = 190.9\ \text{kJ/kg}$$

$$w_{\text{in}} = h_2 - h_1 = 440.71 - 399.04 = \mathbf{41.67\ kJ/kg}$$

$$\text{COP} = \frac{q_L}{w_{\text{in}}} = \frac{149.2\ \text{kJ/kg}}{41.67\ \text{kJ/kg}} = \mathbf{3.582}$$



Discussion In the ideal operation, the refrigeration load decreases by 6.3% and the work input by 19.1% while the COP increases by 15.8%.

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11-26 A vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of cooling, the power input, and the COP are to be determined. Also, the same parameters are to be determined if the cycle operated on the ideal vapor-compression refrigeration cycle between the same pressure limits.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) From the refrigerant-134a tables (Tables A-11 through A-13)

$$T_{\text{sat@200 kPa}} = -10.1°\text{C}$$

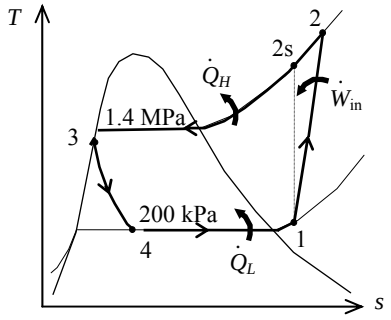
$$\left.\begin{aligned} P_1 &= 200\ \text{kPa} \\ T_1 &= -10.1 + 10.1 = 0°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 253.05\ \text{kJ/kg} \\ s_1 &= 0.9698\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 1400\ \text{kPa} \\ s_1 &= s_1 \end{aligned}\right\} h_{2s} = 295.90\ \text{kJ/kg}$$

$$T_{\text{sat@1400 kPa}} = 52.4°\text{C}$$

$$\left.\begin{aligned} P_3 &= 1400\ \text{kPa} \\ T_3 &= 52.4 - 4.4 = 48°\text{C} \end{aligned}\right\} h_3 \cong h_{f@48°\text{C}} = 120.39\ \text{kJ/kg}$$

$$h_4 = h_3 = 120.39\ \text{kJ/kg}$$



$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.88 = \frac{295.90 - 253.05}{h_2 - 253.05} \longrightarrow h_2 = 301.74\ \text{kJ/kg}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.025\ \text{kg/s})(253.05 - 120.39) = \mathbf{3.317\ kW}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.025\ \text{kg/s})(301.74 - 120.39) = \mathbf{4.534\ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.025\ \text{kg/s})(301.74 - 253.05) = \mathbf{1.217\ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{3.317\ \text{kW}}{1.217\ \text{kW}} = \mathbf{2.725}$$

(b) Ideal vapor-compression refrigeration cycle solution

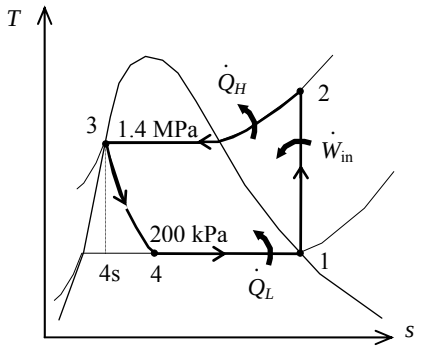
From the refrigerant-134a tables (Tables A-11 through A-13)

$$\left.\begin{aligned} P_1 &= 200\ \text{kPa} \\ x_1 &= 1 \end{aligned}\right\} \begin{aligned} h_1 &= 244.46\ \text{kJ/kg} \\ s_1 &= 0.9377\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 1400\ \text{kPa} \\ s_1 &= s_1 \end{aligned}\right\} h_2 = 285.08\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 1400\ \text{kPa} \\ x_3 &= 0 \end{aligned}\right\} h_3 = 127.22\ \text{kJ/kg}$$

$$h_4 = h_3 = 127.22\ \text{kJ/kg}$$



$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.025\ \text{kg/s})(244.46 - 127.22) = \mathbf{3.931 kW}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.025\ \text{kg/s})(285.08 - 127.22) = \mathbf{3.947\ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.025\ \text{kg/s})(285.08 - 244.46) = \mathbf{1.016\ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{3.931\ \text{kW}}{1.016\ \text{kW}} = \mathbf{2.886}$$

Discussion The cooling load increases by 18.5% while the COP increases by 5.9% when the cycle operates on the ideal vapor-compression cycle.

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Second-Law Analysis of Vapor-Compression Refrigeration Cycles

11-27C The second-law efficiency of a refrigerator operating on the vapor-compression refrigeration cycle is defined as

$$\eta_{\text{II,R}} = \frac{\dot{X}_{\dot{Q}_L}}{\dot{W}} = \frac{\dot{W}_{\min}}{\dot{W}} = 1 - \frac{\dot{X}_{\text{dest,total}}}{\dot{W}}$$

where $\dot{X}_{\dot{Q}_L}$ is the exergy of the heat transferred from the low-temperature medium and it is expressed as

$$\dot{X}_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right).$$

$\dot{X}_{\text{dest,total}}$ is the total exergy destruction in the cycle and $\dot{W}$ is the actual power input to the cycle. The second-law efficiency can also be expressed as the ratio of the actual COP to the Carnot COP:

$$\eta_{\text{II,R}} = \frac{\text{COP}_{\text{R}}}{\text{COP}_{\text{Carnot}}}$$

11-28C The second-law efficiency of a heat pump operating on the a vapor-compression refrigeration cycle is defined as

$$\eta_{\text{II,HP}} = \frac{\dot{E}x_{\dot{Q}_H}}{\dot{W}} = \frac{\dot{W}_{\min}}{\dot{W}} = 1 - \frac{\dot{E}x_{\text{dest,total}}}{\dot{W}}$$

Substituting

$$\dot{W} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} \quad \text{and} \quad \dot{E}x_{\dot{Q}_H} = \dot{Q}_H\left(1 - \frac{T_0}{T_H}\right)$$

into the second-law efficiency equation

$$\eta_{\text{II,HP}} = \frac{\dot{E}x_{\dot{Q}_H}}{\dot{W}} = \frac{\dot{Q}_H\left(1 - \frac{T_0}{T_H}\right)}{\frac{\dot{Q}_H}{\text{COP}_{\text{HP}}}} = \dot{Q}_H\left(1 - \frac{T_0}{T_H}\right)\frac{\text{COP}_{\text{HP}}}{\dot{Q}_H} = \frac{\text{COP}_{\text{HP}}}{\frac{T_H}{T_H - T_L}} = \frac{\text{COP}_{\text{HP}}}{\text{COP}_{\text{Carnot}}}$$

since $T_0 = T_L$.

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11-29C In an isentropic compressor, $s_2 = s_1$ and $h_{2s} = h_2$. Applying these to the two the efficiency definitions, we obtain

$$\eta_{\text{s,Comp}} = \frac{w_{\text{isen}}}{w} = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{h_2 - h_1}{h_2 - h_1} = 1 = 100\%$$

$$\eta_{\text{II,Comp}} = \frac{w_{\text{rev}}}{w} = \frac{h_2 - h_1 - T_0(s_2 - s_1)}{h_2 - h_1} = \frac{h_2 - h_1}{h_2 - h_1} = 1 = 100\%$$

Thus, the isentropic efficiency and the exergy efficiency of an isentropic compressor are both 100%.

The exergy efficiency of a compressor is not necessarily equal to its isentropic efficiency. The two definitions are different as shown in the above equations. In the calculation of isentropic efficiency, the exit enthalpy is found at the hypothetical exit state (at the exit pressure and the inlet entropy) while the exergy efficiency involves the actual exit state. The two efficiencies are usually close but different. In the special case of an isentropic compressor, the two efficiencies become equal to each other as proven above.

11-30 A vapor-compression refrigeration system is used to keep a space at a low temperature. The power input, the COP and the second-law efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The power input is

$$\dot{W}_{\text{in}} = \dot{Q}_H - \dot{Q}_L = 6000 - 3500 = 2500 \text{ kJ/h} = (2500 \text{ kJ/h})\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{0.6944 \text{ kW}}$$

The COP is

$$\text{COP}_{\text{R}} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{(3500/3600) \text{ kW}}{0.6944 \text{ kW}} = \mathbf{1.4}$$

The COP of the Carnot cycle operating between the space and the ambient is

$$\text{COP}_{\text{Carnot}} = \frac{T_L}{T_H - T_L} = \frac{250 \text{ K}}{(298 - 250) \text{ K}} = 5.208$$

The second-law efficiency is then

$$\eta_{\text{II}} = \frac{\text{COP}_{\text{R}}}{\text{COP}_{\text{Carnot}}} = \frac{1.4}{5.208} = 0.2688 = \mathbf{26.9\%}$$

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11-31 A refrigerator is used to cool bananas at a specified rate. The rate of heat absorbed from the bananas, the COP, The minimum power input, the second-law efficiency and the exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The rate of heat absorbed from the bananas is

$$\dot{Q}_L = \dot{m}c_p(T_1 - T_2) = (1140\ \text{kg/h})(3.35\ \text{kJ/kg}\cdot°\text{C})(28-12)°\text{C} = \mathbf{61{,}100\ kJ/h}$$

The COP is

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{(61{,}100/3600)\ \text{kW}}{8.6\ \text{kW}} = \frac{16.97\ \text{kW}}{8.6\ \text{kW}} = \mathbf{1.97}$$

(b) Theminimum power input is equal to the exergy of the heat transferred from the low-temperature medium:

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right) = -(16.97\ \text{kW})\left(1 - \frac{28+273}{20+273}\right) = \mathbf{0.463\ kW}$$

where the dead state temperature is taken as the inlet temperature of the eggplants (T_0 = 28°C) and the temperature of the low-temperature medium is taken as the average temperature of bananas T = (12+28)/2 = 20°C.

(c) The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{E}x_{\dot{Q}_L}}{\dot{W}_{\text{in}}} = \frac{0.463}{8.6} = 0.0539 = \mathbf{5.39\%}$$

The exergy destruction is the difference between the exergy expended (power input) and the exergy recovered (the exergy of the heat transferred from the low-temperature medium):

$$\dot{E}x_{\text{dest}} = \dot{W}_{\text{in}} - \dot{E}x_{\dot{Q}_L} = 8.6 - 0.463 = \mathbf{8.14\ kW}$$

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11-32 A vapor-compression refrigeration cycle is used to keep a space at a low temperature. The power input, the mass flow rate of water in the condenser, the second-law efficiency, and the exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The power input is

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}} = \frac{(24{,}000\ \text{Btu/h})\left(\dfrac{1\ \text{kW}}{3412\ \text{Btu/h}}\right)}{2.05} = \frac{7.034\ \text{kW}}{2.05} = \mathbf{3.431\ kW}$$

(b) From an energy balance on the cycle,

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.034 + 3.431 = 10.46\ \text{kW}$$

The mass flow rate of the water is then determined from

$$\dot{Q}_H = \dot{m} c_{pw} \Delta T_w \longrightarrow \dot{m} = \frac{\dot{Q}_H}{c_{pw} \Delta T_w} = \frac{10.46\ \text{kW}}{(4.18\ \text{kJ/kg} \cdot °\text{C})(12°\text{C})} = \mathbf{0.2086\ kg/s}$$

(c) The exergy of the heat transferred from the low-temperature medium is

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right) = -(7.034\ \text{kW})\left(1 - \frac{20 + 273}{0 + 273}\right) = 0.5153\ \text{kW}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{E}x_{\dot{Q}_L}}{\dot{W}_{\text{in}}} = \frac{0.5153}{3.431} = 0.1502 = \mathbf{15.0\%}$$

The exergy destruction is the difference between the exergy supplied (power input) and the exergy recovered (the exergy of the heat transferred from the low-temperature medium):

$$\dot{E}x_{\text{dest}} = \dot{W}_{\text{in}} - \dot{E}x_{\dot{Q}_L} = 3.431 - 0.5153 = \mathbf{2.916\ kW}$$

Alternative Solution

The exergy efficiency can also be determined as follows:

$$\text{COP}_{\text{R,Carnot}} = \frac{T_L}{T_H - T_L} = \frac{0 + 273}{20 - 0} = 13.65$$

$$\eta_{\text{II}} = \frac{\text{COP}}{\text{COP}_{\text{R,Carnot}}} = \frac{2.05}{13.65} = 0.1502 = 15.0\%$$

The result is identical as expected.

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11-33E A vapor-compression refrigeration cycle is used to keep a space at a low temperature. The mass flow rate of R-134a, the COP, The exergy destruction in each component and the exergy efficiency of the compressor, the second-law efficiency, and the exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The properties of R-134a are (Tables A-11E through A-13E)

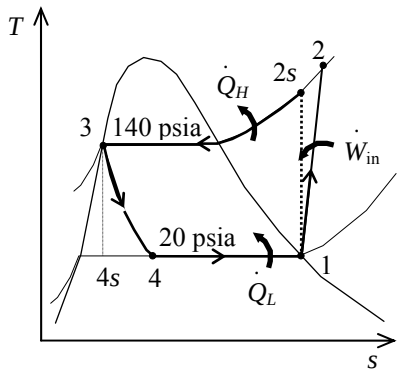
$$\left.\begin{matrix} P_1 = 20\ \text{psia} \\ x_1 = 1 \end{matrix}\right\} \begin{matrix} h_1 = 102.73\ \text{Btu/lbm} \\ s_1 = 0.2257\ \text{Btu/lbm}\cdot\text{R} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 140\ \text{psia} \\ T_2 = 160°\text{F} \end{matrix}\right\} \begin{matrix} h_2 = 131.36\ \text{Btu/lbm} \\ s_2 = 0.2444\ \text{Btu/lbm}\cdot\text{R} \end{matrix}$$

$$\left.\begin{matrix} P_3 = 140\ \text{psia} \\ x_3 = 0 \end{matrix}\right\} \begin{matrix} h_3 = 45.30\ \text{Btu/lbm} \\ s_3 = 0.0921\ \text{Btu/lbm}\cdot\text{R} \end{matrix}$$

$$h_4 = h_3 = 45.30\ \text{Btu/lbm}$$

$$\left.\begin{matrix} P_4 = 20\ \text{psia} \\ h_4 = 45.30\ \text{Btu/lbm} \end{matrix}\right\} s_4 = 0.1001\ \text{Btu/lbm}\cdot\text{R}$$



The energy interactions in each component and the mass flow rate of R-134a are

$$w_{in} = h_2 - h_1 = 131.36 - 102.73 = 28.63\ \text{Btu/lbm}$$

$$q_H = h_2 - h_3 = 131.36 - 45.30 = 86.06\ \text{Btu/lbm}$$

$$q_L = q_H - w_{in} = 86.06 - 28.63 = 57.43\ \text{Btu/lbm}$$

$$\dot{m} = \frac{\dot{Q}_L}{q_L} = \frac{(45{,}000/3600)\ \text{Btu/s}}{57.43\ \text{Btu/lbm}} = \mathbf{0.2177\ lbm/s}$$

The COP is

$$\text{COP} = \frac{q_L}{w_{in}} = \frac{57.43\ \text{Btu/lbm}}{28.63\ \text{Btu/lbm}} = \mathbf{2.006}$$

(b) The exergy destruction in each component of the cycle is determined as follows:

Compressor:

$$s_{gen,1-2} = s_2 - s_1 = 0.2444 - 0.2257 = 0.01874\ \text{Btu/lbm}\cdot\text{R}$$

$$\dot{E}x_{dest,1\text{-}2} = \dot{m}T_0 s_{gen,1\text{-}2} = (0.2177\ \text{lbm/s})(540\ \text{R})(0.01874\ \text{Btu/lbm}\cdot\text{R}) = \mathbf{2.203\ Btu/s}$$

Condenser:

$$s_{gen,2-3} = s_3 - s_2 + \frac{q_H}{T_H} = (0.0921 - 0.2444)\ \text{Btu/lbm}\cdot\text{R} + \frac{86.06\ \text{Btu/lbm}}{540\ \text{R}} = 0.007073\ \text{Btu/lbm}\cdot\text{R}$$

$$\dot{E}x_{dest,2\text{-}3} = \dot{m}T_0 s_{gen,2\text{-}3} = (0.2177\ \text{lbm/s})(540\ \text{R})(0.007073\ \text{Btu/lbm}\cdot\text{R}) = \mathbf{0.8313\ Btu/s}$$

Expansion valve:

$$s_{gen,3-4} = s_4 - s_3 = 0.1001 - 0.0921 = 0.007962\ \text{Btu/lbm}\cdot\text{R}$$

$$\dot{E}x_{dest,3\text{-}4} = \dot{m}T_0 s_{gen,3\text{-}4} = (0.2177\ \text{lbm/s})(540\ \text{R})(0.007962\ \text{Btu/lbm}\cdot\text{R}) = \mathbf{0.9359\ Btu/s}$$

Evaporator:

$$s_{gen,4-1} = s_1 - s_4 - \frac{q_L}{T_L} = (0.2257 - 0.1001)\ \text{Btu/lbm}\cdot\text{R} - \frac{57.43\ \text{Btu/lbm}}{470\ \text{R}} = 0.003400\ \text{Btu/lbm}\cdot\text{R}$$

$$\dot{E}x_{dest,4\text{-}1} = \dot{m}T_0 s_{gen,4\text{-}1} = (0.2177\ \text{lbm/s})(540\ \text{R})(0.003400\ \text{Btu/lbm}\cdot\text{R}) = \mathbf{0.3996\ Btu/s}$$

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The power input and the exergy efficiency of the compressor is determined from

$$\dot{W}_{\text{in}} = \dot{m} w_{\text{in}} = (0.2177 \text{ lbm/s})(28.63 \text{ Btu/lbm}) = 6.232 \text{ Btu/s}$$

$$\eta_{\text{II}} = 1 - \frac{\dot{E}x_{\text{dest},1-2}}{\dot{W}_{\text{in}}} = 1 - \frac{2.203 \text{ Btu/s}}{6.232 \text{ Btu/s}} = 0.6465 = \mathbf{64.7\%}$$

(c) The exergy of the heat transferred from the low-temperature medium is

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L \left(1 - \frac{T_0}{T_L}\right) = -(45000/3600 \text{ Btu/s})\left(1 - \frac{540}{470}\right) = 1.862 \text{ Btu/s}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{E}x_{\dot{Q}_L}}{\dot{W}_{\text{in}}} = \frac{1.862 \text{ Btu/s}}{6.232 \text{ Btu/s}} = 0.2987 = \mathbf{29.9\%}$$

The total exergy destruction in the cycle is the difference between the exergy supplied (power input) and the exergy recovered (the exergy of the heat transferred from the low-temperature medium):

$$\dot{E}x_{\text{dest,total}} = \dot{W}_{\text{in}} - \dot{E}x_{\dot{Q}_L} = 6.232 - 1.862 = \mathbf{4.370 \text{ Btu/s}}$$

The total exergy destruction can also be determined by adding exergy destructions in each component:

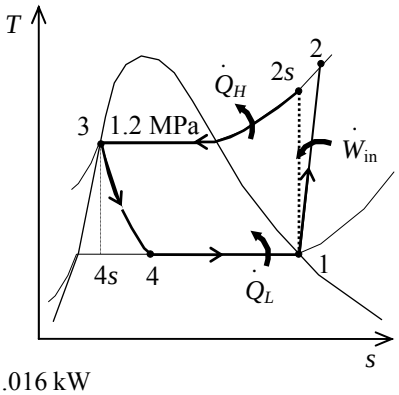
$$\begin{aligned}\dot{E}x_{\text{dest,total}} &= \dot{E}x_{\text{dest,1-2}} + \dot{E}x_{\text{dest,2-3}} + \dot{E}x_{\text{dest,3-4}} + \dot{E}x_{\text{dest,4-1}} \\ &= 2.203 + 0.8313 + 0.9359 + 0.3996 \\ &= 4.370 \text{ Btu/s}\end{aligned}$$

The result is the same as expected.

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11-34 A vapor-compression refrigeration cycle is used to keep a space at a low temperature. The mass flow rate of R-134a, the COP, The exergy destruction in each component and the exergy efficiency of the compressor, the second-law efficiency, and the exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis (a) The properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{aligned} P_2 &= 1.2\ \text{MPa} \\ T_2 &= 50°\text{C} \end{aligned}\right\} \begin{aligned} h_2 &= 278.27\ \text{kJ/kg} \\ s_2 &= 0.9267\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_3 &= 1.2\ \text{MPa} \\ x_3 &= 0 \end{aligned}\right\} \begin{aligned} h_3 &= 117.77\ \text{kJ/kg} \\ s_3 &= 0.4244\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

The rate of heat transferred to the water is the energy change of the water from inlet to exit

$$\dot{Q}_H = \dot{m}_w c_p (T_{w,2} - T_{w,1}) = (0.15\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})(28-20)°\text{C} = 5.016\ \text{kW}$$

The energy decrease of the refrigerant is equal to the energy increase of the water in the condenser. That is,

$$\dot{Q}_H = \dot{m}_R (h_2 - h_3) \longrightarrow \dot{m}_R = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{5.016\ \text{kW}}{(278.27 - 117.77)\ \text{kJ/kg}} = 0.03125\ \text{kg/s}$$

The refrigeration load is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = 5.016 - 2.2 = 2.816\ \text{kW} = (2.816\ \text{kW})\left(\frac{3412\ \text{Btu/h}}{1\ \text{kW}}\right) = \mathbf{9610\ Btu/h}$$

The COP of the refrigerator is determined from its definition,

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{2.816\ \text{kW}}{2.2\ \text{kW}} = \mathbf{1.28}$$

(*b*) The COP of a reversible refrigerator operating between the same temperature limits is

$$\text{COP}_{\text{Carnot}} = \frac{T_L}{T_H - T_L} = \frac{-12 + 273}{(20 + 273) - (-12 + 273)} = 8.156$$

The minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{Carnot}}} = \frac{2.816\ \text{kW}}{8.156} = 0.3453\ \text{kW}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{in,min}}}{\dot{W}_{\text{in}}} = \frac{0.3453}{2.2} = 0.1569 = \mathbf{15.7\%}$$

The total exergy destruction in the cycle is the difference between the actual and the minimum power inputs:

$$\dot{E}x_{\text{dest,total}} = \dot{W}_{\text{in}} - \dot{W}_{\text{in,min}} = 2.2 - 0.3453 = \mathbf{1.85\ kW}$$

(*c*) The entropy generation in the condenser is

$$\begin{aligned} \dot{S}_{\text{gen,cond}} &= \dot{m}_w c_p \ln\left(\frac{T_{w,2}}{T_{w,1}}\right) + \dot{m}_R (s_3 - s_2) \\ &= (0.15\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})\ln\left(\frac{28+273}{20+273}\right) + (0.03125\ \text{kg/s})(0.4004 - 0.9267)\ \text{kJ/kg}\cdot\text{K}) \\ &= 0.001191\ \text{kW/K} \end{aligned}$$

The exergy destruction in the condenser is

$$\dot{E}x_{\text{dest,cond}} = T_0 \dot{S}_{\text{gen,cond}} = (293\ \text{K})(0.001191\ \text{kW/K}) = \mathbf{0.349\ kW}$$

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11-35 An ideal vapor-compression refrigeration cycle is used to keep a space at a low temperature. The cooling load, the COP, the exergy destruction in each component, the total exergy destruction, and the second-law efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The properties of R-134a are (Tables A-11 through A-13)

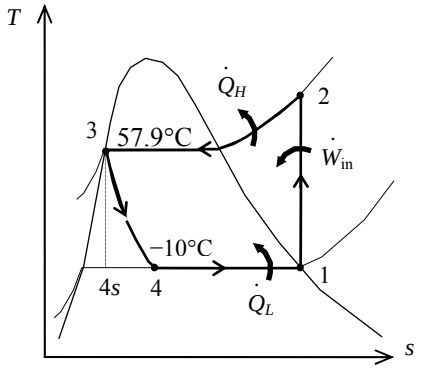
$$\left.\begin{matrix} T_1 = -10°\text{C} \\ x_1 = 1 \end{matrix}\right\} \begin{matrix} h_1 = 244.51\ \text{kJ/kg} \\ s_1 = 0.9377\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_2 = P_{\text{sat@57.9°C}} = 1600\ \text{kPa} \\ s_2 = s_1 \end{matrix}\right\} h_2 = 287.85\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 1600\ \text{kPa} \\ x_3 = 0 \end{matrix}\right\} \begin{matrix} h_3 = 135.93\ \text{kJ/kg} \\ s_3 = 0.4791\ \text{kJ/kg}\cdot\text{K} \end{matrix}$$

$$h_4 = h_3 = 135.93\ \text{kJ/kg}$$

$$\left.\begin{matrix} T_4 = -10°\text{C} \\ h_4 = 135.93\ \text{kJ/kg} \end{matrix}\right\} s_4 = 0.5251\ \text{kJ/kg}\cdot\text{K}$$



The energy interactions in the components and the COP are

$$q_L = h_1 - h_4 = 244.51 - 135.93 = \mathbf{108.6\ kJ/kg}$$

$$q_H = h_2 - h_3 = 287.85 - 135.93 = 151.9\ \text{kJ/kg}$$

$$w_{\text{in}} = h_2 - h_1 = 287.85 - 244.51 = 43.33\ \text{kJ/kg}$$

$$\text{COP} = \frac{q_L}{w_{\text{in}}} = \frac{108.6\ \text{kJ/kg}}{43.33\ \text{kJ/kg}} = \mathbf{2.506}$$

(b) The exergy destruction in each component of the cycle is determined as follows

Compressor:

$$s_{\text{gen},1-2} = s_2 - s_1 = 0$$

$$Ex_{\text{dest},1\text{-}2} = T_0 s_{\text{gen},1\text{-}2} = \mathbf{0}$$

Condenser:

$$s_{\text{gen},2-3} = s_3 - s_2 + \frac{q_H}{T_H} = (0.4791 - 0.9377)\ \text{kJ/kg}\cdot\text{K} + \frac{151.9\ \text{kJ/kg}}{298\ \text{K}} = 0.05124\ \text{kJ/kg}\cdot\text{K}$$

$$Ex_{\text{dest},2\text{-}3} = T_0 s_{\text{gen},2\text{-}3} = (298\ \text{K})(0.05124\ \text{kJ/kg}\cdot\text{K}) = \mathbf{15.27\ kJ/kg}$$

Expansion valve:

$$s_{\text{gen},3-4} = s_4 - s_3 = 0.5251 - 0.4791 = 0.04595\ \text{kJ/kg}\cdot\text{K}$$

$$Ex_{\text{dest},3\text{-}4} = T_0 s_{\text{gen},3\text{-}4} = (298\ \text{K})(0.04595\ \text{kJ/kg}\cdot\text{K}) = \mathbf{13.69\ kJ/kg}$$

Evaporator:

$$s_{\text{gen},4-1} = s_1 - s_4 - \frac{q_L}{T_L} = (0.9377 - 0.5251)\ \text{kJ/kg}\cdot\text{K} - \frac{108.6\ \text{kJ/kg}}{278\ \text{K}} = 0.02201\ \text{kJ/kg}\cdot\text{K}$$

$$Ex_{\text{dest},4\text{-}1} = T_0 s_{\text{gen},4\text{-}1} = (298\ \text{K})(0.02201\ \text{kJ/kg}\cdot\text{K}) = \mathbf{6.56\ kJ/kg}$$

The total exergy destruction can be determined by adding exergy destructions in each component:

$$\begin{aligned} \dot{E}x_{\text{dest,total}} &= \dot{E}x_{\text{dest},1\text{-}2} + \dot{E}x_{\text{dest},2\text{-}3} + \dot{E}x_{\text{dest},3\text{-}4} + \dot{E}x_{\text{dest},4\text{-}1} \\ &= 0 + 15.27 + 13.69 + 6.56 = \mathbf{35.52\ kJ/kg} \end{aligned}$$

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(c) The exergy of the heat transferred from the low-temperature medium is

$$Ex_{q_L} = -q_L\left(1-\frac{T_0}{T_L}\right) = -(108.6\ \text{kJ/kg})\left(1-\frac{298}{278}\right) = 7.812\ \text{kJ/kg}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{Ex_{q_L}}{w_{\text{in}}} = \frac{7.812}{43.33} = 0.1803 = \mathbf{18.0\%}$$

The total exergy destruction in the cycle can also be determined from

$$Ex_{\text{dest,total}} = w_{\text{in}} - Ex_{q_L} = 43.33 - 7.812 = 35.52\ \text{kJ/kg}$$

The result is identical as expected.

The second-law efficiency of the compressor is determined from

$$\eta_{\text{II,Comp}} = \frac{\dot{X}_{\text{recovered}}}{\dot{X}_{\text{expended}}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{act, in}}} = \frac{\dot{m}\left[h_2 - h_1 - T_0(s_2 - s_1)\right]}{\dot{m}(h_2 - h_1)}$$

since the compression through the compressor is isentropic ($s_2 = s_1$), the second-law efficiency is

$$\eta_{\text{II,Comp}} = 1 = \mathbf{100\%}$$

The second-law efficiency of the evaporator is determined from

$$\eta_{\text{II, Evap}} = \frac{\dot{X}_{\text{recovered}}}{\dot{X}_{\text{expended}}} = \frac{\dot{Q}_L(T_0 - T_L)/T_L}{\dot{m}\left[h_4 - h_1 - T_0(s_4 - s_1)\right]} = 1 - \frac{\dot{X}_{\text{dest,4-1}}}{\dot{X}_4 - \dot{X}_1}$$

where

$$\begin{aligned} x_4 - x_1 &= h_4 - h_1 - T_0(s_4 - s_1) \\ &= (135.93 - 244.51)\ \text{kJ/kg} - (298\ \text{K})(0.5251 - 0.9377)\ \text{kJ/kg}\cdot\text{K} \\ &= 14.37\ \text{kJ/kg} \end{aligned}$$

Substituting,

$$\eta_{\text{II, Evap}} = 1 - \frac{x_{\text{dest,4-1}}}{x_4 - x_1} = 1 - \frac{6.56\ \text{kJ/kg}}{14.37\ \text{kJ/kg}} = 0.544 = \mathbf{54.4\%}$$

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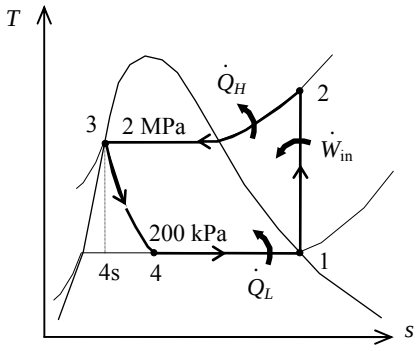
11-36 An ideal vapor-compression refrigeration cycle uses ammonia as the refrigerant. The volume flow rate at the compressor inlet, the power input, the COP, the second-law efficiency and the total exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The properties of ammonia are given in problem statement. An energy balance on the cındenser gives

$$q_H = h_1 - h_4 = 1439.3 - 437.4 = 1361\ \text{kJ/kg}$$

$$\dot{m} = \frac{\dot{Q}_H}{q_H} = \frac{18\ \text{kW}}{1361\ \text{kJ/kg}} = 0.01323\ \text{kg/s}$$



The volume flow rate is determined from

$$\dot{V}_1 = \dot{m} v_1 = (0.01323\ \text{kg/s})(0.5946\ \text{m}^3/\text{kg})$$
$$= 0.007865\ \text{m}^3/\text{s} = \mathbf{7.87\ L/s}$$

(b) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.01323\ \text{kg/s})(1798.3 - 1439.3)\text{kJ/kg} = \mathbf{4.75\ kW}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.01323\ \text{kg/s})(1439.3 - 437.4)\text{kJ/kg} = 13.25\ \text{kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{13.25\ \text{kW}}{4.75\ \text{kW}} = \mathbf{2.79}$$

(c) The exergy of the heat transferred from the low-temperature medium is

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right) = -(13.25\ \text{kW})\left(1 - \frac{300}{264}\right) = 1.81\ \text{kW}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{E}x_{\dot{Q}_L}}{\dot{W}_{\text{in}}} = \frac{1.81}{4.75} = 0.381 = \mathbf{38.1\%}$$

The total exergy destruction in the cycle is the difference between the exergy supplied (power input) and the exergy recovered (the exergy of the heat transferred from the low-temperature medium):

$$\dot{E}x_{\text{dest,total}} = \dot{W}_{\text{in}} - \dot{E}x_{\dot{Q}_L} = 4.75 - 1.81 = \mathbf{2.94\ kW}$$

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11-37 Prob. 11-36 is reconsidered. Using EES software, the problem is to be repeated ammonia, R-134a and R-22 is used as a refrigerant and the effects of evaporator and condenser pressures on the COP, the second-law efficiency and the total exergy destruction are to be investigated.

Analysis The equations as written in EES are

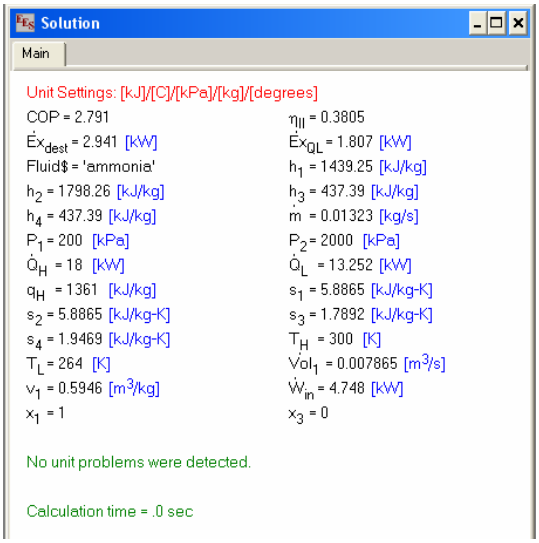
```
"GIVEN"
P_1=200 [kPa]
P_2=2000 [kPa]
Q_dot_H=18 [kW]
T_L=(-9+273) [K]
T_H=(27+273) [K]

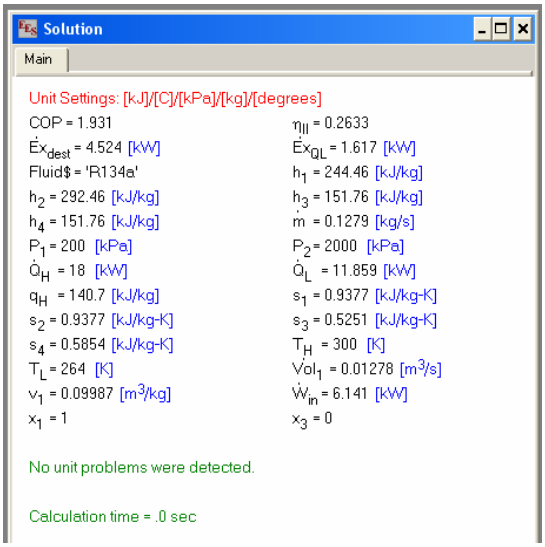
"PROPERTIES"
Fluid$='ammonia'
x_1=1
x_3=0
h_1=enthalpy(Fluid$, P=P_1, x=x_1)
s_1=entropy(Fluid$, P=P_1, x=x_1)
v_1=volume(Fluid$, P=P_1, x=x_1)
h_2=enthalpy(Fluid$, P=P_2, s=s_1)
s_2=s_1
h_3=enthalpy(Fluid$, P=P_2, x=x_3)
s_3=entropy(Fluid$, P=P_2, x=x_3)
h_4=h_3
s_4=entropy(Fluid$, P=P_1, h=h_4)

q_H=h_2-h_3
m_dot=Q_dot_H/q_H
Vol_dot_1=m_dot*v_1
Q_dot_L=m_dot*(h_1-h_4)
W_dot_in=m_dot*(h_2-h_1)
COP=Q_dot_L/W_dot_in
Ex_dot_QL=-Q_dot_L*(1-T_H/T_L)
eta_II=Ex_dot_QL/W_dot_in
Ex_dot_dest=W_dot_in-Ex_dot_QL
```



The solutions in the case of ammonia, R-134a and R-22 are





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Solution

Main

Unit Settings: [kJ]/[C]/[kPa]/[kg]/[degrees]

COP = 2.164	η_{II} = 0.2951
$\dot{E}x_{dest}$ = 4.01 [kW]	$\dot{E}x_{QL}$ = 1.679 [kW]
Fluid$ = 'R22'	h_1 = 394.67 [kJ/kg]
h_2 = 454.51 [kJ/kg]	h_3 = 265.17 [kJ/kg]
h_4 = 265.17 [kJ/kg]	$\dot{m}$ = 0.09507 [kg/s]
P_1 = 200 [kPa]	P_2 = 2000 [kPa]
$\dot{Q}_H$ = 18 [kW]	$\dot{Q}_L$ = 12.311 [kW]
q_H = 189.3 [kJ/kg]	s_1 = 1.7915 [kJ/kg-K]
s_2 = 1.7915 [kJ/kg-K]	s_3 = 1.2139 [kJ/kg-K]
s_4 = 1.2693 [kJ/kg-K]	T_H = 300 [K]
T_L = 264 [K]	$\dot{Vol}_1$ = 0.01068 [m^3/s]
v_1 = 0.1123 [m^3/kg]	$\dot{W}_{in}$ = 5.689 [kW]
x_1 = 1	x_3 = 0

No unit problems were detected.

Calculation time = .0 sec

Now, we investigate the effects of evaporating and condenser pressures on the COP, the second-law efficiency and the total exergy destruction. The results are given by tables and figures.

Parametric Table

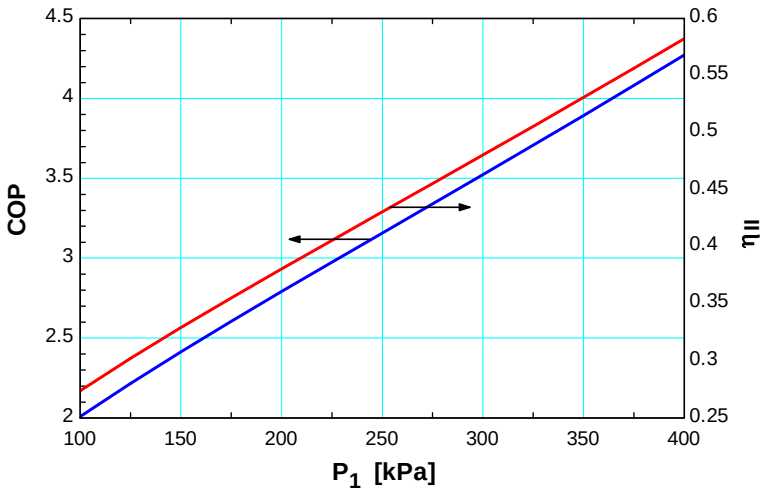
Table 1

1..13	1 P_1 [kPa]	2 COP	3 η_{II}	4 $\dot{E}x_{dest}$ [kW]
Run 1	100	2.008	0.2739	4.345
Run 2	125	2.216	0.3022	3.906
Run 3	150	2.413	0.3291	3.538
Run 4	175	2.604	0.3551	3.221
Run 5	200	2.791	0.3805	2.941
Run 6	225	2.975	0.4056	2.692
Run 7	250	3.157	0.4306	2.466
Run 8	275	3.34	0.4554	2.259
Run 9	300	3.523	0.4804	2.068
Run 10	325	3.707	0.5055	1.891
Run 11	350	3.893	0.5308	1.726
Run 12	375	4.081	0.5564	1.572
Run 13	400	4.271	0.5825	1.426

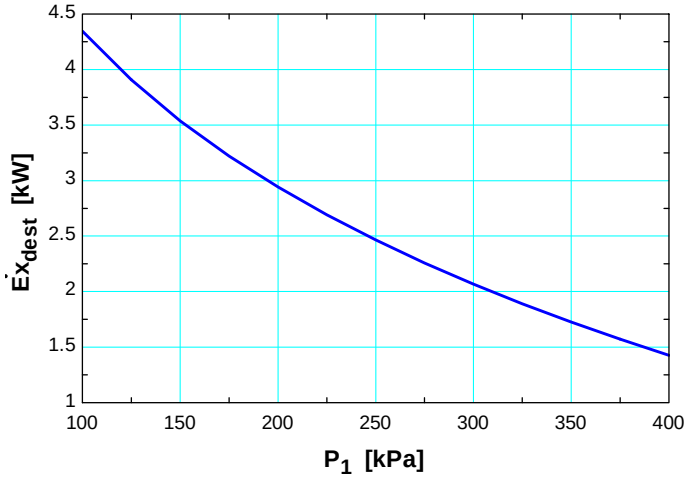
Parametric Table

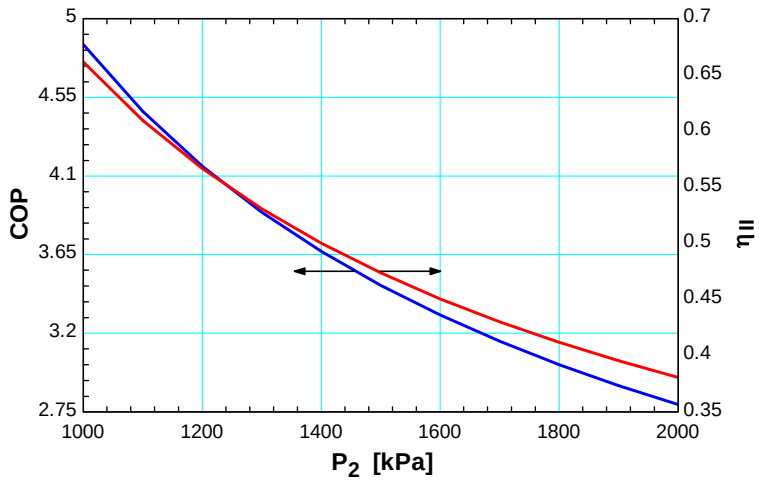
Table 1 Table 2

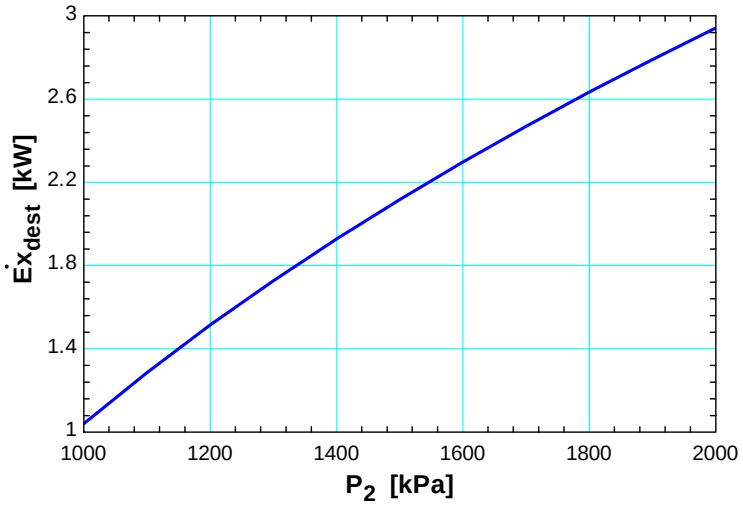
1..11	1 P_2 [kPa]	2 COP	3 η_{II}	4 $\dot{E}x_{dest}$ [kW]
Run 1	1000	4.851	0.6615	1.041
Run 2	1100	4.468	0.6093	1.286
Run 3	1200	4.154	0.5665	1.514
Run 4	1300	3.891	0.5306	1.727
Run 5	1400	3.667	0.5001	1.928
Run 6	1500	3.473	0.4737	2.118
Run 7	1600	3.304	0.4505	2.298
Run 8	1700	3.154	0.4301	2.47
Run 9	1800	3.02	0.4118	2.634
Run 10	1900	2.9	0.3954	2.791
Run 11	2000	2.791	0.3805	2.941



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Selecting the Right Refrigerant

11-38C The desirable characteristics of a refrigerant are to have an evaporator pressure which is above the atmospheric pressure, and a condenser pressure which corresponds to a saturation temperature above the temperature of the cooling medium. Other desirable characteristics of a refrigerant include being nontoxic, noncorrosive, nonflammable, chemically stable, having a high enthalpy of vaporization (minimizes the mass flow rate) and, of course, being available at low cost.

11-39C The minimum pressure that the refrigerant needs to be compressed to is the saturation pressure of the refrigerant at 30°C, which is **0.771 MPa**. At lower pressures, the refrigerant will have to condense at temperatures lower than the temperature of the surroundings, which cannot happen.

11-40C Allowing a temperature difference of 10°C for effective heat transfer, the evaporation temperature of the refrigerant should be -20°C. The saturation pressure corresponding to -20°C is 0.133 MPa. Therefore, the recommended pressure would be 0.12 MPa.

11-41 A refrigerator that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. Reasonable pressures for the evaporator and the condenser are to be selected.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis Allowing a temperature difference of 10°C for effective heat transfer, the evaporation and condensation temperatures of the refrigerant should be -20°C and 35°C, respectively. The saturation pressures corresponding to these temperatures are 0.133 MPa and 0.888 MPa. Therefore, the recommended evaporator and condenser pressures are **0.133 MPa** and **0.888 MPa**, respectively.

11-42 A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. Reasonable pressures for the evaporator and the condenser are to be selected.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis Allowing a temperature difference of 10°C for effective heat transfer, the evaporation and condensation temperatures of the refrigerant should be 4°C and 36°C, respectively. The saturation pressures corresponding to these temperatures are 338 kPa and 912 kPa. Therefore, the recommended evaporator and condenser pressures are **338 kPa** and **912 kPa**, respectively.

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Heat Pump Systems

11-43C A heat pump system is more cost effective in Miami because of the low heating loads and high cooling loads at that location.

11-44C A water-source heat pump extracts heat from water instead of air. Water-source heat pumps have higher COPs than the air-source systems because the temperature of water is higher than the temperature of air in winter.

11-45E A heat pump operating on the ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The COP of the heat pump is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

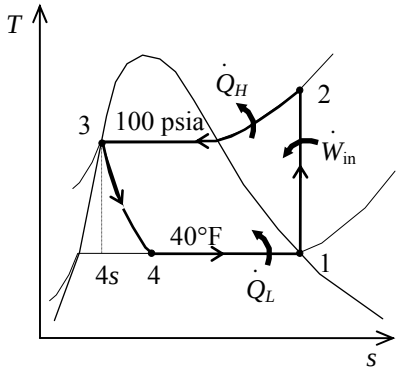
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-11E, A-12E, and A-13E),

$$\left.\begin{array}{l} T_1 = 40°\text{F} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 40°\text{F}} = 108.78\ \text{Btu/lbm} \\ s_1 = s_{g\ @\ 40°\text{F}} = 0.22189\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100\ \text{psia} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 114.98\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 100\ \text{psia} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\ @\ 100\ \text{psia}} = 37.869\ \text{Btu/lbm}$$

$$h_4 \cong h_3 = 37.869\ \text{Btu/lbm} \quad \text{(throttling)}$$



The COP of the heat pump is determined from its definition,

$$\text{COP}_{\text{HP}} = \frac{q_H}{w_{\text{in}}} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{114.98 - 37.869}{114.98 - 108.78} = \mathbf{12.43}$$

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11-46 A heat pump operating on the ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The COP and the rate of heat supplied to the evaporator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

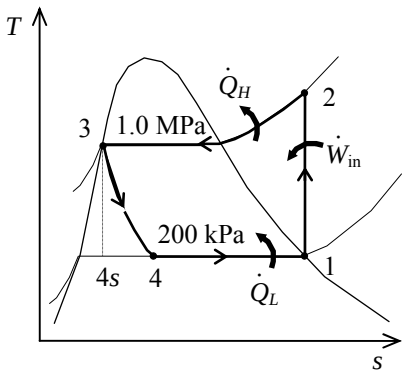
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-11, A-12, and A-13),

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 200\ \text{kPa}} = 244.46\ \text{kJ/kg} \\ s_1 = s_{g\ @\ 200\ \text{kPa}} = 0.93773\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1000\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 277.98\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1000\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\ @\ 1000\ \text{kPa}} = 107.32\ \text{kJ/kg}$$

$$h_4 \cong h_3 = 107.32\ \text{kJ/kg} \quad \text{(throttling)}$$



The mass flow rate of the refrigerant is determined from

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) \longrightarrow \dot{m} = \frac{\dot{W}_{\text{in}}}{h_2 - h_1} = \frac{6\ \text{kJ/s}}{(277.98 - 244.46)\ \text{kJ/kg}} = 0.179\ \text{kg/s}$$

Then the rate of heat supplied to the evaporator is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.179\ \text{kg/s})(244.46 - 107.32)\ \text{kJ/kg} = \mathbf{24.5\ kW}$$

The COP of the heat pump is determined from its definition,

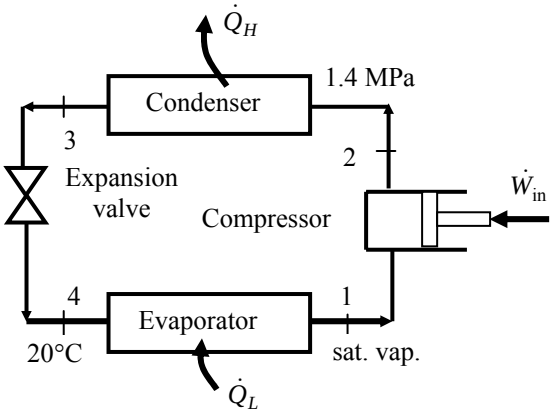
$$\text{COP}_{\text{HP}} = \frac{q_H}{w_{\text{in}}} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{277.98 - 107.32}{277.98 - 244.46} = \mathbf{5.09}$$

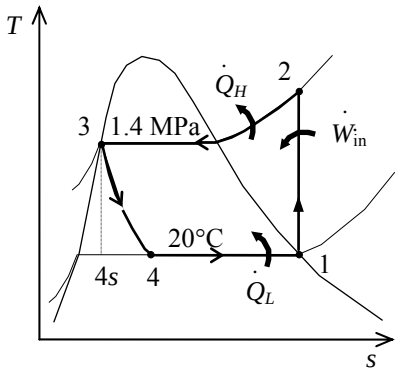
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11-47 A heat pump operating on the ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat transfer to the heated space and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure.





(b) The properties as given in the problem statement are

$$h_4 = h_3 = h_{f\,@\,1400\,\text{kPa}} = 127.2\ \text{kJ/kg}$$

$$h_1 = h_{g\,@\,20°\text{C}} = 261.6\ \text{kJ/kg}.$$

The enthalpy at the compressor exit is

$$w_{in} = h_2 - h_1 \longrightarrow h_2 = h_1 + w_{in} = 261.6\ \text{kJ/kg} + 20\ \text{kJ/kg} = 281.6\ \text{kJ/kg}$$

The mass flow rate through the cycle is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) \longrightarrow \dot{m} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{2.7\ \text{kJ/s}}{(261.6 - 127.2)\ \text{kJ/kg}} = 0.02009\ \text{kg/s}$$

The rate of heat transfer to the heated space is

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.02009\ \text{kg/s})(281.6 - 127.2)\ \text{kJ/kg} = \mathbf{3.10\ kW}$$

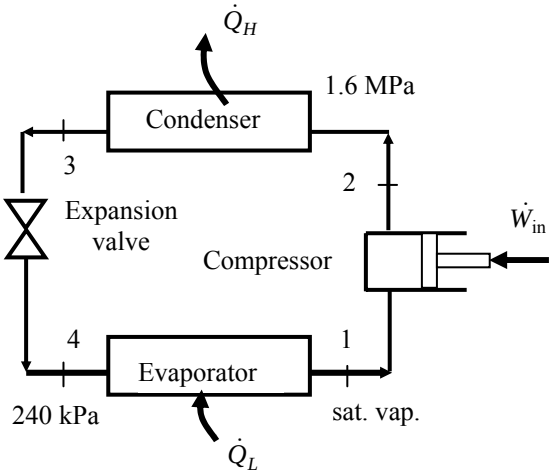
(c) The COP of the heat pump is

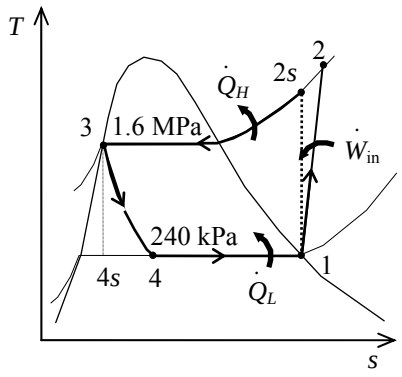
$$\text{COP}_{HP} = \frac{\dot{Q}_H}{\dot{W}_{in}} = \frac{\dot{Q}_H}{\dot{m}w_{in}} = \frac{3.10\ \text{kW}}{(0.02009\ \text{kg/s})(20\ \text{kJ/kg})} = \mathbf{7.72}$$

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11-48 A heat pump vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The hardware and the *T*-*s* diagram for this heat pump are to be sketched. The power input and the COP are to be determined.

Analysis (a) In a normal vapor-compression refrigeration cycle, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure.





(b) The properties as given in the problem statement are

$h_4 = h_3 = h_{f\,@\,1600\,\text{kPa}} = 134$ kJ/kg

$h_1 = h_{g\,@\,240\,\text{kPa}} = 244$ kJ/kg.

$h_{2s} = 285$ kJ/kg.

From the definition of isentropic efficiency for a compressor,

$$\eta_{\text{Comp}} = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_{\text{Comp}}} = 244 + \frac{285 - 244}{0.85} = 292.2\ \text{kJ/kg}$$

Then the work input to the compressor is

$$w_{\text{in}} = h_2 - h_1 = 292.2 - 244 = 48.2\ \text{kJ/kg}$$

The mass flow rate through the cycle is

$$\dot{Q}_H = \dot{m}(h_2 - h_3) \longrightarrow \dot{m} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{(2\ \text{ton})\left(\frac{211/60\ \text{kJ/s}}{1\ \text{ton}}\right)}{(292.2 - 134)\ \text{kJ/kg}} = 0.04446\ \text{kg/s}$$

Then the power input to the compressor is

$$\dot{W}_{\text{in}} = \dot{m} w_{\text{in}} = (0.04446\ \text{kg/s})(48.2\ \text{kJ/kg}) = \mathbf{2.14\ kW}$$

The COP of the heat pump is

$$\text{COP}_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{(2\ \text{ton})\left(\frac{211/60\ \text{kJ/s}}{1\ \text{ton}}\right)}{2.14\ \text{kW}} = \mathbf{3.29}$$

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11-49 A geothermal heat pump is considered. The degrees of subcooling done on the refrigerant in the condenser, the mass flow rate of the refrigerant, the heating load, the COP of the heat pump, the minimum power input are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

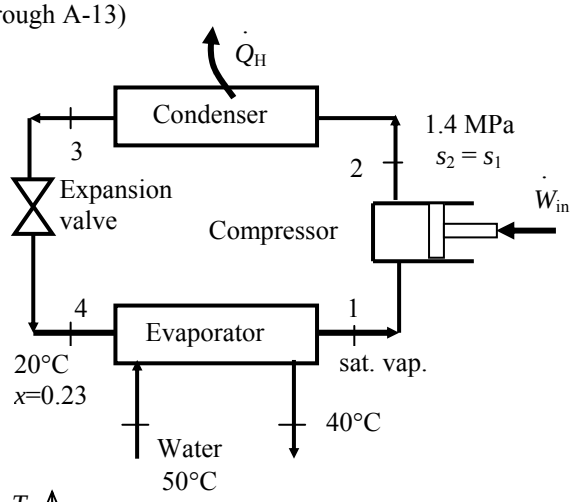
Analysis (*a*) From the refrigerant-134a tables (Tables A-11 through A-13)

$$\left.\begin{aligned} T_4 &= 20°\text{C} \\ x_4 &= 0.23 \end{aligned}\right\} \begin{aligned} P_4 &= 572.1\,\text{kPa} \\ h_4 &= 121.24\,\text{kJ/kg} \end{aligned}$$

$$h_3 = h_4$$

$$\left.\begin{aligned} P_1 &= 572.1\,\text{kPa} \\ x_1 &= 1\,(\text{sat. vap.}) \end{aligned}\right\} \begin{aligned} h_1 &= 261.59\,\text{kJ/kg} \\ s_1 &= 0.9223\,\text{kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 1400\,\text{kPa} \\ s_2 &= s_1 \end{aligned}\right\} h_2 = 280.00\,\text{kJ/kg}$$



From the steam tables (Table A-4)

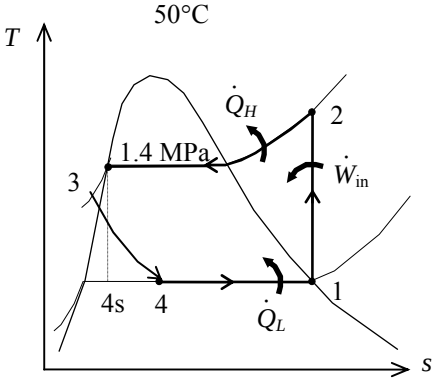
$$h_{w1} = h_{f\,@\,50°\text{C}} = 209.34\,\text{kJ/kg}$$

$$h_{w2} = h_{f\,@\,40°\text{C}} = 167.53\,\text{kJ/kg}$$

The saturation temperature at the condenser pressure of 1400 kPa and the actual temperature at the condenser outlet are

$$T_{\text{sat}\,@\,1400\,\text{kPa}} = 52.40°\text{C}$$

$$\left.\begin{aligned} P_3 &= 1400\,\text{kPa} \\ h_3 &= 121.24\,\text{kJ/kg} \end{aligned}\right\} T_3 = 48.59°\text{C} \ \text{(from EES)}$$



Then, the degrees of subcooling is

$$\Delta T_{\text{subcool}} = T_{\text{sat}} - T_3 = 52.40 - 48.59 = \mathbf{3.81°C}$$

(*b*) The rate of heat absorbed from the geothermal water in the evaporator is

$$\dot{Q}_L = \dot{m}_w (h_{w1} - h_{w2}) = (0.065\,\text{kg/s})(209.34 - 167.53)\text{kJ/kg} = 2.718\,\text{kW}$$

This heat is absorbed by the refrigerant in the evaporator

$$\dot{m}_R = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{2.718\,\text{kW}}{(261.59 - 121.24)\text{kJ/kg}} = \mathbf{0.01936\ kg/s}$$

(*c*) The power input to the compressor, the heating load and the COP are

$$\dot{W}_{\text{in}} = \dot{m}_R (h_2 - h_1) + \dot{Q}_{\text{out}} = (0.01936\,\text{kg/s})(280.00 - 261.59)\text{kJ/kg} = 0.6564\,\text{kW}$$

$$\dot{Q}_H = \dot{m}_R (h_2 - h_3) = (0.01936\,\text{kg/s})(280.00 - 121.24)\text{kJ/kg} = \mathbf{3.074\ kW}$$

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{in}} = \frac{3.074\,\text{kW}}{0.6564\,\text{kW}} = \mathbf{4.68}$$

(*d*) The reversible COP of the cycle is

$$\text{COP}_{\text{rev}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (25 + 273)/(50 + 273)} = 12.92$$

The corresponding minimum power input is

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{rev}}} = \frac{3.074\,\text{kW}}{12.92} = \mathbf{0.238\ kW}$$

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11-50 An actual heat pump cycle with R-134a as the refrigerant is considered. The isentropic efficiency of the compressor, the rate of heat supplied to the heated room, the COP of the heat pump, and the COP and the rate of heat supplied to the heated room if this heat pump operated on the ideal vapor-compression cycle between the same pressure limits are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The properties of refrigerant-134a are (Tables A-11 through A-13)

$$\left.\begin{matrix} P_2 = 800\ \text{kPa} \\ T_2 = 55°\text{C} \end{matrix}\right\} h_2 = 291.76\ \text{kJ/kg}$$

$$T_3 = T_{\text{sat@750 kPa}} = 29.06°\text{C}$$

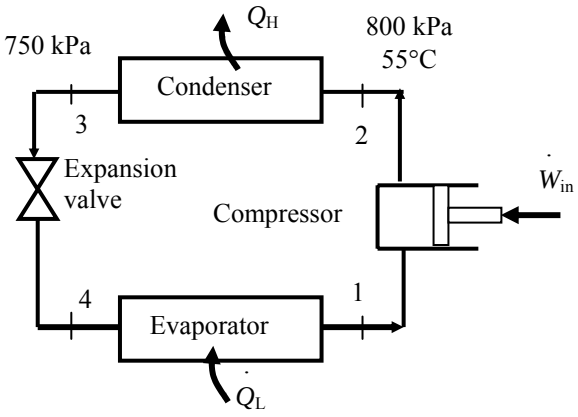
$$\left.\begin{matrix} P_3 = 750\ \text{kPa} \\ T_3 = (29.06-3)°\text{C} \end{matrix}\right\} h_3 = 87.91\ \text{kJ/kg}$$

$$h_4 = h_3 = 87.91\ \text{kJ/kg}$$

$$T_{\text{sat@200 kPa}} = -10.09°\text{C}$$

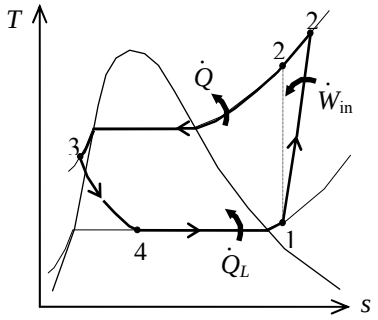
$$\left.\begin{matrix} P_1 = 200\ \text{kPa} \\ T_1 = (-10.09+4)°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = 247.87\ \text{kJ/kg} \\ s_1 = 0.9506\ \text{kJ/kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 800\ \text{kPa} \\ s_2 = s_1 \end{matrix}\right\} h_{2s} = 277.26$$



The isentropic efficiency of the compressor is

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{277.26 - 247.87}{291.76 - 247.87} = \mathbf{0.670}$$



(*b*) The rate of heat supplied to the room is

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.018\ \text{kg/s})(291.76 - 87.91)\text{kJ/kg} = \mathbf{3.67\ kW}$$

(*c*) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.018\ \text{kg/s})(291.76 - 247.87)\text{kJ/kg} = 0.790\ \text{kW}$$

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.67}{0.790} = \mathbf{4.64}$$

(*d*) The ideal vapor-compression cycle analysis of the cycle is as follows:

$$h_1 = h_{g@\ 200\ \text{kPa}} = 244.46\ \text{kJ/kg}$$

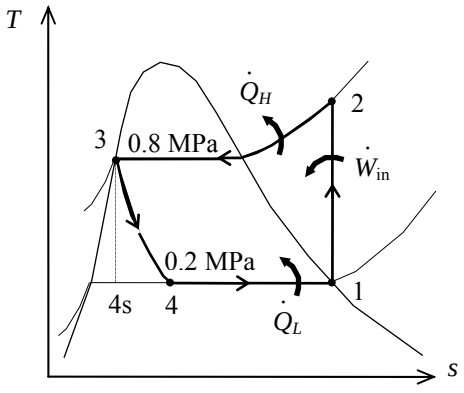
$$s_1 = s_{g@\ 200\ \text{kPa}} = 0.9377\ \text{kJ/kg.K}$$

$$\left.\begin{matrix} P_2 = 800\ \text{kPa} \\ s_2 = s_1 \end{matrix}\right\} h_2 = 273.25\ \text{kJ/kg}$$

$$h_3 = h_{f@800\ \text{kPa}} = 95.47\ \text{kJ/kg}$$

$$h_4 = h_3$$

$$\text{COP} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{273.25 - 95.47}{273.25 - 244.46} = \mathbf{6.18}$$



$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.018\ \text{kg/s})(273.25 - 95.47)\text{kJ/kg} = \mathbf{3.20\ kW}$$

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Innovative Refrigeration Systems

11-51C Performing the refrigeration in stages is called cascade refrigeration. In cascade refrigeration, two or more refrigeration cycles operate in series. Cascade refrigerators are more complex and expensive, but they have higher COP's, they can incorporate two or more different refrigerants, and they can achieve much lower temperatures.

11-52C Cascade refrigeration systems have higher COPs than the ordinary refrigeration systems operating between the same pressure limits.

11-53C The saturation pressure of refrigerant-134a at -32°C is 77 kPa, which is below the atmospheric pressure. In reality a pressure below this value should be used. Therefore, a cascade refrigeration system with a different refrigerant at the bottoming cycle is recommended in this case.

11-54C We would favor the two-stage compression refrigeration system with a flash chamber since it is simpler, cheaper, and has better heat transfer characteristics.

11-55C Yes, by expanding the refrigerant in stages in several throttling devices.

11-56C To take advantage of the cooling effect by throttling from high pressures to low pressures.

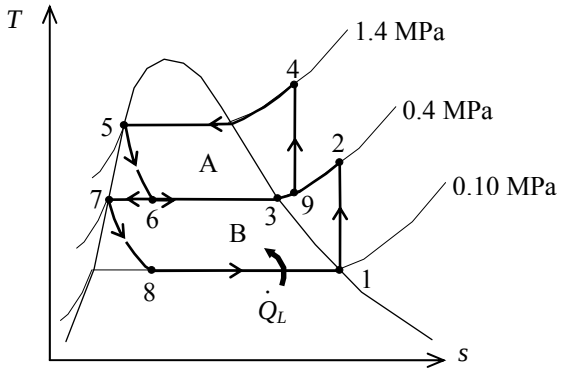
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11-57 A two-stage compression refrigeration system with refrigerant-134a as the working fluid is considered. The fraction of the refrigerant that evaporates as it is throttled to the flash chamber, the rate of heat removed from the refrigerated space, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The flash chamber is adiabatic.

Analysis (*a*) The enthalpies of the refrigerant at several states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 234.44 \text{ kJ/kg}, \quad h_2 = 262.68 \text{ kJ/kg}$$
$$h_3 = 255.55 \text{ kJ/kg},$$
$$h_5 = 127.22 \text{ kJ/kg}, \quad h_6 = 127.22 \text{ kJ/kg}$$
$$h_7 = 63.94 \text{ kJ/kg}, \quad h_8 = 63.94 \text{ kJ/kg}$$



The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6,

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{127.22 - 63.94}{191.62} = \mathbf{0.3303}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$
$$h_9 = (0.3303)(255.55) + (1 - 0.3303)(262.68) = 260.33 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 0.4 \text{ MPa} \\ h_9 = 260.33 \text{ kJ/kg} \end{array}\right\} s_9 = 0.9437 \text{ kJ/kg} \cdot \text{K}$$

also,

$$\left.\begin{array}{l} P_4 = 1.4 \text{ MPa} \\ s_4 = s_9 = 0.9437 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_4 = 287.07 \text{ kJ/kg}$$

Then the rate of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$\dot{m}_B = (1 - x_6)\dot{m}_A = (1 - 0.3303)(0.25 \text{ kg/s}) = 0.1674 \text{ kg/s}$$

$$\dot{Q}_L = \dot{m}_B (h_1 - h_8) = (0.1674 \text{ kg/s})(234.44 - 63.94) \text{ kJ/kg} = \mathbf{28.55\ kW}$$

$$\dot{W}_{\text{in}} = \dot{W}_{\text{compI,in}} + \dot{W}_{\text{compII,in}} = \dot{m}_A (h_4 - h_9) + \dot{m}_B (h_2 - h_1)$$
$$= (0.25 \text{ kg/s})(287.07 - 260.33) \text{ kJ/kg} + (0.1674 \text{ kg/s})(262.68 - 234.44) \text{ kJ/kg}$$
$$= 11.41 \text{ kW}$$

(*c*) The coefficient of performance is determined from

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{28.55 \text{ kW}}{11.41 \text{ kW}} = \mathbf{2.50}$$

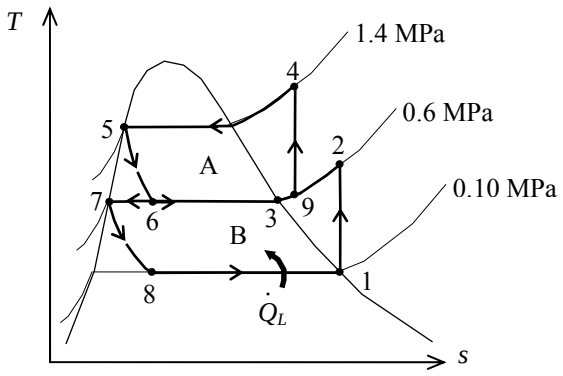
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11-58 A two-stage compression refrigeration system with refrigerant-134a as the working fluid is considered. The fraction of the refrigerant that evaporates as it is throttled to the flash chamber, the rate of heat removed from the refrigerated space, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The flash chamber is adiabatic.

Analysis (*a*) The enthalpies of the refrigerant at several states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 234.44 \text{ kJ/kg}, \quad h_2 = 271.40 \text{ kJ/kg}$$
$$h_3 = 262.40 \text{ kJ/kg},$$
$$h_5 = 127.22 \text{ kJ/kg}, \quad h_6 = 127.22 \text{ kJ/kg}$$
$$h_7 = 81.51 \text{ kJ/kg}, \quad h_8 = 81.51 \text{ kJ/kg}$$



The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6,

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{127.22 - 81.51}{180.90} = \mathbf{0.2527}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$
$$h_9 = (0.2527)(262.40) + (1 - 0.2527)(271.40) = 269.13 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 0.6 \text{ MPa} \\ h_9 = 269.13 \text{ kJ/kg} \end{array}\right\} s_9 = 0.9443 \text{ kJ/kg} \cdot \text{K}$$

also,

$$\left.\begin{array}{l} P_4 = 1.4 \text{ MPa} \\ s_4 = s_9 = 0.9443 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_4 = 287.28 \text{ kJ/kg}$$

Then the rate of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$\dot{m}_B = (1 - x_6)\dot{m}_A = (1 - 0.2527)(0.25 \text{ kg/s}) = 0.1868 \text{ kg/s}$$

$$\dot{Q}_L = \dot{m}_B (h_1 - h_8) = (0.1868 \text{ kg/s})(234.44 - 81.51) \text{ kJ/kg} = \mathbf{28.57 \ kW}$$

$$\dot{W}_{\text{in}} = \dot{W}_{\text{compI,in}} + \dot{W}_{\text{compII,in}} = \dot{m}_A (h_4 - h_9) + \dot{m}_B (h_2 - h_1)$$
$$= (0.25 \text{ kg/s})(287.28 - 269.13) \text{ kJ/kg} + (0.1868 \text{ kg/s})(271.40 - 234.44) \text{ kJ/kg}$$
$$= 11.44 \text{ kW}$$

(*c*) The coefficient of performance is determined from

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{28.57 \text{ kW}}{11.44 \text{ kW}} = \mathbf{2.50}$$

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11-59 Problem 11-57 is reconsidered. The effects of the various refrigerants in EES data bank for compressor efficiencies of 80, 90, and 100 percent is to be investigated.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
Fluid$='R134a'
"Input Data"
P[1]=100 [kPa]
 P[4] = 1400 [kPa]
P[6]=400 [kPa]
"Eta_comp =1.0"
m_dot_A=0.25 [kg/s]

"High Pressure Compressor A"
P[9]=P[6]
h4s=enthalpy(Fluid$,P=P[4],s=s[9]) "State 4s is the isentropic value of state 4"
h[9]+w_compAs=h4s "energy balance on isentropic compressor"

w_compA=w_compAs/Eta_comp"definition of compressor isentropic efficiency"
h[9]+w_compA=h[4] "energy balance on real compressor-assumed adiabatic"
s[4]=entropy(Fluid$,h=h[4],P=P[4])  "properties for state 4"
T[4]=temperature(Fluid$,h=h[4],P=P[4])
W_dot_compA=m_dot_A*w_compA

"Condenser"
P[5]=P[4] "neglect pressure drops across condenser"
T[5]=temperature(Fluid$,P=P[5],x=0) "properties for state 5, assumes sat. liq. at cond. exit"
h[5]=enthalpy(Fluid$,T=T[5],x=0) "properties for state 5"
s[5]=entropy(Fluid$,T=T[5],x=0)
h[4]=q_H+h[5] "energy balance on condenser"
Q_dot_H = m_dot_A*q_H

"Throttle Valve A"
h[6]=h[5] "energy balance on throttle - isenthalpic"
x6=quality(Fluid$,h=h[6],P=P[6]) "properties for state 6"
s[6]=entropy(Fluid$,h=h[6],P=P[6])
T[6]=temperature(Fluid$,h=h[6],P=P[6])

"Flash Chamber"
m_dot_B = (1-x6) * m_dot_A
P[7] = P[6]
h[7]=enthalpy(Fluid$, P=P[7], x=0)
s[7]=entropy(Fluid$,h=h[7],P=P[7])
T[7]=temperature(Fluid$,h=h[7],P=P[7])

"Mixing Chamber"
x6*m_dot_A*h[3] + m_dot_B*h[2] =(x6* m_dot_A + m_dot_B)*h[9]
P[3] = P[6]
h[3]=enthalpy(Fluid$, P=P[3], x=1) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=1)
T[3]=temperature(Fluid$,P=P[3],x=x1)
s[9]=entropy(Fluid$,h=h[9],P=P[9]) "properties for state 9"
T[9]=temperature(Fluid$,h=h[9],P=P[9])

"Low Pressure Compressor B"
x1=1   "assume flow to compressor inlet to be saturated vapor"
h[1]=enthalpy(Fluid$,P=P[1],x=x1)  "properties for state 1"
T[1]=temperature(Fluid$,P=P[1], x=x1)
```

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```
s[1]=entropy(Fluid$,P=P[1],x=x1)
P[2]=P[6]
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) " state 2s is isentropic state at comp. exit"
h[1]+w_compBs=h2s "energy balance on isentropic compressor"

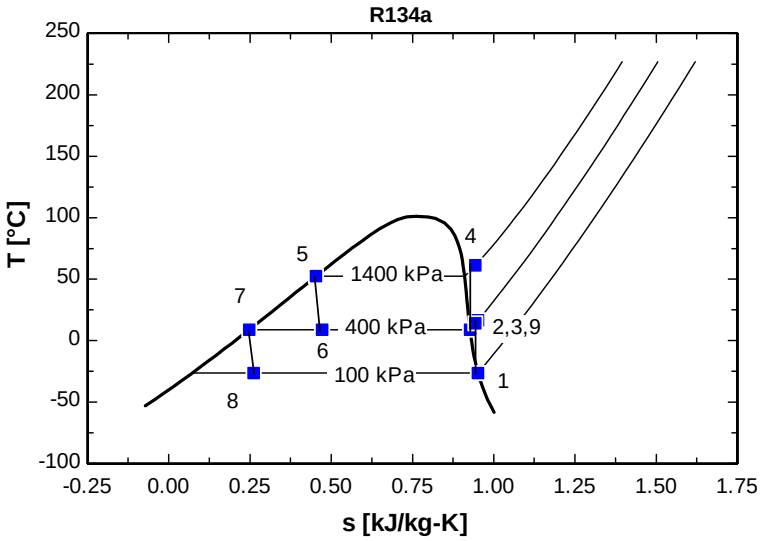
w_compB=w_compBs/Eta_comp"definition of compressor isentropic efficiency"
h[1]+w_compB=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_compB=m_dot_B*w_compB

"Throttle Valve B"
h[8]=h[7] "energy balance on throttle - isenthalpic"
x8=quality(Fluid$,h=h[8],P=P[8]) "properties for state 8"
s[8]=entropy(Fluid$,h=h[8],P=P[8])
T[8]=temperature(Fluid$,h=h[8],P=P[8])

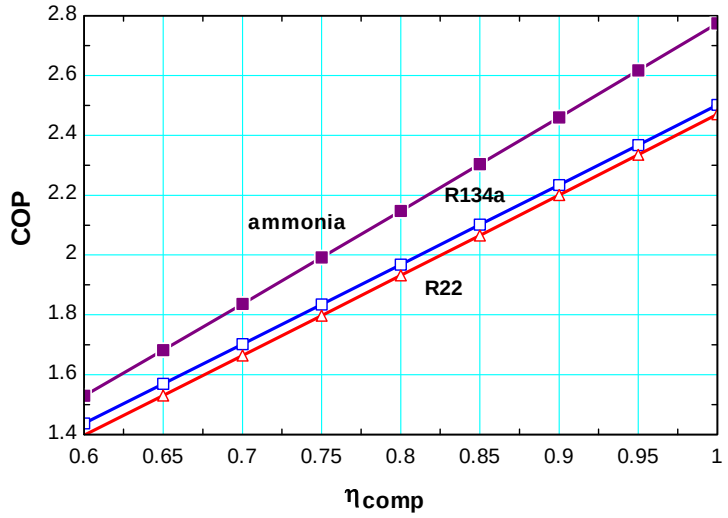
"Evaporator"
P[8]=P[1] "neglect pressure drop across evaporator"
q_L + h[8]=h[1] "energy balance on evaporator"
Q_dot_L=m_dot_B*q_L

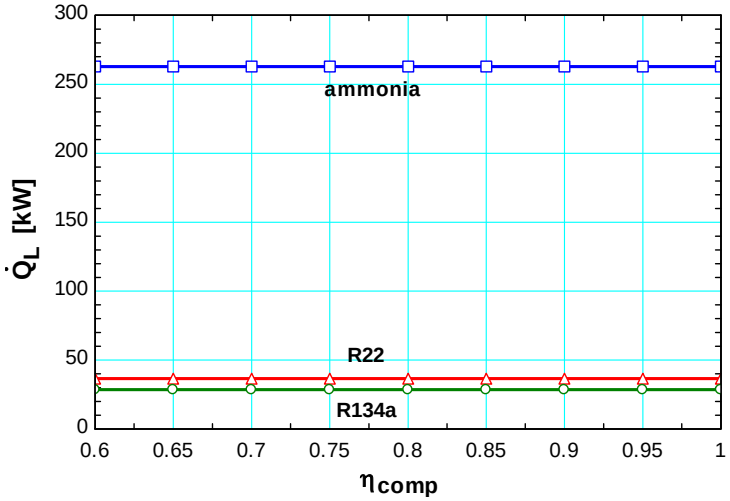
"Cycle Statistics"
W_dot_in_total = W_dot_compA + W_dot_compB
COP=Q_dot_L/W_dot_in_total "definition of COP"
```

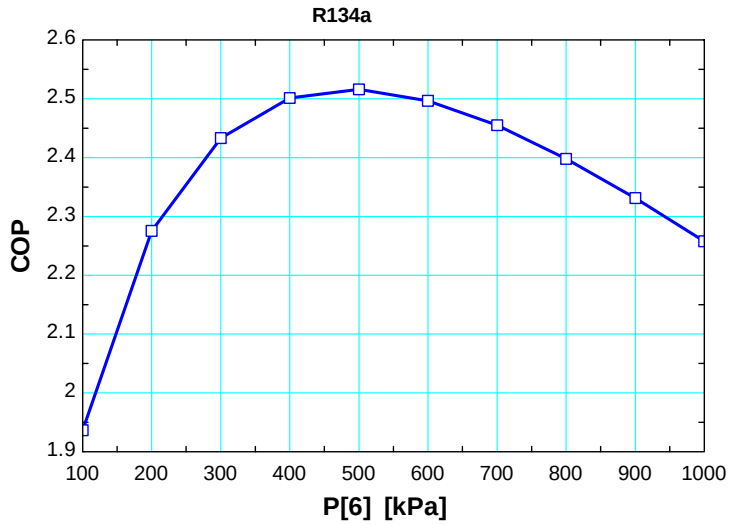
η_{comp}	Q_L [kW]	COP
0.6	28.55	1.438
0.65	28.55	1.57
0.7	28.55	1.702
0.75	28.55	1.835
0.8	28.55	1.968
0.85	28.55	2.101
0.9	28.55	2.234
0.95	28.55	2.368
1	28.55	2.501



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11-60 A two-stage cascade refrigeration cycle is considered. The mass flow rate of the refrigerant through the upper cycle, the rate of heat removal from the refrigerated space, and the COP of the refrigerator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The properties are to be obtained from the refrigerant tables (Tables A-11 through A-13):

$$h_1 = h_{g@\ 200\ \text{kPa}} = 244.46\ \text{kJ/kg}$$
$$s_1 = s_{g@\ 200\ \text{kPa}} = 0.9377\ \text{kJ/kg.K}$$

$$\left.\begin{matrix} P_2 = 500\ \text{kPa} \\ s_2 = s_1 \end{matrix}\right\} h_{2s} = 263.30\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.80 = \frac{263.30 - 244.46}{h_2 - 244.46} \longrightarrow h_2 = 268.01\ \text{kJ/kg}$$

$$h_3 = h_{f@\ 500\ \text{kPa}} = 73.33\ \text{kJ/kg}$$
$$h_4 = h_3 = 73.33\ \text{kJ/kg}$$

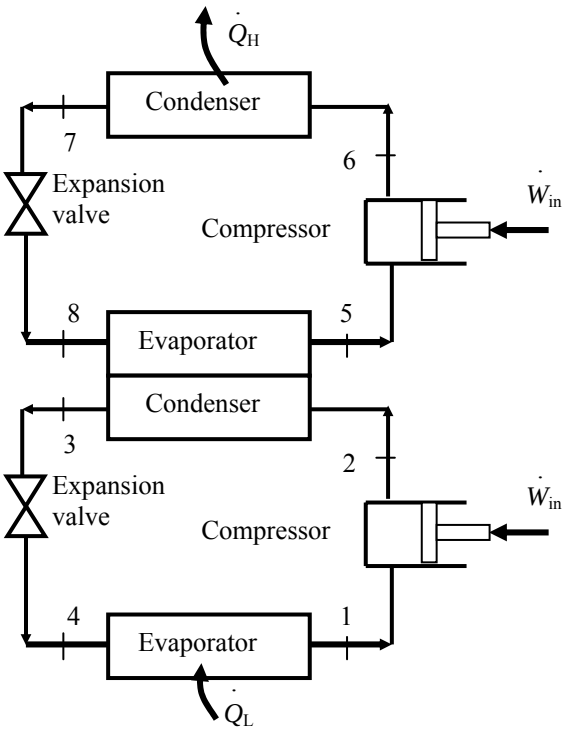
$$h_5 = h_{g@\ 400\ \text{kPa}} = 255.55\ \text{kJ/kg}$$
$$s_5 = s_{g@\ 400\ \text{kPa}} = 0.9269\ \text{kJ/kg.K}$$

$$\left.\begin{matrix} P_6 = 1200\ \text{kPa} \\ s_6 = s_5 \end{matrix}\right\} h_{6s} = 278.33\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{6s} - h_5}{h_6 - h_5}$$

$$0.80 = \frac{278.33 - 255.55}{h_6 - 255.55} \longrightarrow h_6 = 284.02\ \text{kJ/kg}$$

$$h_7 = h_{f@1200\ \text{kPa}} = 117.77\ \text{kJ/kg}$$
$$h_8 = h_7 = 117.77\ \text{kJ/kg}$$



The mass flow rate of the refrigerant through the upper cycle is determined from an energy balance on the heat exchanger

$$\dot{m}_A(h_5 - h_8) = \dot{m}_B(h_2 - h_3)$$

$$\dot{m}_A(255.55 - 117.77)\text{kJ/kg} = (0.15\ \text{kg/s})(268.01 - 73.33)\text{kJ/kg} \longrightarrow \dot{m}_A = \mathbf{0.212\ kg/s}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}_B(h_1 - h_4) = (0.15\ \text{kg/s})(244.46 - 73.33)\text{kJ/kg} = \mathbf{25.67\ kW}$$

(*c*) The power input and the COP are

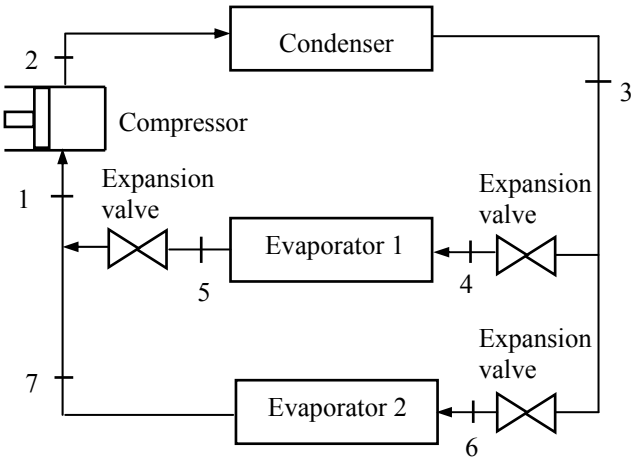
$$\dot{W}_{\text{in}} = \dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1)$$
$$= (0.15\ \text{kg/s})(284.02 - 255.55)\text{kJ/kg} + (0.212\ \text{kg/s})(268.01 - 244.46)\text{kJ/kg} = 9.566\ \text{kW}$$

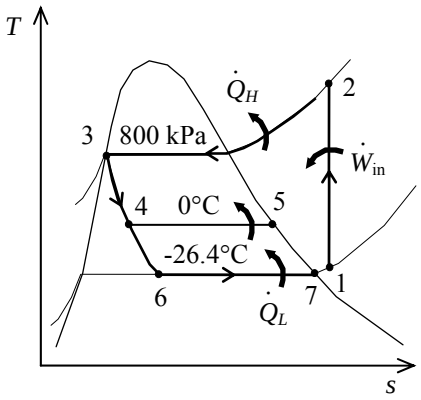
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{25.67}{9.566} = \mathbf{2.68}$$

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11-61 A two-evaporator compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The cooling rate of the high-temperature evaporator, the power required by the compressor, and the COP of the system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.





Analysis From the refrigerant tables (Tables A-11, A-12, and A-13),

$$\left.\begin{array}{l} P_3 = 800\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \ h_3 = h_{f\ @\ 800\ \text{kPa}} = 95.47\ \text{kJ/kg}$$

$$h_4 = h_6 \cong h_3 = 95.47\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_5 = 0°\text{C} \\ \text{sat. vapor} \end{array}\right\} \ h_5 = h_{g\ @\ 0°\text{C}} = 250.45\ \text{kJ/kg}$$

$$\left.\begin{array}{l} T_7 = -26.4°\text{C} \\ \text{sat. vapor} \end{array}\right\} \ h_7 = h_{g\ @\ -26.4°\text{C}} = 234.44\ \text{kJ/kg}$$

The mass flow rate through the low-temperature evaporator is found by

$$\dot{Q}_L = \dot{m}_2 (h_7 - h_6) \longrightarrow \dot{m}_2 = \frac{\dot{Q}_L}{h_7 - h_6} = \frac{8\ \text{kJ/s}}{(234.44 - 95.47)\ \text{kJ/kg}} = 0.05757\ \text{kg/s}$$

The mass flow rate through the warmer evaporator is then

$$\dot{m}_1 = \dot{m} - \dot{m}_2 = 0.1 - 0.05757 = 0.04243\ \text{kg/s}$$

Applying an energy balance to the point in the system where the two evaporator streams are recombined gives

$$\dot{m}_1 h_5 + \dot{m}_2 h_7 = \dot{m} h_1 \longrightarrow h_1 = \frac{\dot{m}_1 h_5 + \dot{m}_2 h_7}{\dot{m}} = \frac{(0.04243)(250.45) + (0.05757)(234.44)}{0.1} = 241.23\ \text{kJ/kg}$$

Then,

$$\left.\begin{array}{l} P_1 = P_{\text{sat}\ @\ -26.4°\text{C}} \cong 100\ \text{kPa} \\ h_1 = 241.23\ \text{kJ/kg} \end{array}\right\} \ s_1 = 0.9789\ \text{kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} \ h_2 = 286.26\ \text{kJ/kg}$$

The cooling rate of the high-temperature evaporator is

$$\dot{Q}_L = \dot{m}_1 (h_5 - h_4) = (0.04243\ \text{kg/s})(250.45 - 95.47)\ \text{kJ/kg} = \mathbf{6.58\ kW}$$

The power input to the compressor is

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.1\ \text{kg/s})(286.26 - 241.23)\ \text{kJ/kg} = \mathbf{4.50\ kW}$$

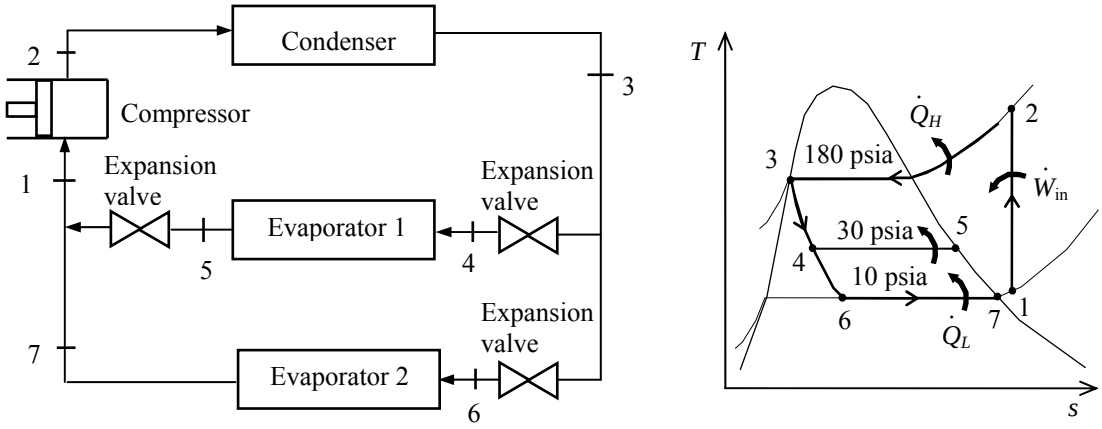
The COP of this refrigeration system is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{(8 + 6.58)\ \text{kW}}{4.50\ \text{kW}} = \mathbf{3.24}$$

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11-62E A two-evaporator compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The power required by the compressor and the COP of the system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From the refrigerant tables (Tables A-11E, A-12E, and A-13E),

$$\left.\begin{array}{l} P_3 = 180 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\,@\,180\text{ psia}} = 51.50 \text{ Btu/lbm}$$

$$h_4 = h_6 \cong h_3 = 51.50 \text{ Btu/lbm} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} P_5 = 30 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \quad h_5 = h_{g\,@\,30\text{ psia}} = 105.32 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_7 = 10 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \quad h_7 = h_{g\,@\,10\text{ psia}} = 98.68 \text{ Btu/lbm}$$

The mass flow rates through the high-temperature and low-temperature evaporators are found by

$$\dot{Q}_{L,1} = \dot{m}_1(h_5 - h_4) \longrightarrow \dot{m}_1 = \frac{\dot{Q}_{L,1}}{h_5 - h_4} = \frac{9000 \text{ Btu/h}}{(105.32 - 51.50) \text{ Btu/lbm}} = 167.2 \text{ lbm/h}$$

$$\dot{Q}_{L,2} = \dot{m}_2(h_7 - h_6) \longrightarrow \dot{m}_2 = \frac{\dot{Q}_{L,2}}{h_7 - h_6} = \frac{24{,}000 \text{ Btu/h}}{(98.68 - 51.50) \text{ Btu/lbm}} = 508.6 \text{ lbm/h}$$

Applying an energy balance to the point in the system where the two evaporator streams are recombined gives

$$\dot{m}_1 h_5 + \dot{m}_2 h_7 = (\dot{m}_1 + \dot{m}_2) h_1 \longrightarrow h_1 = \frac{\dot{m}_1 h_5 + \dot{m}_2 h_7}{\dot{m}_1 + \dot{m}_2} = \frac{(167.2)(105.32) + (508.6)(98.68)}{167.2 + 508.6} = 100.33 \text{ Btu/lbm}$$

Then,

$$\left.\begin{array}{l} P_1 = 10 \text{ psia} \\ h_1 = 100.33 \text{ Btu/lbm} \end{array}\right\} \quad s_1 = 0.2333 \text{ Btu/lbm} \cdot \text{R}$$

$$\left.\begin{array}{l} P_2 = 180 \text{ psia} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 127.05 \text{ Btu/lbm}$$

The power input to the compressor is

$$\dot{W}_{in} = (\dot{m}_1 + \dot{m}_2)(h_2 - h_1) = (167.2 + 508.6) \text{ lbm/h}(127.05 - 100.33) \text{ Btu/lbm}\left(\frac{1 \text{ kW}}{3412.14 \text{ Btu/h}}\right) = \mathbf{5.29\ kW}$$

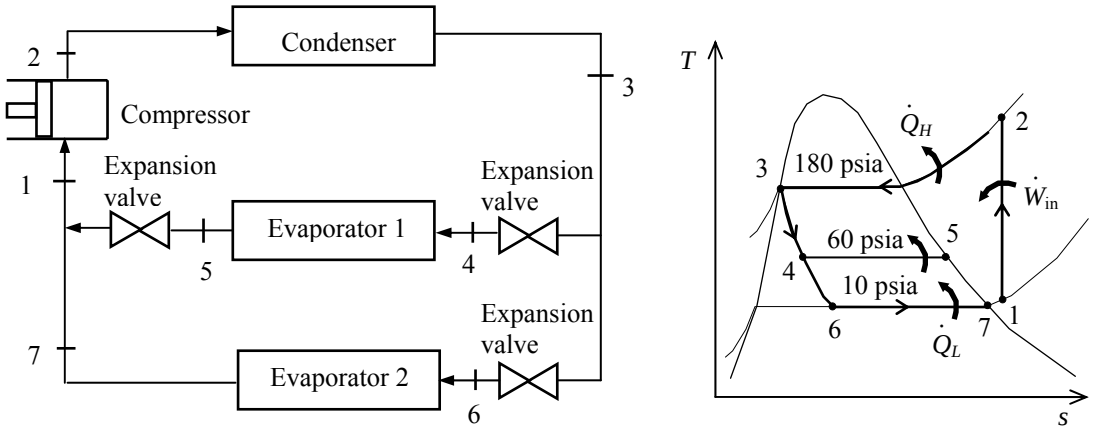
The COP of this refrigeration system is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{(24{,}000 + 9000) \text{ Btu/h}}{5.29 \text{ kW}}\left(\frac{1 \text{ kW}}{3412.14 \text{ Btu/h}}\right) = \mathbf{1.83}$$

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11-63E A two-evaporator compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The power required by the compressor and the COP of the system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From the refrigerant tables (Tables A-11E, A-12E, and A-13E),

$$\left.\begin{array}{l} P_3 = 180 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\,@\,180\text{ psia}} = 51.50 \text{ Btu/lbm}$$

$$h_4 = h_6 \cong h_3 = 51.50 \text{ Btu/lbm} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} P_5 = 60 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \quad h_5 = h_{g\,@\,60\text{ psia}} = 110.11 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_7 = 10 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \quad h_7 = h_{g\,@\,10\text{ psia}} = 98.68 \text{ Btu/lbm}$$

The mass flow rates through the high-temperature and low-temperature evaporators are found by

$$\dot{Q}_{L,1} = \dot{m}_1(h_5 - h_4) \longrightarrow \dot{m}_1 = \frac{\dot{Q}_{L,1}}{h_5 - h_4} = \frac{30{,}000 \text{ Btu/h}}{(110.11 - 51.50) \text{ Btu/lbm}} = 511.8 \text{ lbm/h}$$

$$\dot{Q}_{L,2} = \dot{m}_2(h_7 - h_6) \longrightarrow \dot{m}_2 = \frac{\dot{Q}_{L,2}}{h_7 - h_6} = \frac{24{,}000 \text{ Btu/h}}{(98.68 - 51.50) \text{ Btu/lbm}} = 508.6 \text{ lbm/h}$$

Applying an energy balance to the point in the system where the two evaporator streams are recombined gives

$$\dot{m}_1 h_5 + \dot{m}_2 h_7 = (\dot{m}_1 + \dot{m}_2) h_1 \longrightarrow h_1 = \frac{\dot{m}_1 h_5 + \dot{m}_2 h_7}{\dot{m}_1 + \dot{m}_2} = \frac{(511.8)(110.11) + (508.6)(98.68)}{511.8 + 508.6} = 104.41 \text{ Btu/lbm}$$

Then,

$$\left.\begin{array}{l} P_1 = 10 \text{ psia} \\ h_1 = 104.41 \text{ Btu/lbm} \end{array}\right\} \quad s_1 = 0.2423 \text{ Btu/lbm} \cdot \text{R}$$

$$\left.\begin{array}{l} P_2 = 180 \text{ psia} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 132.69 \text{ Btu/lbm}$$

The power input to the compressor is

$$\dot{W}_{\text{in}} = (\dot{m}_1 + \dot{m}_2)(h_2 - h_1) = (511.8 + 508.6) \text{ lbm/h}(132.69 - 104.41) \text{ Btu/lbm}\left(\frac{1 \text{ kW}}{3412.14 \text{ Btu/h}}\right) = \mathbf{8.46\ kW}$$

The COP of this refrigeration system is determined from its definition,

$$\text{COP}_{\text{R}} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{(24{,}000 + 30{,}000) \text{ Btu/h}}{8.46 \text{ kW}}\left(\frac{1 \text{ kW}}{3412.14 \text{ Btu/h}}\right) = \mathbf{1.87}$$

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11-64 A two-stage cascade refrigeration system is considered. Each stage operates on the ideal vapor-compression cycle with upper cycle using water and lower cycle using refrigerant-134a as the working fluids. The mass flow rate of R-134a and water in their respective cycles and the overall COP of this system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The heat exchanger is adiabatic.

Analysis From the water and refrigerant tables (Tables A-4, A-5, A-6, A-11, A-12, and A-13),

$$\left.\begin{array}{l} T_1 = 5°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,5°\text{C}} = 2510.1\ \text{kJ/kg} \\ s_1 = s_{g\,@\,5°\text{C}} = 9.0249\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.6\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 5083.4\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.6\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\,@\,1.6\,\text{MPa}} = 858.44\ \text{kJ/kg}$$

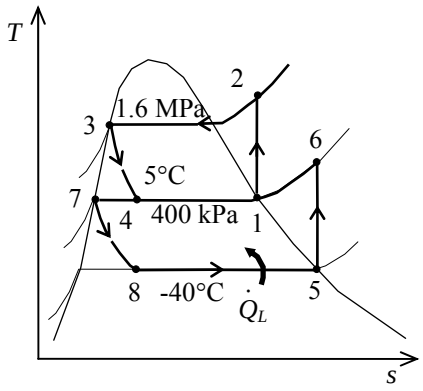
$$h_4 \cong h_3 = 858.44\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_5 = -40°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_5 = h_{g\,@\,-40°\text{C}} = 225.86\ \text{kJ/kg} \\ s_5 = s_{g\,@\,-40°\text{C}} = 0.96866\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 400\ \text{kPa} \\ s_6 = s_5 \end{array}\right\} \quad h_6 = 267.59\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 400\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \quad h_7 = h_{f\,@\,400\,\text{kPa}} = 63.94\ \text{kJ/kg}$$

$$h_8 \cong h_7 = 63.94\ \text{kJ/kg} \quad \text{(throttling)}$$



The mass flow rate of R-134a is determined from

$$\dot{Q}_L = \dot{m}_R(h_5 - h_8) \longrightarrow \dot{m}_R = \frac{\dot{Q}_L}{h_5 - h_8} = \frac{20\ \text{kJ/s}}{(225.86 - 63.94)\ \text{kJ/kg}} = \mathbf{0.1235\ kg/s}$$

An energy balance on the heat exchanger gives the mass flow rate of water

$$\dot{m}_R(h_6 - h_7) = \dot{m}_w(h_1 - h_4)$$

$$\longrightarrow \dot{m}_w = \dot{m}_R \frac{h_6 - h_7}{h_1 - h_4} = (0.1235\ \text{kg/s})\frac{267.59 - 63.94}{2510.1 - 858.44} = \mathbf{0.01523\ kg/s}$$

The total power input to the compressors is

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{m}_R(h_6 - h_5) + \dot{m}_w(h_2 - h_1) \\ &= (0.1235\ \text{kg/s})(267.59 - 225.86)\ \text{kJ/kg} + (0.01523\ \text{kg/s})(5083.4 - 2510.1)\ \text{kJ/kg} \\ &= 44.35\ \text{kJ/s} \end{aligned}$$

The COP of this refrigeration system is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{20\ \text{kJ/s}}{44.35\ \text{kJ/s}} = \mathbf{0.451}$$

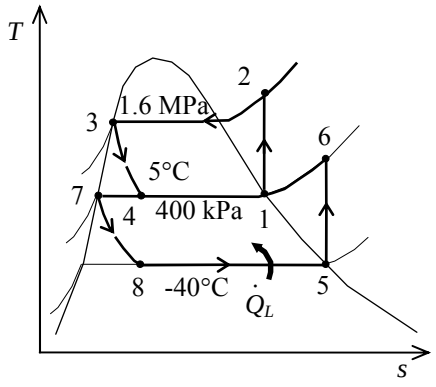
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11-65 A two-stage vapor-compression refrigeration system with refrigerant-134a as the working fluid is considered. The process with the greatest exergy destruction is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Prob. 11-55 and the water and refrigerant tables (Tables A-4, A-5, A-6, A-11, A-12, and A-13),

$$s_1 = s_2 = 9.0249\ \text{kJ/kg}\cdot\text{K}$$
$$s_3 = 2.3435\ \text{kJ/kg}\cdot\text{K}$$
$$s_4 = 3.0869\ \text{kJ/kg}\cdot\text{K}$$
$$s_5 = s_6 = 0.96866\ \text{kJ/kg}\cdot\text{K}$$
$$s_7 = 0.24757\ \text{kJ/kg}\cdot\text{K}$$
$$s_8 = 0.27423\ \text{kJ/kg}\cdot\text{K}$$
$$\dot{m}_R = 0.1235\ \text{kg/s}$$
$$\dot{m}_w = 0.01523\ \text{kg/s}$$
$$q_L = h_5 - h_8 = 161.92\ \text{kJ/kg}$$
$$q_H = h_2 - h_3 = 4225.0\ \text{kJ/kg}$$
$$T_L = -30°\text{C} = 243\ \text{K}$$
$$T_H = 30°\text{C} = 303\ \text{K}$$
$$T_0 = 30°\text{C} = 303\ \text{K}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$\dot{X}_{\text{destroyed},23} = \dot{m}_w T_0\left(s_3 - s_2 + \frac{q_H}{T_H}\right)$$
$$= (0.01523)(303\ \text{K})\left(2.3435 - 9.0249 + \frac{4225.0}{303}\right) = 33.52\ \text{kJ/s}$$
$$\dot{X}_{\text{destroyed},34} = \dot{m}_{\text{w}} T_0 (s_4 - s_3) = (0.01523)(303)(3.0869 - 2.3435) = 3.43\ \text{kJ/s}$$
$$\dot{X}_{\text{destroyed},78} = \dot{m}_{\text{R}} T_0 (s_8 - s_7) = (0.1235)(303)(0.27423 - 0.24757) = 0.996\ \text{kJ/s}$$
$$\dot{X}_{\text{destroyed},85} = \dot{m}_{\text{R}} T_0\left(s_5 - s_8 - \frac{q_L}{T_L}\right)$$
$$= (0.1235)(303)\left(0.96866 - 0.27423 - \frac{161.92}{243}\right) = 1.05\ \text{kJ/s}$$
$$\dot{X}_{\text{destroyed, heat exch}} = T_0\left[\dot{m}_{\text{w}}(s_1 - s_4) + \dot{m}_{\text{R}}(s_7 - s_6)\right]$$
$$= (303)\left[(0.01523)(9.0249 - 3.0869) + (0.1235)(0.24757 - 0.96866)\right] = 0.417\ \text{kJ/s}$$

For isentropic processes, the exergy destruction is zero:

$$\dot{X}_{\text{destroyed},12} = 0$$
$$\dot{X}_{\text{destroyed},56} = 0$$

Note that heat is absorbed from a reservoir at -30°C (243 K) and rejected to a reservoir at 30°C (303 K), which is also taken as the dead state temperature. Alternatively, one may use the standard 25°C (298 K) as the dead state temperature, and perform the calculations accordingly. The greatest exergy destruction occurs in the condenser.

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11-66 A two-stage cascade refrigeration cycle with a flash chamber with refrigerant-134a as the working fluid is considered. The mass flow rate of the refrigerant through the high-pressure compressor, the rate of refrigeration, the COP are to be determined. Also, the rate of refrigeration and the COP are to be determined if this refrigerator operated on a single-stage vapor-compression cycle under similar conditions.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) From the refrigerant-134a tables (Tables A-11 through A-13)

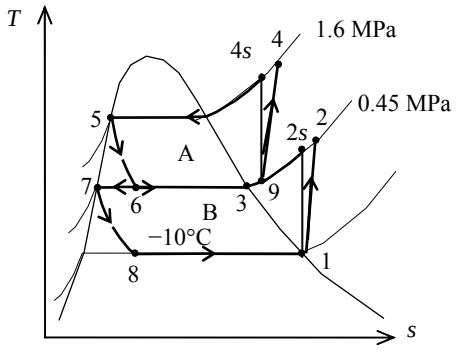
$$h_1 = h_{g@-10°C} = 244.51 \text{ kJ/kg}$$
$$s_1 = s_{g@-10°C} = 0.9377 \text{ kJ/kg.K}$$

$$\left.\begin{matrix} P_2 = 450 \text{ kPa} \\ s_2 = s_1 \end{matrix}\right\} h_{2s} = 261.07 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$
$$0.86 = \frac{261.07 - 244.51}{h_2 - 244.51} \longrightarrow h_2 = 263.76 \text{ kJ/kg}$$

$$h_3 = h_{g@450 \text{ kPa}} = 257.53 \text{ kJ/kg}$$
$$h_5 = h_{f@1600 \text{ kPa}} = 135.93 \text{ kJ/kg}$$
$$h_6 = h_5 = 135.93 \text{ kJ/kg}$$
$$h_7 = h_{f@450 \text{ kPa}} = 68.81 \text{ kJ/kg}$$
$$h_8 = h_7 = 68.81 \text{ kJ/kg}$$

$$\left.\begin{matrix} h_6 = 135.93 \text{ kJ/kg} \\ P_6 = 450 \text{ kPa} \end{matrix}\right\} x_6 = 0.3557$$



The mass flow rate of the refrigerant through the high pressure compressor is determined from a mass balance on the flash chamber

$$\dot{m} = \frac{\dot{m}_7}{1 - x_6} = \frac{0.11 \text{ kg/s}}{1 - 0.3557} = \mathbf{0.1707 \text{ kg/s}}$$

Also,

$$\dot{m}_3 = \dot{m} - \dot{m}_7 = 0.1707 - 0.11 = 0.06072 \text{ kg/s}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{m}h_9 = \dot{m}_7 h_2 + \dot{m}_3 h_3$$
$$(0.1707 \text{ kg/s})h_9 = (0.11 \text{ kg/s})(263.76 \text{ kJ/kg}) + (0.06072 \text{ kg/s})(257.53 \text{ kJ/kg}) \longrightarrow h_9 = 261.54 \text{ kJ/kg}$$

Then,

$$\left.\begin{matrix} P_9 = 450 \text{ kPa} \\ h_9 = 261.54 \text{ kJ/kg} \end{matrix}\right\} s_9 = 0.9393 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_4 = 1600 \text{ kPa} \\ s_4 = s_9 \end{matrix}\right\} h_{4s} = 288.41 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{4s} - h_9}{h_4 - h_9}$$
$$0.86 = \frac{288.41 - 261.54}{h_4 - 261.54} \longrightarrow h_4 = 292.78 \text{ kJ/kg}$$

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The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}_7(h_1 - h_8) = (0.11\,\text{kg/s})(244.51 - 68.81)\text{kJ/kg} = \mathbf{19.33\,kW}$$

(*c*) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}_7(h_2 - h_1) + \dot{m}(h_4 - h_9)$$
$$= (0.11\,\text{kg/s})(263.76 - 244.51)\text{kJ/kg} + (0.1707\,\text{kg/s})(292.78 - 261.54)\text{kJ/kg} = 7.45\,\text{kW}$$

$$\text{COP} = \frac{\dot{Q}_{\text{L}}}{\dot{W}_{\text{in}}} = \frac{19.33}{7.45} = \mathbf{2.59}$$

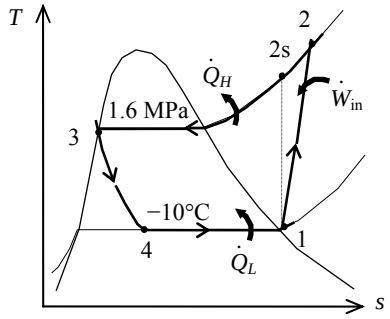
(*d*) If this refrigerator operated on a single-stage cycle between the same pressure limits, we would have

$$h_1 = h_{g@-10°\text{C}} = 244.51\,\text{kJ/kg}$$
$$s_1 = s_{g@-10°\text{C}} = 0.9377\,\text{kJ/kg.K}$$

$$\left.\begin{matrix} P_2 = 1600\,\text{kPa} \\ s_2 = s_1 \end{matrix}\right\} h_{2s} = 287.85\,\text{kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.86 = \frac{287.85 - 244.51}{h_2 - 244.51} \longrightarrow h_2 = 294.90\,\text{kJ/kg}$$



$$h_3 = h_{f@1600\,\text{kPa}} = 135.93\,\text{kJ/kg}$$
$$h_4 = h_3 = 135.93\,\text{kJ/kg}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.1707\,\text{kg/s})(244.51 - 135.93)\text{kJ/kg} = \mathbf{18.54\,kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.1707\,\text{kg/s})(294.90 - 244.51)\text{kJ/kg} = 8.60\,\text{kW}$$

$$\text{COP} = \frac{\dot{Q}_{\text{L}}}{\dot{W}_{\text{in}}} = \frac{18.54}{8.60} = \mathbf{2.16}$$

Discussion The cooling load decreases by 4.1% while the COP decreases by 16.6% when the cycle operates on the single-stage vapor-compression cycle.

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Gas Refrigeration Cycles

11-67C The ideal gas refrigeration cycle is identical to the Brayton cycle, except it operates in the reversed direction.

11-68C In the ideal gas refrigeration cycle, the heat absorption and the heat rejection processes occur at constant pressure instead of at constant temperature.

11-69C The reversed Stirling cycle is identical to the Stirling cycle, except it operates in the reversed direction. Remembering that the Stirling cycle is a totally reversible cycle, the reversed Stirling cycle is also totally reversible, and thus its COP is

$$\text{COP}_{\text{R, Stirling}} = \frac{1}{T_H / T_L - 1}$$

11-70C In aircraft cooling, the atmospheric air is compressed by a compressor, cooled by the surrounding air, and expanded in a turbine. The cool air leaving the turbine is then directly routed to the cabin.

11-71C No; because $h = h(T)$ for ideal gases, and the temperature of air will not drop during a throttling ($h_1 = h_2$) process.

11-72C By regeneration.

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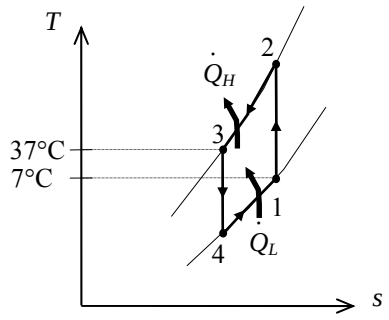
11-73 An ideal-gas refrigeration cycle with air as the working fluid is considered. The rate of refrigeration, the net power input, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Analysis (*a*) We assume both the turbine and the compressor to be isentropic, the turbine inlet temperature to be the temperature of the surroundings, and the compressor inlet temperature to be the temperature of the refrigerated space. From the air table (Table A-17),

$$T_1 = 280\ \text{K} \longrightarrow \begin{matrix} h_1 = 280.13\ \text{kJ/kg} \\ P_{r_1} = 1.0889 \end{matrix}$$

$$T_3 = 310\ \text{K} \longrightarrow \begin{matrix} h_3 = 310.24\ \text{kJ/kg} \\ P_{r_3} = 1.5546 \end{matrix}$$



Thus,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \left(\frac{160}{35}\right)(1.0889) = 4.978 \longrightarrow \begin{matrix} T_2 = 431.5\ \text{K} \\ h_2 = 432.96\ \text{kJ/kg} \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{35}{160}\right)(1.5546) = 0.3401 \longrightarrow \begin{matrix} T_4 = 200.6\ \text{K} \\ h_4 = 200.57\ \text{kJ/kg} \end{matrix}$$

Then the rate of refrigeration is

$$\dot{Q}_L = \dot{m}(q_L) = \dot{m}(h_1 - h_4) = (0.2\ \text{kg/s})(280.13 - 200.57)\ \text{kJ/kg} = \mathbf{15.9\ kW}$$

(*b*) The net power input is determined from

$$\dot{W}_{\text{net, in}} = \dot{W}_{\text{comp, in}} - \dot{W}_{\text{turb, out}}$$

where

$$\dot{W}_{\text{comp,in}} = \dot{m}(h_2 - h_1) = (0.2\ \text{kg/s})(432.96 - 280.13)\ \text{kJ/kg} = 30.57\ \text{kW}$$
$$\dot{W}_{\text{turb,out}} = \dot{m}(h_3 - h_4) = (0.2\ \text{kg/s})(310.24 - 200.57)\ \text{kJ/kg} = 21.93\ \text{kW}$$

Thus,

$$\dot{W}_{\text{net,in}} = 30.57 - 21.93 = \mathbf{8.64\ kW}$$

(*c*) The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{15.9\ \text{kW}}{8.64\ \text{kW}} = \mathbf{1.84}$$

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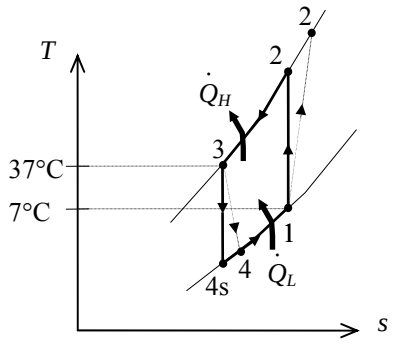
11-74 An ideal-gas refrigeration cycle with air as the working fluid is considered. The rate of refrigeration, the net power input, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Analysis (*a*) We assume the turbine inlet temperature to be the temperature of the surroundings, and the compressor inlet temperature to be the temperature of the refrigerated space. From the air table (Table A-17),

$$T_1 = 280\ \text{K} \longrightarrow \begin{matrix} h_1 = 280.13\ \text{kJ/kg} \\ P_{r_1} = 1.0889 \end{matrix}$$

$$T_3 = 310\ \text{K} \longrightarrow \begin{matrix} h_3 = 310.24\ \text{kJ/kg} \\ P_{r_3} = 1.5546 \end{matrix}$$



Thus,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \left(\frac{160}{35}\right)(1.0889) = 4.978 \longrightarrow \begin{matrix} T_{2s} = 431.5\ \text{K} \\ h_{2s} = 432.96\ \text{kJ/kg} \end{matrix}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{35}{160}\right)(1.5546) = 0.3401 \longrightarrow \begin{matrix} T_{4s} = 200.6\ \text{K} \\ h_{4s} = 200.57\ \text{kJ/kg} \end{matrix}$$

Also,

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow \begin{aligned} h_4 &= h_3 - \eta_T (h_3 - h_{4s}) \\ &= 310.24 - (0.85)(310.24 - 200.57) \\ &= 217.02\ \text{kJ/kg} \end{aligned}$$

Then the rate of refrigeration is

$$\dot{Q}_L = \dot{m}(q_L) = \dot{m}(h_1 - h_4) = (0.2\ \text{kg/s})(280.13 - 217.02)\ \text{kJ/kg} = \mathbf{12.6\ kW}$$

(*b*) The net power input is determined from

$$\dot{W}_{\text{net, in}} = \dot{W}_{\text{comp, in}} - \dot{W}_{\text{turb, out}}$$

where

$$\begin{aligned} \dot{W}_{\text{comp,in}} &= \dot{m}(h_2 - h_1) = \dot{m}(h_{2s} - h_1)/\eta_C \\ &= (0.2\ \text{kg/s})[(432.96 - 280.13)\ \text{kJ/kg}]/(0.80) = 38.21\ \text{kW} \end{aligned}$$

$$\dot{W}_{\text{turb,out}} = \dot{m}(h_3 - h_4) = (0.2\ \text{kg/s})(310.24 - 217.02)\ \text{kJ/kg} = 18.64\ \text{kW}$$

Thus,

$$\dot{W}_{\text{net,in}} = 38.21 - 18.64 = \mathbf{19.6\ kW}$$

(*c*) The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{12.6\ \text{kW}}{19.6\ \text{kW}} = \mathbf{0.643}$$

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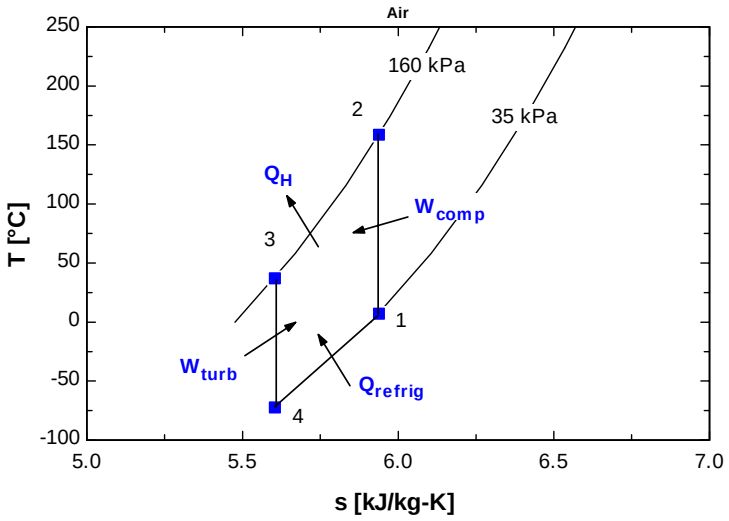
11-75 Problem 11-74 is reconsidered. The effects of compressor and turbine isentropic efficiencies on the rate of refrigeration, the net power input, and the COP are to be investigated.

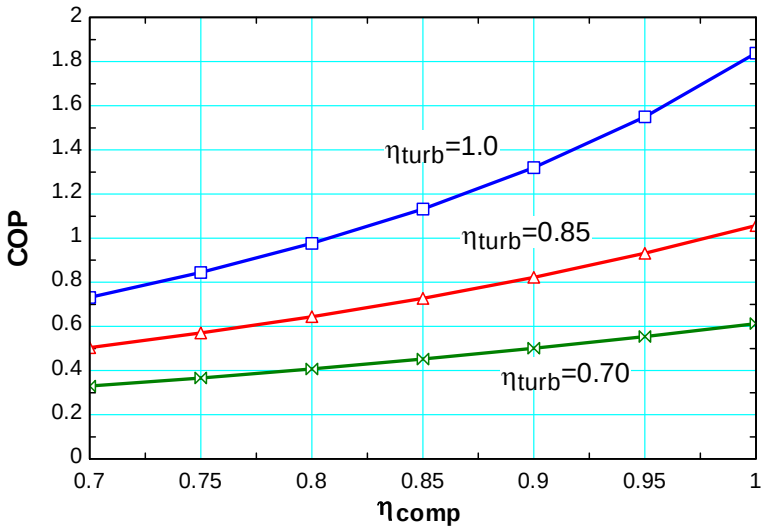
Analysis The problem is solved using EES, and the solution is given below.

```
"Input data"
T[1] = 7 [C]
P[1]= 35 [kPa]
T[3] = 37 [C]
P[3]=160 [kPa]
m_dot=0.2 [kg/s]
Eta_comp = 1.00
Eta_turb = 1.0
"Compressor anaysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s2s=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = P[3]
s2s=ENTROPY(Air,T=Ts2,P=P[2])"Ts2 is the isentropic value of T[2] at compressor exit"
Eta_comp = W_dot_comp_isen/W_dot_comp "compressor adiabatic efficiency,
W_dot_comp > W_dot_comp_isen"
m_dot*h[1] + W_dot_comp_isen = m_dot*hs2"SSSF First Law for the isentropic compressor,
assuming: adiabatic, ke=pe=0, m_dot is the mass flow rate in kg/s"
h[1]=ENTHALPY(Air,T=T[1])
hs2=ENTHALPY(Air,T=Ts2)
m_dot*h[1] + W_dot_comp = m_dot*h[2]"SSSF First Law for the actual compressor,
assuming: adiabatic, ke=pe=0"
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,h=h[2],P=P[2])
"Heat Rejection Process 2-3, assumed SSSF constant pressure process"
m_dot*h[2] + Q_dot_out = m_dot*h[3]"SSSF First Law for the heat exchanger,
assuming W=0, ke=pe=0"
h[3]=ENTHALPY(Air,T=T[3])
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s4s=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[1]
s4s=ENTROPY(Air,T=Ts4,P=P[4])"Ts4 is the isentropic value of T[4] at turbine exit"
Eta_turb = W_dot_turb /W_dot_turb_isen "turbine adiabatic efficiency, W_dot_turb_isen > W_dot_turb"
m_dot*h[3] = W_dot_turb_isen + m_dot*hs4"SSSF First Law for the isentropic turbine, assuming:
adiabatic, ke=pe=0"
hs4=ENTHALPY(Air,T=Ts4)
m_dot*h[3] = W_dot_turb + m_dot*h[4]"SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,h=h[4],P=P[4])
"Refrigeration effect:"
m_dot*h[4] + Q_dot_Refrig = m_dot*h[1]
"Cycle analysis"
W_dot_in_net=W_dot_comp-W_dot_turb"External work supplied to compressor"
COP= Q_dot_Refrig/W_dot_in_net
"The following is for plotting data only:"
Ts[1]=Ts2
ss[1]=s2s
Ts[2]=Ts4
ss[2]=s4s
```

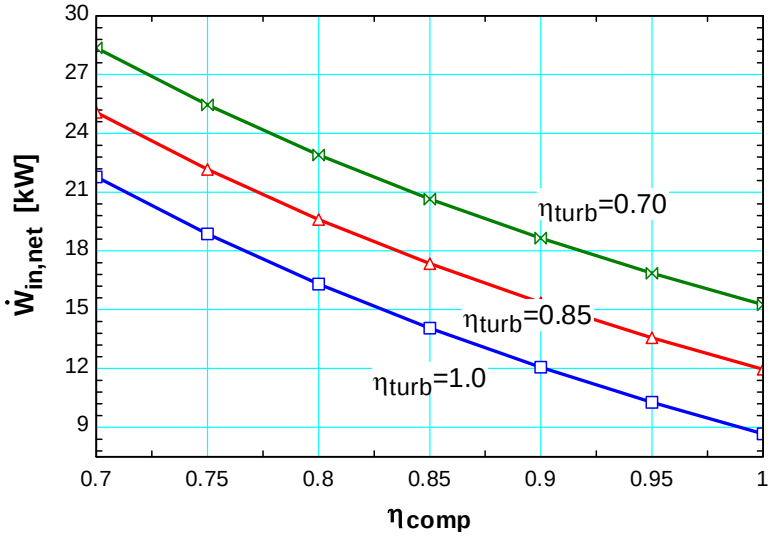
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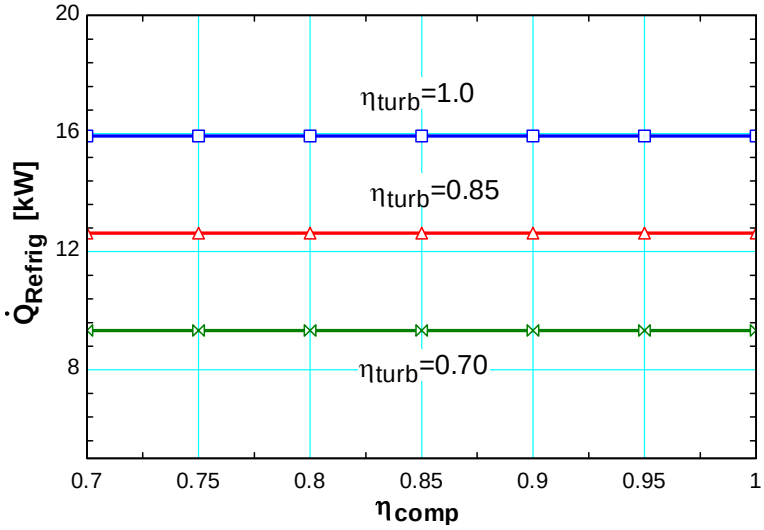
η_{comp}	COP	Q_{Refrig} [kW]	W_{innet} [kW]
0.7	0.3291	9.334	28.36
0.75	0.3668	9.334	25.45
0.8	0.4077	9.334	22.9
0.85	0.4521	9.334	20.65
0.9	0.5006	9.334	18.65
0.95	0.5538	9.334	16.86
1	0.6123	9.334	15.24





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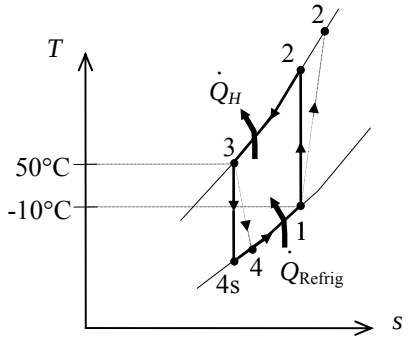


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11-76 A gas refrigeration cycle with helium as the working fluid is considered. The minimum temperature in the cycle, the COP, and the mass flow rate of the helium are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Helium is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of helium are $c_p = 5.1926$ kJ/kg·K and $k = 1.667$ (Table A-2).



Analysis (*a*) From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (263\text{K})(3)^{0.667/1.667} = 408.2\text{K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (323\text{K})\left(\frac{1}{3}\right)^{0.667/1.667} = 208.1\text{K}$$

and

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow T_4 = T_3 - \eta_T(T_3 - T_{4s}) = 323 - (0.80)(323 - 208.1)$$
$$= \mathbf{231.1\ K} = T_{\min}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \longrightarrow T_2 = T_1 + (T_{2s} - T_1)/\eta_C = 263 + (408.2 - 263)/(0.80)$$
$$= 444.5\text{ K}$$

(*b*) The COP of this gas refrigeration cycle is determined from

$$\text{COP}_R = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}}$$
$$= \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)}$$
$$= \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)}$$
$$= \frac{263 - 231.1}{(444.5 - 263) - (323 - 231.1)} = \mathbf{0.356}$$

(*c*) The mass flow rate of helium is determined from

$$\dot{m} = \frac{\dot{Q}_{\text{refrig}}}{q_L} = \frac{\dot{Q}_{\text{refrig}}}{h_1 - h_4} = \frac{\dot{Q}_{\text{refrig}}}{c_p(T_1 - T_4)} = \frac{18\text{ kJ/s}}{(5.1926\text{ kJ/kg}\cdot\text{K})(263 - 231.1)\text{K}} = \mathbf{0.109\ \ kg/s}$$

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11-77E An ideal gas refrigeration cycle with air as the working fluid has a compression ratio of 4. The COP of the cycle is to be determined.

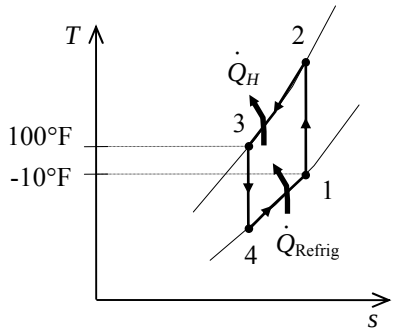
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis From the isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (450\text{ R})(4)^{0.4/1.4} = 668.7\text{ R}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (560\text{ R})\left(\frac{1}{4}\right)^{0.4/1.4} = 376.8\text{ R}$$



The COP of this ideal gas refrigeration cycle is determined from

$$\begin{aligned}\text{COP}_\text{R} &= \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}} \\ &= \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)} \\ &= \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)} \\ &= \frac{450 - 376.8}{(668.7 - 450) - (560 - 376.8)} \\ &= \mathbf{2.06}\end{aligned}$$

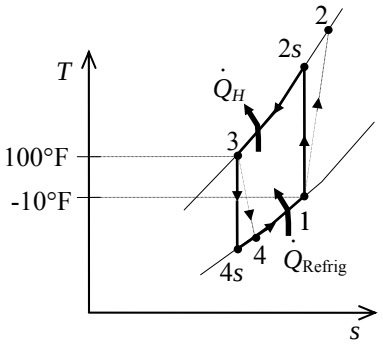
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11-78E An gas refrigeration cycle with air as the working fluid has a compression ratio of 4. The COP of the cycle is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are $c_p = 0.240$ Btu/lbm·R and $k = 1.4$ (Table A-2Ea).

Analysis From the isentropic relations,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (450\text{ R})(4)^{0.4/1.4} = 668.7\text{ R}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (560\text{ R})\left(\frac{6\text{ psia}}{19\text{ psia}}\right)^{0.4/1.4} = 402.9\text{ R}$$

and

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow T_4 = T_3 - \eta_T(T_3 - T_{4s}) = 560 - (0.94)(560 - 402.9) = 412.3\text{ R}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \longrightarrow T_2 = T_1 + (T_{2s} - T_1)/\eta_C = 450 + (668.7 - 450)/(0.87) = 701.4\text{ R}$$

The COP of this gas refrigeration cycle is determined from

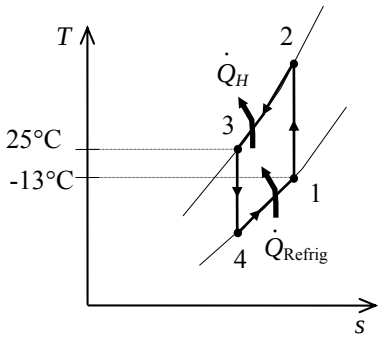
$$\begin{aligned}\text{COP}_R &= \frac{q_L}{w_{net,in}} = \frac{q_L}{w_{comp,in} - w_{turb,out}} \\ &= \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)} \\ &= \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)} \\ &= \frac{450 - 412.3}{(701.4 - 450) - (560 - 412.3)} \\ &= \mathbf{0.364}\end{aligned}$$

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11-79 An ideal gas refrigeration cycle with air as the working fluid is considered. The minimum pressure ratio for this system to operate properly is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).



Analysis An energy balance on process 4-1 gives

$$q_{\text{Refrig}} = c_p (T_1 - T_4)$$

$$T_4 = T_1 - \frac{q_{\text{Refrig}}}{c_p} = 260\ \text{K} - \frac{36\ \text{kJ/kg}}{1.005\ \text{kJ/kg} \cdot \text{K}} = 224.2\ \text{K}$$

The minimum temperature at the turbine inlet would be the same as that to which the heat is rejected. That is,

$$T_3 = 298\ \text{K}$$

Then the minimum pressure ratio is determined from the isentropic relation to be

$$\frac{P_3}{P_4} = \left(\frac{T_3}{T_4}\right)^{k/(k-1)} = \left(\frac{298\ \text{K}}{224.2\ \text{K}}\right)^{1.4/0.4} = \mathbf{2.71}$$

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11-80 A regenerative gas refrigeration cycle using air as the working fluid is considered. The effectiveness of the regenerator, the rate of heat removal from the refrigerated space, the COP of the cycle, and the refrigeration load and the COP if this system operated on the simple gas refrigeration cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

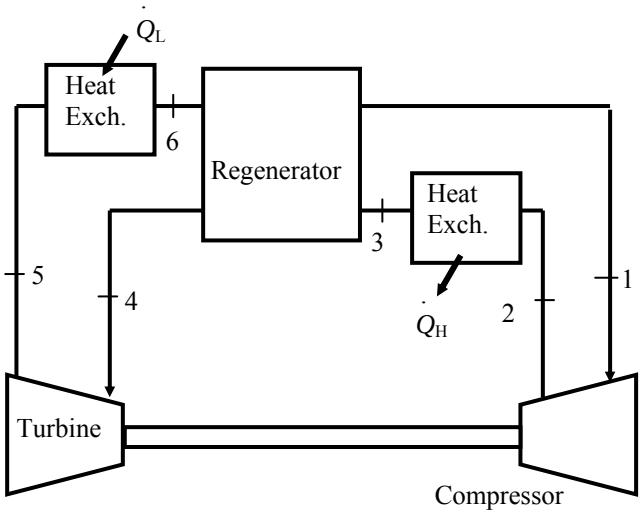
Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2).

Analysis (*a*) From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (273.2\text{ K})(5)^{0.4/1.4} = 432.4\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{T_{2s}-T_1}{T_2-T_1}$$

$$0.80 = \frac{432.4-273.2}{T_2-273.2} \longrightarrow T_2 = 472.5\text{ K}$$



The temperature at state 4 can be determined by solving the following two equations simultaneously:

$$T_{5s} = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = T_4\left(\frac{1}{5}\right)^{0.4/1.4}$$

$$\eta_T = \frac{h_4-h_5}{h_4-h_{5s}} \rightarrow 0.85 = \frac{T_4-193.2}{T_4-T_{5s}}$$

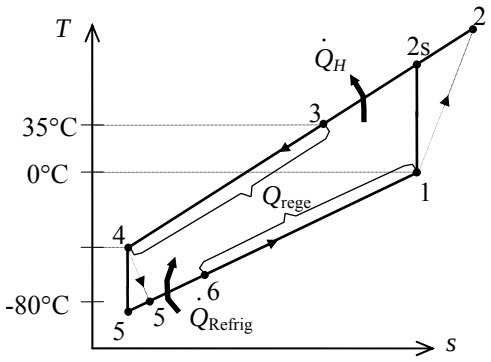
Using EES, we obtain $T_4 = 281.3$ K.

An energy balance on the regenerator may be written as

$$\dot{m}c_p(T_3-T_4) = \dot{m}c_p(T_1-T_6) \longrightarrow T_3-T_4 = T_1-T_6$$

or,

$$T_6 = T_1 - T_3 + T_4 = 273.2 - 308.2 + 281.3 = 246.3\text{ K}$$



The effectiveness of the regenerator is

$$\varepsilon_{regen} = \frac{h_3-h_4}{h_3-h_6} = \frac{T_3-T_4}{T_3-T_6} = \frac{308.2-281.3}{308.2-246.3} = \mathbf{0.434}$$

(*b*) The refrigeration load is

$$\dot{Q}_L = \dot{m}c_p(T_6-T_5) = (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(246.3-193.2)\text{K} = \mathbf{21.36\text{ kW}}$$

(*c*) The turbine and compressor powers and the COP of the cycle are

$$\dot{W}_{C,in} = \dot{m}c_p(T_2-T_1) = (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(472.5-273.2)\text{K} = 80.13\text{ kW}$$

$$\dot{W}_{T,out} = \dot{m}c_p(T_4-T_5) = (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(281.3-193.2)\text{kJ/kg} = 35.43\text{ kW}$$

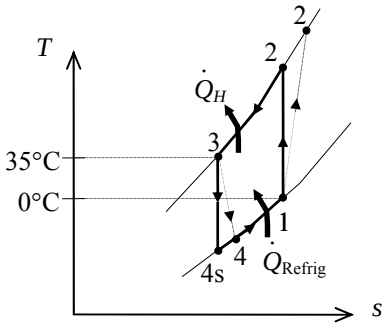
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{\dot{Q}_L}{\dot{W}_{C,in}-\dot{W}_{T,out}} = \frac{21.36}{80.13-35.43} = \mathbf{0.478}$$

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(*d*) The simple gas refrigeration cycle analysis is as follows:

$$T_{4s} = T_3\left(\frac{1}{r}\right)^{(k-1)/k} = (308.2\text{ K})\left(\frac{1}{5}\right)^{0.4/1.4} = 194.6\text{ K}$$

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow 0.85 = \frac{308.2 - T_4}{308.2 - 194.6} \longrightarrow T_4 = 211.6\text{ K}$$



$$\dot{Q}_L = \dot{m}c_p(T_1 - T_4)$$
$$= (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(273.2 - 211.6)\text{kJ/kg}$$
$$= \mathbf{24.74\text{ kW}}$$

$$\dot{W}_{\text{net,in}} = \dot{m}c_p(T_2 - T_1) - \dot{m}c_p(T_3 - T_4)$$
$$= (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})\left[(472.5 - 273.2) - (308.2 - 211.6)\text{kJ/kg}\right]$$
$$= 41.32\text{ kW}$$

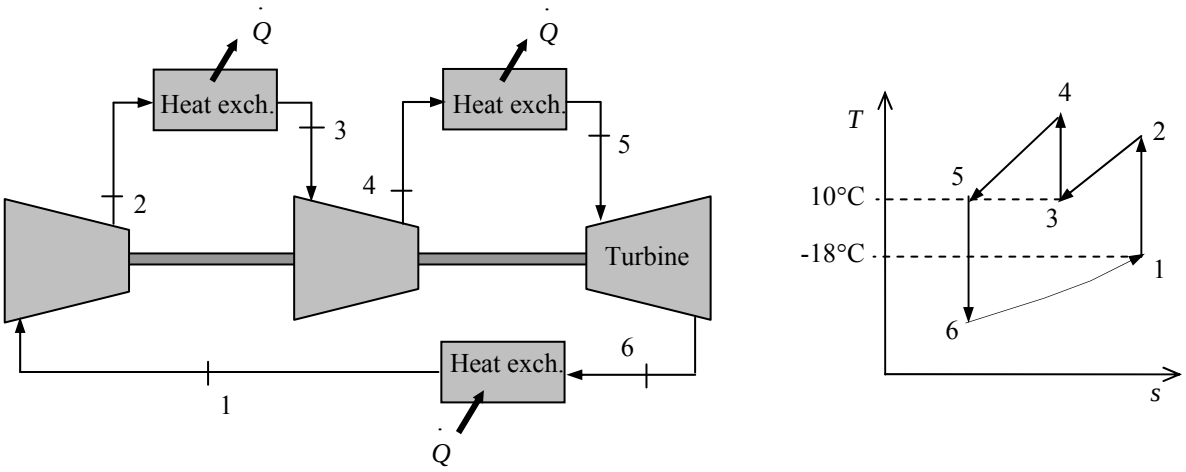
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{24.74}{41.32} = \mathbf{0.599}$$

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11-81 An ideal gas refrigeration cycle with with two stages of compression with intercooling using air as the working fluid is considered. The COP of this system and the mass flow rate of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).



Analysis From the isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (255\text{ K})(4)^{0.4/1.4} = 378.9\text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (283\text{ K})(4)^{0.4/1.4} = 420.5\text{ K}$$

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (283\text{ K})\left(\frac{1}{16}\right)^{0.4/1.4} = 128.2\text{ K}$$

The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_R = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}}$$

$$= \frac{h_1 - h_6}{(h_2 - h_1) + (h_4 - h_3) - (h_5 - h_6)}$$

$$= \frac{T_1 - T_6}{(T_2 - T_1) + (T_4 - T_3) - (T_5 - T_6)}$$

$$= \frac{255 - 128.2}{(378.9 - 255) + (420.5 - 283) - (283 - 128.2)} = \mathbf{1.19}$$

The mass flow rate of the air is determined from

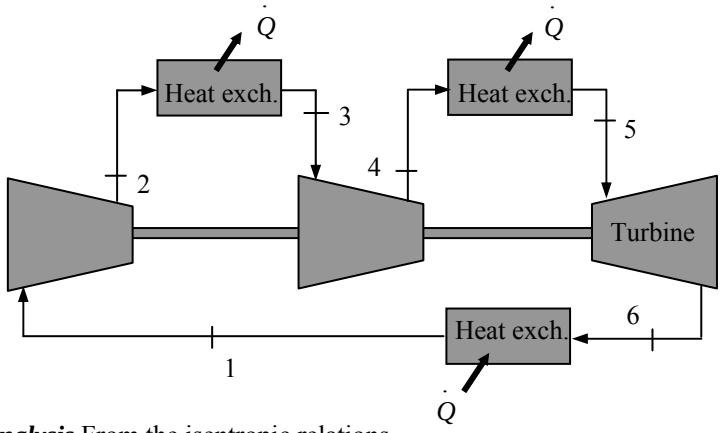
$$\dot{Q}_{\text{Refrig}} = \dot{m}c_p(T_1 - T_6) \longrightarrow \dot{m} = \frac{\dot{Q}_{\text{Refrig}}}{c_p(T_1 - T_6)} = \frac{(75{,}000/3600)\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot\text{K})(255 - 128.2)\text{ K}} = \mathbf{0.163\ kg/s}$$

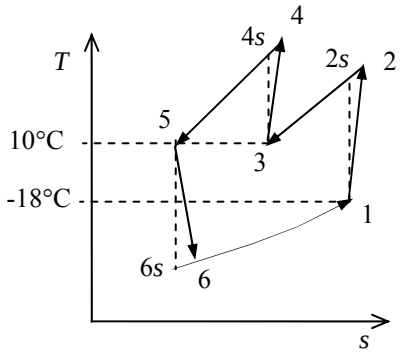
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11-82 A gas refrigeration cycle with with two stages of compression with intercooling using air as the working fluid is considered. The COP of this system and the mass flow rate of air are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas with constant specific heats. 3 Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$ (Table A-2a).





Analysis From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (255\text{ K})(4)^{0.4/1.4} = 378.9\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (283\text{ K})(4)^{0.4/1.4} = 420.5\text{ K}$$

$$T_{6s} = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (283\text{ K})\left(\frac{1}{16}\right)^{0.4/1.4} = 128.2\text{ K}$$

and

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{T_{2s}-T_1}{T_2-T_1} \longrightarrow T_2 = T_1 + (T_{2s}-T_1)/\eta_C = 255 + (378.9-255)/0.85 = 400.8\text{ K}$$

$$\eta_C = \frac{h_{4s}-h_3}{h_4-h_3} = \frac{T_{4s}-T_3}{T_4-T_3} \longrightarrow T_4 = T_3 + (T_{4s}-T_3)/\eta_C = 283 + (420.5-283)/0.85 = 444.8\text{ K}$$

$$\eta_T = \frac{h_5-h_6}{h_5-h_{6s}} = \frac{T_5-T_6}{T_5-T_{6s}} \longrightarrow T_6 = T_5 - \eta_T(T_5-T_{6s}) = 283-(0.95)(283-128.2) = 135.9\text{ K}$$

The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}}$$

$$= \frac{h_1-h_6}{(h_2-h_1)+(h_4-h_3)-(h_5-h_6)}$$

$$= \frac{T_1-T_6}{(T_2-T_1)+(T_4-T_3)-(T_5-T_6)}$$

$$= \frac{255-135.9}{(400.8-255)+(444.8-283)-(283-135.9)} = \mathbf{0.742}$$

The mass flow rate of the air is determined from

$$\dot{Q}_{\text{Refrig}} = \dot{m}c_p(T_1-T_6) \longrightarrow \dot{m} = \frac{\dot{Q}_{\text{Refrig}}}{c_p(T_1-T_6)} = \frac{(75{,}000/3600)\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot\text{K})(255-135.9)\text{ K}} = \mathbf{0.174\text{ kg/s}}$$

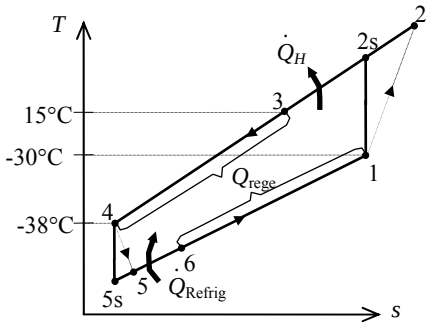
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11-83 A regenerative gas refrigeration cycle with argon as the working fluid is considered. Te refrigeration load, the COP, the minimum power input, the second-law efficiency, and the total exergy destruction in the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Properties The properties of argon are $c_p = 0.5203$ kJ/kg·K and $k = 1.667$.

Analysis (a) From the isentropic relations,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (243\text{ K})(5)^{0.667/1.667} = 462.7\text{ K}$$

$$T_{5s} = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = (235\text{ K})\left(\frac{1}{5}\right)^{0.667/1.667} = 123.4\text{ K}$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} = \frac{T_4 - T_5}{T_4 - T_{5s}}$$

$$\longrightarrow T_5 = T_4 - \eta_T(T_4 - T_{5s}) = 235 - (0.82)(235 - 123.4) = 143.5\text{ K}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$\longrightarrow T_2 = T_1 + (T_{2s} - T_1)/\eta_C = 243 + (462.7 - 243)/0.82 = 510.9\text{ K}$$

From an energy balance on the regenerator,

$$\dot{m}c_p(T_3 - T_4) = \dot{m}c_p(T_1 - T_6) \longrightarrow T_3 - T_4 = T_1 - T_6$$

or

$$T_6 = T_1 - T_3 + T_4 = 243 - 288 + 235 = 190\text{ K}$$

$$\dot{Q}_L = \dot{m}c_p(T_6 - T_5) = (0.08\text{ kg/s})(0.5203\text{ kJ/kg}\cdot\text{K})(190 - 143.5)\text{ K} = \mathbf{1.935\ kW}$$

$$\dot{W}_{net} = \dot{m}c_p[(T_2 - T_1) - (T_4 - T_5)]$$
$$= (0.08\text{ kg/s})(0.5203\text{ kJ/kg}\cdot\text{K})[(510.9 - 243) - (235 - 143.5)]\text{ K} = 7.343\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net}}} = \frac{1.935}{7.343} = \mathbf{0.2636}$$

(b) The exergy of the heat transferred from the low-temperature medium is

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right) = -(1.935\text{ kW})\left(1 - \frac{273}{228}\right) = 0.382\text{ kW}$$

This is the minimum power input:

$$\dot{W}_{\min} = \dot{E}x_{\dot{Q}_L} = \mathbf{0.382\ kW}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{E}x_{\dot{Q}_L}}{\dot{W}_{\text{net}}} = \frac{0.382}{7.343} = 0.05202 = \mathbf{5.2\%}$$

The total exergy destruction in the cycle can be determined from

$$\dot{E}x_{\text{dest,total}} = \dot{W}_{\text{net}} - \dot{E}x_{\dot{Q}_L} = 7.343 - 0.382 = \mathbf{6.961kW}$$

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Absorption Refrigeration Systems

11-84C In absorption refrigeration, water can be used as the refrigerant in air conditioning applications since the temperature of water never needs to fall below the freezing point.

11-85C Absorption refrigeration is the kind of refrigeration that involves the absorption of the refrigerant during part of the cycle. In absorption refrigeration cycles, the refrigerant is compressed in the liquid phase instead of in the vapor form.

11-86C The main advantage of absorption refrigeration is its being economical in the presence of an inexpensive heat source. Its disadvantages include being expensive, complex, and requiring an external heat source.

11-87C The fluid in the absorber is cooled to maximize the refrigerant content of the liquid; the fluid in the generator is heated to maximize the refrigerant content of the vapor.

11-88C The coefficient of performance of absorption refrigeration systems is defined as

$$\text{COP}_\text{R} = \frac{\text{desiredoutput}}{\text{requiredinput}} = \frac{Q_L}{Q_\text{gen} + W_\text{pump,in}} \cong \frac{Q_L}{Q_\text{gen}}$$

11-89C The rectifier separates the water from NH_3 and returns it to the generator. The regenerator transfers some heat from the water-rich solution leaving the generator to the NH_3-rich solution leaving the pump.

11-90 The COP of an absorption refrigeration system that operates at specified conditions is given. It is to be determined whether the given COP value is possible.

Analysis The maximum COP that this refrigeration system can have is

$$\text{COP}_\text{R,max} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{292\ \text{K}}{368\ \text{K}}\right)\left(\frac{273}{292 - 273}\right) = 2.97$$

which is smaller than 3.1. Thus the claim is **not possible**.

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11-91 The conditions at which an absorption refrigeration system operates are specified. The maximum COP this absorption refrigeration system can have is to be determined.

Analysis The maximum COP that this refrigeration system can have is

$$\text{COP}_{\text{R,max}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{298\ \text{K}}{393\ \text{K}}\right)\left(\frac{273}{298 - 273}\right) = \mathbf{2.64}$$

11-92 The conditions at which an absorption refrigeration system operates are specified. The maximum rate at which this system can remove heat from the refrigerated space is to be determined.

Analysis The maximum COP that this refrigeration system can have is

$$\text{COP}_{\text{R,max}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{298\ \text{K}}{403\ \text{K}}\right)\left(\frac{243}{298 - 243}\right) = 1.15$$

Thus,

$$\dot{Q}_{\text{L,max}} = \text{COP}_{\text{R,max}}\dot{Q}_{\text{gen}} = (1.15)(5 \times 10^5\ \text{kJ/h}) = \mathbf{5.75 \times 10^5\ kJ/h}$$

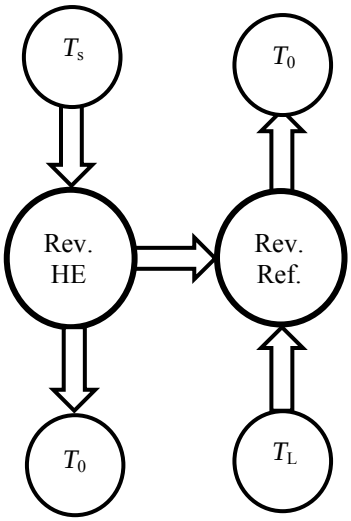
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11-93 A reversible absorption refrigerator consists of a reversible heat engine and a reversible refrigerator. The rate at which the steam condenses, the power input to the reversible refrigerator, and the second law efficiency of an actual chiller are to be determined.

Properties The enthalpy of vaporization of water at 150°C is h_{fg} = 2113.8 kJ/kg (Table A-4).

Analysis (*a*) The thermal efficiency of the reversible heat engine is

$$\eta_{\text{th,rev}} = 1 - \frac{T_0}{T_s} = 1 - \frac{(25 + 273.15)\,\text{K}}{(150 + 273.15)\,\text{K}} = 0.2954$$



The COP of the reversible refrigerator is

$$\text{COP}_{\text{R,rev}} = \frac{T_L}{T_0 - T_L} = \frac{(-15 + 273.15)\,\text{K}}{(25 + 273.15) - (-15 + 273.15)\,\text{K}} = 6.454$$

The COP of the reversible absorption refrigerator is

$$\text{COP}_{\text{abs,rev}} = \eta_{\text{th,rev}}\text{COP}_{\text{R,rev}} = (0.2954)(6.454) = 1.906$$

The heat input to the reversible heat engine is

$$\dot{Q}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{abs,rev}}} = \frac{70\,\text{kW}}{1.906} = 36.72\,\text{kW}$$

Then, the rate at which the steam condenses becomes

$$\dot{m}_s = \frac{\dot{Q}_{\text{in}}}{h_{fg}} = \frac{36.72\,\text{kJ/s}}{2113.8\,\text{kJ/kg}} = \mathbf{0.0174\,kg/s}$$

(*b*) The power input to the refrigerator is equal to the power output from the heat engine

$$\dot{W}_{\text{in,R}} = \dot{W}_{\text{out,HE}} = \eta_{\text{th,rev}}\dot{Q}_{\text{in}} = (0.2954)(36.72\,\text{kW}) = \mathbf{10.9\,kW}$$

(*c*) The second-law efficiency of an actual absorption chiller with a COP of 0.8 is

$$\eta_{\text{II}} = \frac{\text{COP}_{\text{actual}}}{\text{COP}_{\text{abs,rev}}} = \frac{0.8}{1.906} = 0.420 = \mathbf{42.0\%}$$

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11-94E An ammonia-water absorption refrigeration cycle is considered. The rate of cooling, the COP, and the second-law efficiency of the system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Properties The properties of ammonia are as given in the problem statement. The specific heat of geothermal water is given to be 1.0 Btu/lbm·°F.

Analysis (a) The rate of cooling provided by the system is

$$\dot{Q}_L = \dot{m}_R (h_1 - h_4) = (0.04 \text{ lbm/s})(619.2 - 190.9) \text{ Btu/lbm}$$
$$= 17.13 \text{ Btu/s} = \mathbf{61{,}700 \text{ Btu/h}}$$

(b) The rate of heat input to the generator is

$$\dot{Q}_{\text{gen}} = \dot{m}_{\text{geo}} c_p (T_{\text{geo;in}} - T_{\text{geo;out}}) = (0.55 \text{ lbm/s})(1.0 \text{ Btu/lbm} \cdot \text{°F})(240 - 200)\text{°F} = 22.0 \text{ Btu/s}$$

Then the COP becomes

$$\text{COP} = \frac{\dot{Q}_L}{\dot{Q}_{\text{gen}}} = \frac{17.13 \text{ Btu/s}}{22.0 \text{ Btu/s}} = \mathbf{0.779}$$

(c) The reversible COP of the system is

$$\text{COP}_{\text{abs,rev}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{(70 + 460)}{(220 + 460)}\right)\left(\frac{(25 + 460)}{70 - 25}\right) = 2.38$$

The temperature of the heat source is taken as the average temperature of the geothermal water: (240+200)/2=220°F. Then the second-law efficiency becomes

$$\eta_{\text{II}} = \frac{\text{COP}}{\text{COP}_{\text{abs,rev}}} = \frac{0.779}{2.38} = 0.328 = \mathbf{32.8\%}$$

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Special Topic: Thermoelectric Power Generation and Refrigeration Systems

11-95C The circuit that incorporates both thermal and electrical effects is called a thermoelectric circuit.

11-96C When two wires made from different metals joined at both ends (junctions) forming a closed circuit and one of the joints is heated, a current flows continuously in the circuit. This is called the Seebeck effect. When a small current is passed through the junction of two dissimilar wires, the junction is cooled. This is called the Peltier effect.

11-97C No.

11-98C No.

11-99C Yes.

11-100C When a thermoelectric circuit is broken, the current will cease to flow, and we can measure the voltage generated in the circuit by a voltmeter. The voltage generated is a function of the temperature difference, and the temperature can be measured by simply measuring voltages.

11-101C The performance of thermoelectric refrigerators improves considerably when semiconductors are used instead of metals.

11-102C The efficiency of a thermoelectric generator is limited by the Carnot efficiency because a thermoelectric generator fits into the definition of a heat engine with electrons serving as the working fluid.

11-103E A thermoelectric generator that operates at specified conditions is considered. The maximum thermal efficiency this thermoelectric generator can have is to be determined.

Analysis The maximum thermal efficiency of this thermoelectric generator is the Carnot efficiency,

$$\eta_{\text{th,max}} = \eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} = 1 - \frac{550\text{R}}{800\text{R}} = \mathbf{31.3\%}$$

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11-104 A thermoelectric refrigerator that operates at specified conditions is considered. The maximum COP this thermoelectric refrigerator can have and the minimum required power input are to be determined.

Analysis The maximum COP of this thermoelectric refrigerator is the COP of a Carnot refrigerator operating between the same temperature limits,

$$\mathrm{COP}_{\max} = \mathrm{COP}_{\mathrm{R,Carnot}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(293\ \mathrm{K})/(268\ \mathrm{K}) - 1} = \mathbf{10.72}$$

Thus,

$$\dot{W}_{\mathrm{in,min}} = \frac{\dot{Q}_L}{\mathrm{COP}_{\max}} = \frac{130\ \mathrm{W}}{10.72} = \mathbf{12.1\ W}$$

11-105 A thermoelectric cooler that operates at specified conditions with a given COP is considered. The required power input to the thermoelectric cooler is to be determined.

Analysis The required power input is determined from the definition of COP_R,

$$\mathrm{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\mathrm{in}}} \quad \longrightarrow \quad \dot{W}_{\mathrm{in}} = \frac{\dot{Q}_L}{\mathrm{COP}_R} = \frac{180\ \mathrm{W}}{0.15} = \mathbf{1200\ W}$$

11-106E A thermoelectric cooler that operates at specified conditions with a given COP is considered. The rate of heat removal is to be determined.

Analysis The required power input is determined from the definition of COP_R,

$$\mathrm{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\mathrm{in}}} \longrightarrow \dot{Q}_L = \mathrm{COP}_R \dot{W}_{\mathrm{in}} = (0.18)(1.8\ \mathrm{hp})\left(\frac{42.41\ \mathrm{Btu/min}}{1\ \mathrm{hp}}\right) = \mathbf{13.7\ Btu/min}$$

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11-107 A thermoelectric refrigerator powered by a car battery cools 9 canned drinks in 12 h. The average COP of this refrigerator is to be determined.

Assumptions Heat transfer through the walls of the refrigerator is negligible.

Properties The properties of canned drinks are the same as those of water at room temperature, $\rho = 1$ kg/L and $c_p = 4.18$ kJ/kg·°C (Table A-3).

Analysis The cooling rate of the refrigerator is simply the rate of decrease of the energy of the canned drinks,

$$m = \rho \boldsymbol{V} = 9 \times (1\,\text{kg/L})(0.350\,\text{L}) = 3.15\,\text{kg}$$
$$Q_{\text{cooling}} = mc\Delta T = (3.15\,\text{kg})(4.18\,\text{kJ/kg}\cdot\text{°C})(25-3)\text{°C} = 290\,\text{kJ}$$
$$\dot{Q}_{\text{cooling}} = \frac{Q_{\text{cooling}}}{\Delta t} = \frac{290\,\text{kJ}}{12 \times 3600\,\text{s}} = 0.00671\,\text{kW} = 6.71\,\text{W}$$

The electric power consumed by the refrigerator is

$$\dot{W}_{\text{in}} = \mathbf{V}I = (12\,\text{V})(3\,\text{A}) = 36\,\text{W}$$

Then the COP of the refrigerator becomes

$$\text{COP} = \frac{\dot{Q}_{\text{cooling}}}{\dot{W}_{\text{in}}} = \frac{6.71\,\text{W}}{36\,\text{W}} = \mathbf{0.186} \approx 0.20$$

11-108E A thermoelectric cooler is said to cool a 12-oz drink or to heat a cup of coffee in about 15 min. The average rate of heat removal from the drink, the average rate of heat supply to the coffee, and the electric power drawn from the battery of the car are to be determined.

Assumptions Heat transfer through the walls of the refrigerator is negligible.

Properties The properties of canned drinks are the same as those of water at room temperature, $c_p = 1.0$ Btu/lbm.°F (Table A-3E).

Analysis (*a*) The average cooling rate of the refrigerator is simply the rate of decrease of the energy content of the canned drinks,

$$Q_{\text{cooling}} = mc_p\Delta T = (0.771\ \text{lbm})(1.0\ \text{Btu/lbm}\cdot\text{°F})(78-38)\text{°F} = 30.84\,\text{Btu}$$
$$\dot{Q}_{\text{cooling}} = \frac{Q_{\text{cooling}}}{\Delta t} = \frac{30.84\,\text{Btu}}{15 \times 60\,\text{s}}\left(\frac{1055\,\text{J}}{1\,\text{Btu}}\right) = \mathbf{36.2\ W}$$

(*b*) The average heating rate of the refrigerator is simply the rate of increase of the energy content of the canned drinks,

$$Q_{\text{heating}} = mc_p\Delta T = (0.771\,\text{lbm})(1.0\ \text{Btu/lbm}\cdot\text{°F})(130-75)\text{°F} = 42.4\,\text{Btu}$$
$$\dot{Q}_{\text{heating}} = \frac{Q_{\text{heating}}}{\Delta t} = \frac{42.4\,\text{Btu}}{15 \times 60\,\text{s}}\left(\frac{1055\,\text{J}}{1\,\text{Btu}}\right) = \mathbf{49.7\ W}$$

(*c*) The electric power drawn from the car battery during cooling and heating is

$$\dot{W}_{\text{in,cooling}} = \frac{\dot{Q}_{\text{cooling}}}{\text{COP}_{\text{cooling}}} = \frac{36.2\,\text{W}}{0.2} = \mathbf{181\ W}$$
$$\text{COP}_{\text{heating}} = \text{COP}_{\text{cooling}} + 1 = 0.2 + 1 = 1.2$$
$$\dot{W}_{\text{in,heating}} = \frac{\dot{Q}_{\text{heating}}}{\text{COP}_{\text{heating}}} = \frac{49.7\,\text{W}}{1.2} = \mathbf{41.4\ W}$$

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11-109 The maximum power a thermoelectric generator can produce is to be determined.

Analysis The maximum thermal efficiency this thermoelectric generator can have is

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{295 \text{ K}}{363 \text{ K}} = 0.1873$$

Thus,

$$\dot{W}_{\text{out,max}} = \eta_{\text{th,max}} \dot{Q}_{\text{in}} = (0.1873)(7 \times 10^6 \text{ kJ/h}) = 1.31 \times 10^6 \text{ kJ/h} = \mathbf{364 \text{ kW}}$$

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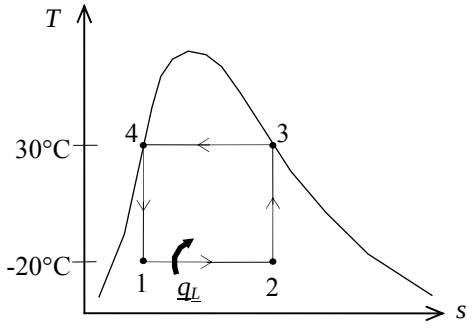
Review Problems

11-110 A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The COP, the condenser and evaporator pressures, and the net work input are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The COP of this refrigeration cycle is determined from

$$\text{COP}_{\text{R,C}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(303\text{ K})/(253\text{ K}) - 1} = \mathbf{5.06}$$



(*b*) The condenser and evaporative pressures are (Table A-11)

$$P_{\text{evap}} = P_{\text{sat@}-20°\text{C}} = \mathbf{132.82\ kPa}$$
$$P_{\text{cond}} = P_{\text{sat@}30°\text{C}} = \mathbf{770.64\ kPa}$$

(*c*) The net work input is determined from

$$h_1 = \left(h_f + x_1 h_{fg}\right)_{@-20°\text{C}} = 25.49 + (0.15)(212.91) = 57.43\text{ kJ/kg}$$
$$h_2 = \left(h_f + x_2 h_{fg}\right)_{@-20°\text{C}} = 25.49 + (0.80)(212.91) = 195.82\text{ kJ/kg}$$

$$q_L = h_2 - h_1 = 195.82 - 57.43 = 138.4\text{kJ/kg}$$
$$w_{\text{net,in}} = \frac{q_L}{\text{COP}_{\text{R}}} = \frac{138.4\text{ kJ/kg}}{5.06} = \mathbf{27.35\ kJ/kg}$$

11-111 A room is cooled adequately by a 5000 Btu/h window air-conditioning unit. The rate of heat gain of the room when the air-conditioner is running continuously is to be determined.

Assumptions **1** The heat gain includes heat transfer through the walls and the roof, infiltration heat gain, solar heat gain, internal heat gain, etc. **2** Steady operating conditions exist.

Analysis The rate of heat gain of the room in steady operation is simply equal to the cooling rate of the air-conditioning system,

$$\dot{Q}_{\text{heat gain}} = \dot{Q}_{\text{cooling}} = \mathbf{5{,}000\ Btu/h}$$

11-112 A heat pump water heater has a COP of 3.4 and consumes 6 kW when running. It is to be determined if this heat pump can be used to meet the cooling needs of a room by absorbing heat from it.

Assumptions The COP of the heat pump remains constant whether heat is absorbed from the outdoor air or room air.

Analysis The COP of the heat pump is given to be 3.4. Then the COP of the air-conditioning system becomes

$$\text{COP}_{\text{air-cond}} = \text{COP}_{\text{heat pump}} - 1 = 3.4 - 1 = 2.4$$

Then the rate of cooling (heat absorption from the air) becomes

$$\dot{Q}_{\text{cooling}} = \text{COP}_{\text{air-cond}} \dot{W}_{in} = (3.4)(6\text{ kW}) = 20.4\text{ kW} = 51{,}840\text{ kJ/h}$$

since 1 kW = 3600 kJ/h. We conclude that this heat pump **can meet** the cooling needs of the room since its cooling rate is greater than the rate of heat gain of the room.

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11-113 A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid is used to heat a house. The rate of heat supply to the house, the volume flow rate of the refrigerant at the compressor inlet, and the COP of this heat pump are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

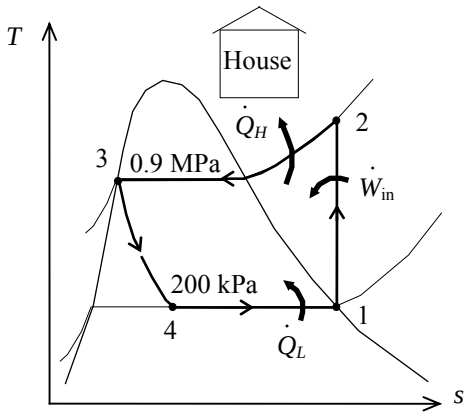
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,200\,\text{kPa}} = 244.46\ \text{kJ/kg} \\ s_1 = s_{g\,@\,200\,\text{kPa}} = 0.93773\ \text{kJ/kg}\cdot\text{K} \\ \nu_1 = \nu_{g\,@\,200\,\text{kPa}} = 0.099867\ \text{m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.9\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 275.75\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.9\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.9\,\text{MPa}} = 101.61\ \text{kJ/kg}$$

$$h_4 \cong h_3 = 101.61\ \text{kJ/kg}\ \ (\text{throttling})$$



The rate of heat supply to the house is determined from

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.32\ \text{kg/s})(275.75 - 101.61)\ \text{kJ/kg} = \mathbf{55.73\ kW}$$

(*b*) The volume flow rate of the refrigerant at the compressor inlet is

$$\dot{V}_1 = \dot{m}\nu_1 = (0.32\ \text{kg/s})(0.099867\ \text{m}^3/\text{kg}) = \mathbf{0.0320\ \ m^3/s}$$

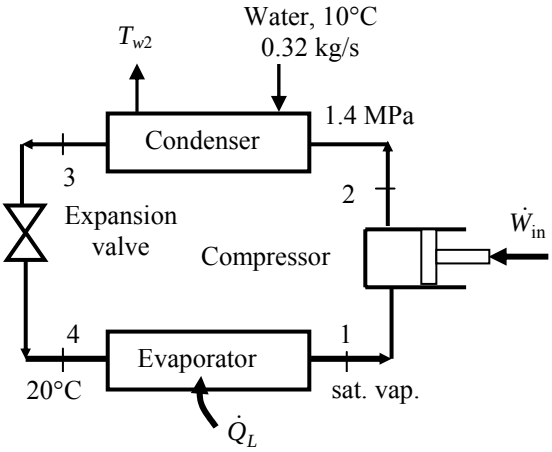
(*c*) The COP of t his heat pump is determined from

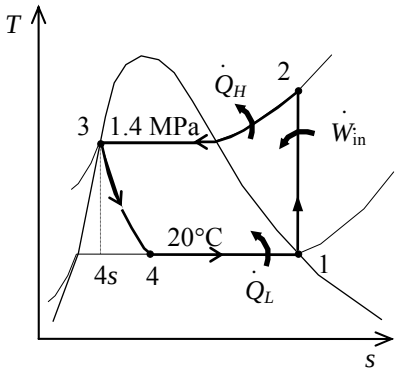
$$\text{COP}_R = \frac{q_L}{w_{\text{in}}} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{275.75 - 101.61}{275.75 - 244.46} = \mathbf{5.57}$$

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11-114 A ground-coupled heat pump that operates on the vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The hardware and the *T*-*s* diagram for this air conditioner are to be sketched. The exit temperature of the water in the condenser and the COP are to be determined.

Analysis (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure.





(b) The properties as given in the problem statement are

$$h_4 = h_3 = h_{f\,@\,1400\,\text{kPa}} = 127.2\ \text{kJ/kg}$$

$$h_1 = h_{g\,@\,20°\text{C}} = 261.6\ \text{kJ/kg}.$$

The rate of heat transfer in the condenser is determined from

$$\dot{Q}_H - \dot{Q}_L = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \dot{W}_\text{in}$$

$$\dot{Q}_H = \dot{Q}_L\left(1 + \frac{1}{\text{COP}_\text{R}}\right) = (18\,\text{kW})\left(1 + \frac{1}{6}\right) = 21\,\text{kW}$$

An energy balance on the condenser gives

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = \dot{m}_w c_{pw}(T_{2w} - T_{1w})$$

$$T_{2w} = T_{1w} + \frac{\dot{Q}_H}{\dot{m}_w c_{pw}} = 10°\text{C} + \frac{21\,\text{kW}}{(0.32\,\text{kg/s})(4.18\,\text{kJ/kg°C})} = \mathbf{25.7°C}$$

(c) The COP of the heat pump is

$$\text{COP}_\text{HP} = \frac{\dot{Q}_H}{\dot{Q}_H - \dot{Q}_L} = \frac{21\,\text{kW}}{21\,\text{kW} - 18\,\text{kW}} = \mathbf{7}$$

It may also be determined from

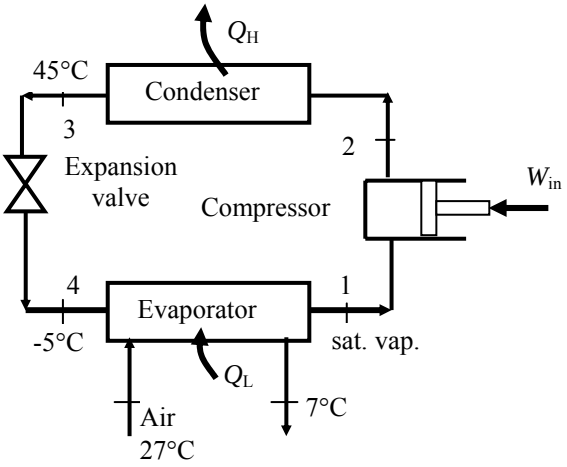
$$\text{COP}_\text{HP} = \text{COP}_\text{R} + 1 = 6 + 1 = \mathbf{7}$$

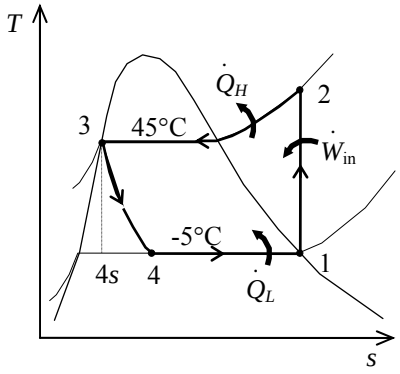
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11-115 An ideal vapor-compression refrigeration cycle with refrigerant-22 as the working fluid is considered. The evaporator is located inside the air handler of building. The hardware and the T-s diagram for this heat pump application are to be sketched. The COP of the unit and the ratio of volume flow rate of air entering the air handler to mass flow rate of R-22 through the air handler are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant-22 data from the problem statement,





$$\left.\begin{array}{l} T_1 = -5°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,-5°\text{C}} = 248.1\,\text{kJ/kg} \\ s_1 = s_{g\,@\,-5°\text{C}} = 0.9344\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1728\,\text{kPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 283.7\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1728\,\text{kPa} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\,@\,1728\,\text{kPa}} = 101\,\text{kJ/kg}$$

$$h_4 \cong h_3 = 101\,\text{kJ/kg} \quad \text{(throttling)}$$

(*b*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{248.1 - 101}{283.7 - 248.1} = \mathbf{4.13}$$

(*c*) An energy balance on the evaporator gives

$$\dot{Q}_L = \dot{m}_R (h_1 - h_4) = \dot{m}_a c_p \Delta T = \frac{\dot{V}_a}{v_a} c_p \Delta T$$

Rearranging, we obtain the ratio of volume flow rate of air entering the air handler to mass flow rate of R-22 through the air handler

$$\frac{\dot{V}_a}{\dot{m}_R} = \frac{h_1 - h_4}{(1/v) c_p \Delta T} = \frac{(248.1 - 101)\,\text{kJ/kg}}{(1/0.8323\,\text{m}^3/\text{kg})(1.005\,\text{kJ/kg}\cdot\text{K}))(20\,\text{K})}$$

$$= 6.091\,(\text{m}^3\ \text{air/s})/(\text{kg R22/s})$$

$$= \mathbf{365\ (m^3\ air/min)/(kg\,R22/s)}$$

Note that the specific volume of air is obtained from ideal gas equation taking the pressure of air to be 100 kPa (given) and using the average temperature of air (17°C = 290 K) to be 0.8323 m^3/kg.

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11-116 An air conditioner operates on the vapor-compression refrigeration cycle. The rate of cooling provided to the space, the COP, the isentropic efficiency and the exergetic efficiency of the compressor, the exergy destruction in each component of the cycle, the total exergy destruction, the minimum power input, and the second-law efficiency of the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) The properties of R-134a are (Tables A-11 through A-13)

$$T_{\text{sat@180 kPa}} = -12.7°\text{C}$$

$$\left.\begin{aligned} P_1 &= 180\ \text{kPa} \\ T_1 &= -12.7 + 2.7 = 10°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 245.14\ \text{kJ/kg} \\ s_1 &= 0.9483\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 1200\ \text{kPa} \\ s_1 &= s_1 \end{aligned}\right\} h_{2s} = 285.32\ \text{kJ/kg}$$

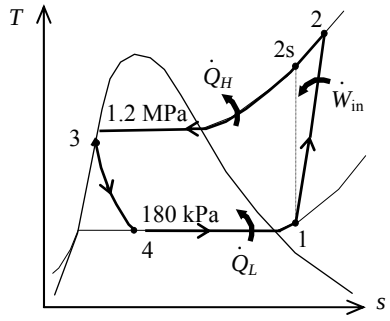
$$\left.\begin{aligned} P_2 &= 1200\ \text{kPa} \\ T_2 &= 60°\text{C} \end{aligned}\right\} \begin{aligned} h_2 &= 289.64\ \text{kJ/kg} \\ s_2 &= 0.9614\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$T_{\text{sat@1200 kPa}} = 46.3°\text{C}$$

$$\left.\begin{aligned} P_3 &= 1200\ \text{kPa} \\ T_3 &= 46.3 - 6.3 = 40°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &\cong h_{f@40°\text{C}} = 108.26\ \text{kJ/kg} \\ s_3 &\cong s_{f@40°\text{C}} = 0.3948\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$h_4 = h_3 = 108.26\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_4 &= 180\ \text{kPa} \\ h_4 &= 108.26\ \text{kJ/kg} \end{aligned}\right\} s_4 = 0.4228\ \text{kJ/kg}\cdot\text{K}$$



The cooling load and the COP are

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.06\ \text{kg/s})(245.14 - 108.26)\text{kJ/kg} = 8.213\ \text{kW}$$
$$= (8.213\ \text{kW})\left(\frac{3412\ \text{Btu/h}}{1\ \text{kW}}\right) = \mathbf{28{,}020\ Btu/h}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.06\ \text{kg/s})(289.64 - 108.26)\text{kJ/kg} = 10.88\ \text{kW}$$
$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.06\ \text{kg/s})(289.64 - 245.14)\text{kJ/kg} = 2.670\ \text{kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{8.213\ \text{kW}}{2.670\ \text{kW}} = \mathbf{3.076}$$

(b) The isentropic efficiency of the compressor is

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{285.32 - 245.14}{289.64 - 245.14} = 0.9029 = \mathbf{90.3\%}$$

The reversible power and the exergy efficiency for the compressor are

$$\dot{W}_{rev} = \dot{m}\left[(h_2 - h_1) - T_0(s_2 - s_1)\right]$$
$$= (0.06\ \text{kg/s})\left[(289.64 - 245.14)\text{kJ/kg} - (310\ \text{K})(0.9614 - 0.9483)\text{kJ/kg}\cdot\text{K}\right]$$
$$= 2.428\ \text{kW}$$

$$\eta_{ex,C} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{in}}} = \frac{2.428\ \text{kW}}{2.670\ \text{kW}} = 0.9091 = \mathbf{90.9\%}$$

(c) The exergy destruction in each component of the cycle is determined as follows

Compressor:

$$\dot{S}_{\text{gen,1-2}} = \dot{m}(s_2 - s_1) = (0.06\ \text{kg/s})(0.9614 - 0.9483)\ \text{kJ/kg}\cdot\text{K} = 0.0007827\ \text{kW/K}$$

$$\dot{E}x_{\text{dest,1-2}} = T_0\dot{S}_{\text{gen,1-2}} = (310\ \text{K})(0.0007827\ \text{kW/K}) = \mathbf{0.2426\ kW}$$

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Condenser:

$$\dot{S}_{\text{gen,2–3}} = \dot{m}(s_3 - s_2) + \frac{\dot{Q}_H}{T_H} = (0.06\ \text{kg/s})(0.3948 - 0.9614)\ \text{kJ/kg} \cdot \text{K} + \frac{10.88\ \text{kW}}{310\ \text{K}} = 0.001114\ \text{kW/K}$$

$$\dot{E}x_{\text{dest,2-3}} = T_0 \dot{S}_{\text{gen,2-3}} = (310\ \text{K})(0.001114\ \text{kJ/kg} \cdot \text{K}) = \mathbf{0.3452\ kW}$$

Expansion valve:

$$\dot{S}_{\text{gen,3–4}} = \dot{m}(s_4 - s_3) = (0.06\ \text{kg/s})(0.4228 - 00.3948)\ \text{kJ/kg} \cdot \text{K} = 0.001678\ \text{kW/K}$$

$$\dot{E}x_{\text{dest,3-4}} = T_0 \dot{S}_{\text{gen,3-4}} = (310\ \text{K})(0.001678\ \text{kJ/kg} \cdot \text{K}) = \mathbf{0.5203\ kJ/kg}$$

Evaporator:

$$\dot{S}_{\text{gen,4–1}} = \dot{m}(s_1 - s_4) - \frac{\dot{Q}_L}{T_L} = (0.06\ \text{kg/s})(0.9483 - 0.4228)\ \text{kJ/kg} \cdot \text{K} - \frac{8.213\ \text{kW}}{294\ \text{K}} = 0.003597\ \text{kW/K}$$

$$\dot{E}x_{\text{dest,4-1}} = T_0 \dot{S}_{\text{gen,4-1}} = (310\ \text{K})(0.003597\ \text{kJ/kg} \cdot \text{K}) = \mathbf{1.115\ kW}$$

The total exergy destruction can be determined by adding exergy destructions in each component:

$$\begin{aligned}\dot{E}x_{\text{dest,total}} &= \dot{E}x_{\text{dest,1-2}} + \dot{E}x_{\text{dest,2-3}} + \dot{E}x_{\text{dest,3-4}} + \dot{E}x_{\text{dest,4-1}} \\ &= 0.2426 + 0.3452 + 0.5203 + 1.115 \\ &= \mathbf{2.223\ kW}\end{aligned}$$

(d) The exergy of the heat transferred from the low-temperature medium is

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right) = -(8.213\ \text{kW})\left(1 - \frac{310}{294}\right) = 0.4470\ \text{kW}$$

This is the minimum power input to the cycle:

$$\dot{W}_{\text{in,min}} = \dot{E}x_{\dot{Q}_L} = \mathbf{0.4470\ kW}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{in,min}}}{\dot{W}_{\text{in}}} = \frac{0.4470}{2.670} = 0.1674 = \mathbf{16.7\%}$$

The total exergy destruction in the cycle can also be determined from

$$\dot{E}x_{\text{dest,total}} = \dot{W}_{\text{in}} - Ex_{\dot{Q}_L} = 2.670 - 0.4470 = 2.223\ \text{kW}$$

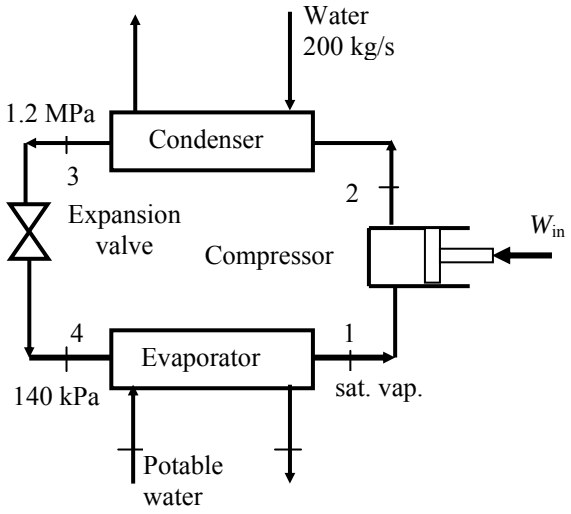
The result is the same as expected.

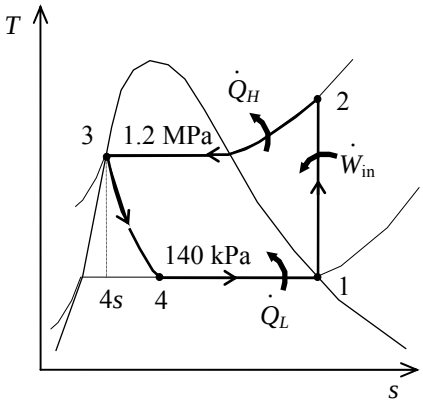
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11-117 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. Cooling water flows through the water jacket surrounding the condenser. To produce ice, potable water is supplied to the chiller section of the refrigeration cycle. The hardware and the T-s diagram for this refrigerant-ice making system are to be sketched. The mass flow rates of the refrigerant and the potable water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant-134a tables,





$$\left.\begin{array}{l} T_1 = 140\ \text{kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 140\ \text{kPa}} = 239.16\ \text{kJ/kg} \\ s_1 = s_{g\ @\ 140\ \text{kPa}} = 0.94456\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1200\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 284.07\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1200\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\ @\ 1200\ \text{kPa}} = 117.77\ \text{kJ/kg}$$

$$h_4 \cong h_3 = 117.77\ \text{kJ/kg} \quad \text{(throttling)}$$

(*b*) An energy balance on the condenser gives

$$\dot{Q}_H = \dot{m}_R (h_2 - h_3) = \dot{m}_w c_p \Delta T$$

Solving for the mass flow rate of the refrigerant

$$\dot{m}_R = \frac{\dot{m}_w c_p \Delta T}{h_2 - h_3} = \frac{(200\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot\text{K})(10\ \text{K})}{(284.07 - 117.77)\text{kJ/kg}} = \mathbf{50.3\ kg/s}$$

(*c*) An energy balance on the evaporator gives

$$\dot{Q}_L = \dot{m}_R (h_1 - h_4) = \dot{m}_w h_{if}$$

Solving for the mass flow rate of the potable water

$$\dot{m}_w = \frac{\dot{m}_R (h_1 - h_4)}{h_{if}} = \frac{(50.3\ \text{kg/s})(239.16 - 117.77)\text{kJ/kg}}{333\ \text{kJ/kg}} = \mathbf{18.3\ kg/s}$$

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11-118 A refrigerator operating on a vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The process with the greatest exergy loss is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis In this cycle, the refrigerant leaves the condenser as saturated liquid at the condenser pressure. The compression process is not isentropic. From the refrigerant tables (Tables A-11, A-12, and A-13),

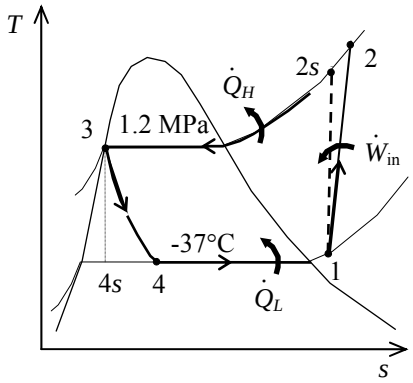
$$\left.\begin{array}{l} P_1 = P_{\text{sat @ } -37^\circ\text{C}} = 60\ \text{kPa} \\ T_1 = -37 + 7 = -30^\circ\text{C} \end{array}\right\} \begin{array}{l} h_1 = 233.09\ \text{kJ/kg} \\ s_1 = 0.9867\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} \quad h_{2s} = 298.11\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.2\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\ @\ 1.2\ \text{MPa}} = 117.77\ \text{kJ/kg} \\ s_3 = s_{f\ @\ 1.2\ \text{MPa}} = 0.42441\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$h_4 \cong h_3 = 117.77\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_4 = -37^\circ\text{C} \\ h_4 = 117.77\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} x_4 = 0.5089 \\ s_4 = 0.4988\ \text{kJ/kg}\cdot\text{K} \end{array}$$



The actual enthalpy at the compressor exit is determined by using the compressor efficiency:

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C} = 233.09 + \frac{298.11 - 233.09}{0.90} = 305.33\ \text{kJ/kg}$$

and $$\left.\begin{array}{l} P_2 = 1.2\ \text{MPa} \\ h_2 = 305.33\ \text{Btu/lbm} \end{array}\right\} \quad s_2 = 1.0075\ \text{kJ/kg}\cdot\text{K}$$

The heat added in the evaporator and that rejected in the condenser are

$$q_L = h_1 - h_4 = (233.09 - 117.77)\ \text{kJ/kg} = 115.32\ \text{kJ/kg}$$
$$q_H = h_2 - h_3 = (305.33 - 117.77)\ \text{kJ/kg} = 187.56\ \text{kJ/kg}$$

The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0 \left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}} \right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed},12} = T_0 (s_2 - s_1) = (303\ \text{K})(1.0075 - 0.9867)\ \text{kJ/kg}\cdot\text{K} = 6.37\ \text{kJ/kg}$$

$$x_{\text{destroyed},23} = T_0 \left(s_3 - s_2 + \frac{q_H}{T_H} \right) = (303\ \text{K}) \left(0.42441 - 1.0075 + \frac{187.56\ \text{kJ/kg}}{303\ \text{K}} \right) = 10.88\ \text{kJ/kg}$$

$$x_{\text{destroyed},34} = T_0 (s_4 - s_3) = (303\ \text{K})(0.4988 - 0.42441)\ \text{kJ/kg}\cdot\text{K} = \mathbf{22.54\ kJ/kg}$$

$$x_{\text{destroyed},41} = T_0 \left(s_1 - s_4 - \frac{q_L}{T_L} \right) = (303\ \text{K}) \left(0.9867 - 0.4988 - \frac{115.3\ \text{kJ/kg}}{(-34 + 273)\ \text{K}} \right) = 1.66\ \text{kJ/kg}$$

The greatest exergy destruction occurs in the expansion valve. Note that heat is absorbed from fruits at -34°C (239 K) and rejected to the ambient air at 30°C (303 K), which is also taken as the dead state temperature. Alternatively, one may use the standard 25°C (298 K) as the dead state temperature, and perform the calculations accordingly.

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11-119 A refrigerator operating on a vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The process with the greatest exergy loss is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From the refrigerant tables (Tables A-11, A-12, and A-13),

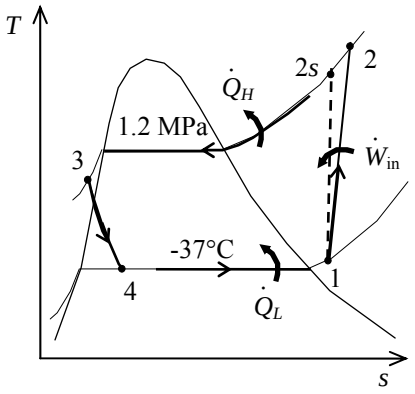
$$\left.\begin{array}{l} P_1 = P_{\text{sat @ } -37^\circ\text{C}} = 60\ \text{kPa} \\ T_1 = -37 + 7 = -30^\circ\text{C} \end{array}\right\} \begin{array}{l} h_1 = 233.09\ \text{kJ/kg} \\ s_1 = 0.9867\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} \quad h_{2s} = 298.11\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.2\ \text{MPa} \\ T_3 = T_{\text{sat @ 1.2 MPa}} - 6.3 \\ \quad = 46.3 - 6.3 = 40^\circ\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f\ @\ 40^\circ\text{C}} = 108.26\ \text{kJ/kg} \\ s_3 \cong s_{f\ @\ 40^\circ\text{C}} = 0.39486\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$h_4 \cong h_3 = 108.26\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_4 = -37^\circ\text{C} \\ h_4 = 108.26\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} x_4 = 0.4665 \\ s_4 = 0.4585\ \text{kJ/kg}\cdot\text{K} \end{array}$$



The actual enthalpy at the compressor exit is determined by using the compressor efficiency:

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C} = 233.09 + \frac{298.11 - 233.09}{0.90} = 305.33\ \text{kJ/kg}$$

and $$\left.\begin{array}{l} P_2 = 1.2\ \text{MPa} \\ h_2 = 305.33\ \text{Btu/lbm} \end{array}\right\} \quad s_2 = 1.0075\ \text{kJ/kg}\cdot\text{K}$$

The heat added in the evaporator and that rejected in the condenser are

$$q_L = h_1 - h_4 = (233.09 - 108.26)\ \text{kJ/kg} = 124.83\ \text{kJ/kg}$$
$$q_H = h_2 - h_3 = (305.33 - 108.26)\ \text{kJ/kg} = 197.07\ \text{kJ/kg}$$

The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0 \left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}} \right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed},12} = T_0 (s_2 - s_1) = (303\ \text{K})(1.0075 - 0.9867)\ \text{kJ/kg}\cdot\text{K} = 6.37\ \text{kJ/kg}$$

$$x_{\text{destroyed},23} = T_0 \left(s_3 - s_2 + \frac{q_H}{T_H} \right) = (303\ \text{K}) \left(0.39486 - 1.0075 + \frac{197.07\ \text{kJ/kg}}{303\ \text{K}} \right) = 11.44\ \text{kJ/kg}$$

$$x_{\text{destroyed},34} = T_0 (s_4 - s_3) = (303\ \text{K})(0.4585 - 0.39486)\ \text{kJ/kg}\cdot\text{K} = \mathbf{19.28\ kJ/kg}$$

$$x_{\text{destroyed},41} = T_0 \left(s_1 - s_4 - \frac{q_L}{T_L} \right) = (303\ \text{K}) \left(0.9867 - 0.4585 - \frac{124.83\ \text{kJ/kg}}{(-34 + 273)\ \text{K}} \right) = 1.80\ \text{kJ/kg}$$

The greatest exergy destruction occurs in the expansion valve. Note that heat is absorbed from fruits at -34°C (239 K) and rejected to the ambient air at 30°C (303 K), which is also taken as the dead state temperature. Alternatively, one may use the standard 25°C (298 K) as the dead state temperature, and perform the calculations accordingly.

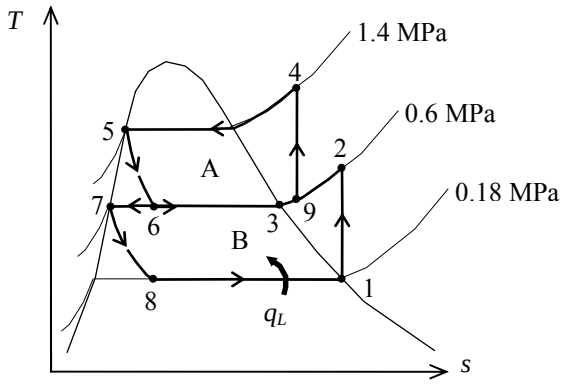
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11-120 A two-stage compression refrigeration system using refrigerant-134a as the working fluid is considered. The fraction of the refrigerant that evaporates as it is throttled to the flash chamber, the amount of heat removed from the refrigerated space, the compressor work, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The flashing chamber is adiabatic.

Analysis (*a*) The enthalpies of the refrigerant at several states are determined from the refrigerant tables to be (Tables A-11, A-12, and A-13)

$$h_1 = 242.86 \text{ kJ/kg}, \quad h_2 = 267.72 \text{ kJ/kg}$$
$$h_3 = 262.40 \text{ kJ/kg},$$
$$h_5 = 127.22 \text{ kJ/kg}, \quad h_6 = 127.22 \text{ kJ/kg}$$
$$h_7 = 81.51 \text{ kJ/kg}, \quad h_8 = 81.51 \text{ kJ/kg}$$



The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6,

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{127.22 - 81.51}{180.90} = \mathbf{0.2527}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$
$$h_9 = (0.2527)(262.40) + (1 - 0.2527)(267.72) = 266.38 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 0.6 \text{ MPa} \\ h_9 = 266.38 \text{ kJ/kg} \end{array}\right\} s_9 = 0.93516 \text{ kJ/kg} \cdot \text{K}$$

Also,

$$\left.\begin{array}{l} P_4 = 1.4 \text{ MPa} \\ s_4 = s_9 = 0.93516 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_4 = 284.23 \text{ kJ/kg}$$

Then the amount of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$q_L = (1 - x_6)(h_1 - h_8) = (1 - 0.2527)(242.86 - 81.51) \text{ kJ/kg} = \mathbf{120.6 \text{ kJ/kg}}$$

$$w_{\text{in}} = w_{\text{compI,in}} + w_{\text{compII,in}} = (1 - x_6)(h_2 - h_1) + (1)(h_4 - h_9)$$
$$= (1 - 0.2527)(267.72 - 242.86) \text{kJ/kg} + (284.23 - 266.38) \text{kJ/kg} = \mathbf{36.43 \text{ kJ/kg}}$$

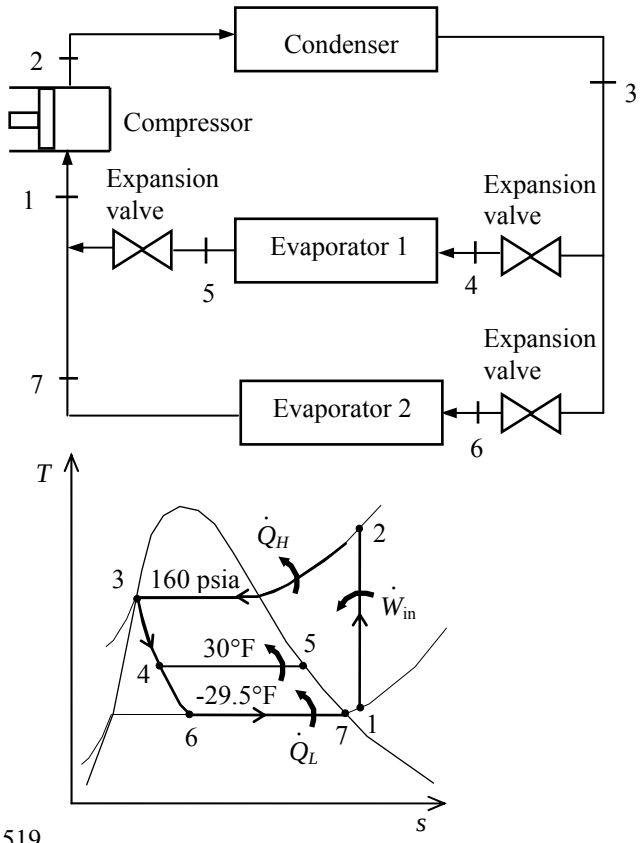
(*c*) The coefficient of performance is determined from

$$\text{COP}_{\text{R}} = \frac{q_L}{w_{\text{in}}} = \frac{120.6 \text{ kJ/kg}}{36.43 \text{ kJ/kg}} = \mathbf{3.31}$$

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11-121E A two-evaporator compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The cooling load of both evaporators per unit of flow through the compressor and the COP of the system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From the refrigerant tables (Tables A-11E, A-12E, and A-13E),

$$\left.\begin{array}{l} P_3 = 160 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \; h_3 = h_{f\,@\,160\text{ psia}} = 48.519 \text{ Btu/lbm}$$

$$h_4 = h_6 \cong h_3 = 48.519 \text{ Btu/lbm} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_5 = 30°\text{F} \\ \text{sat. vapor} \end{array}\right\} \; h_5 = h_{g\,@\,30°\text{F}} = 107.40 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} T_7 = -29.5°\text{F} \\ \text{sat. vapor} \end{array}\right\} \; h_7 = h_{g\,@\,-29.5°\text{F}} = 98.68 \text{ Btu/lbm}$$

For a unit mass flowing through the compressor, the fraction of mass flowing through Evaporator II is denoted by x and that through Evaporator I is y ($y = 1\text{-}x$). From the cooling loads specification,

$$\dot{Q}_{L,\text{e vap 1}} = 2\dot{Q}_{L,\text{e vap 2}}$$
$$x(h_5 - h_4) = 2y(h_7 - h_6)$$

where $x = 1 - y$

Combining these results and solving for y gives

$$y = \frac{h_5 - h_4}{2(h_7 - h_6) + (h_5 - h_4)} = \frac{107.40 - 48.519}{2(98.68 - 48.519) + (107.40 - 48.519)} = 0.3698$$

Then, $x = 1 - y = 1 - 0.3698 = 0.6302$

Applying an energy balance to the point in the system where the two evaporator streams are recombined gives

$$xh_5 + yh_7 = h_1 \longrightarrow h_1 = \frac{xh_5 + yh_7}{1} = \frac{(0.6302)(107.40) + (0.3698)(98.68)}{1} = 104.18 \text{ Btu/lbm}$$

Then,

$$\left.\begin{array}{l} P_1 = P_{\text{sat}\,@\,-29.5°\text{F}} \cong 10 \text{ psia} \\ h_1 = 104.18 \text{ Btu/lbm} \end{array}\right\} \; s_1 = 0.2418 \text{ Btu/lbm} \cdot \text{R}$$

$$\left.\begin{array}{l} P_2 = 160 \text{ psia} \\ s_2 = s_1 \end{array}\right\} \; h_2 = 131.14 \text{ Btu/lbm}$$

The cooling load of both evaporators per unit mass through the compressor is

$$\begin{aligned} q_L &= x(h_5 - h_4) + y(h_7 - h_6) \\ &= (0.6302)(107.40 - 48.519) \text{ Btu/lbm} + (0.3698)(98.68 - 48.519) \text{ Btu/lbm} \\ &= \mathbf{55.7 \text{ Btu/lbm}} \end{aligned}$$

The work input to the compressor is

$$w_{\text{in}} = h_2 - h_1 = (131.14 - 104.18) \text{ Btu/lbm} = 27.0 \text{ Btu/lbm}$$

The COP of this refrigeration system is determined from its definition,

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{in}}} = \frac{55.7 \text{ Btu/lbm}}{27.0 \text{ Btu/lbm}} = \mathbf{2.06}$$

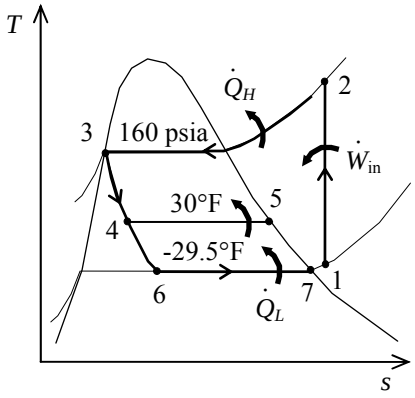
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11-122E A two-evaporator compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The process with the greatest exergy destruction is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Prob. 11-121E and the refrigerant tables (Tables A-11E, A-12E, and A-13E),

$$\begin{aligned}
s_1 &= s_2 = 0.2418\ \text{Btu/lbm}\cdot\text{R}\\
s_3 &= 0.09774\ \text{Btu/lbm}\cdot\text{R}\\
s_4 &= 0.1024\ \text{Btu/lbm}\cdot\text{R}\\
s_5 &= 0.2226\ \text{Btu/lbm}\cdot\text{R}\\
s_6 &= 0.1129\ \text{Btu/lbm}\cdot\text{R}\\
s_7 &= 0.2295\ \text{Btu/lbm}\cdot\text{R}\\
x &= 0.6302\\
y &= 1 - x = 0.3698\\
q_{L,45} &= h_5 - h_4 = 58.88\ \text{Btu/lbm}\\
q_{L,67} &= h_7 - h_6 = 50.17\ \text{Btu/lbm}\\
q_H &= 82.61\ \text{Btu/lbm}
\end{aligned}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives the exergy destructions per unit mass flowing through the compressor:

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 + \frac{q_H}{T_H}\right) = (540\ \text{R})\left(0.09774 - 0.2418 + \frac{82.61\ \text{Btu/lbm}}{540\ \text{R}}\right) = 4.82\ \text{Btu/lbm}$$

$$\begin{aligned}
x_{\text{destroyed},346} &= T_0(xs_4 + ys_6 - s_3)\\
&= (540\ \text{R})(0.6302\times 0.1024 + 0.3698\times 0.1129 - 0.09774)\ \text{Btu/lbm}\cdot\text{R} = 4.60\ \text{Btu/lbm}
\end{aligned}$$

$$\begin{aligned}
x_{\text{destroyed},45} &= xT_0\left(s_5 - s_4 - \frac{q_{L,45}}{T_L}\right)\\
&= (0.6302)(540\ \text{R})\left(0.2226 - 0.1024 - \frac{58.88\ \text{Btu/lbm}}{500\ \text{R}}\right) = 0.84\ \text{Btu/lbm}
\end{aligned}$$

$$\begin{aligned}
x_{\text{destroyed},67} &= yT_0\left(s_7 - s_6 - \frac{q_{L,67}}{T_L}\right)\\
&= (0.3698)(540\ \text{R})\left(0.2295 - 0.1129 - \frac{50.17\ \text{Btu/lbm}}{445\ \text{R}}\right) = 0.77\ \text{Btu/lbm}
\end{aligned}$$

$$\begin{aligned}
\dot{X}_{\text{destroyed, mixing}} &= T_0(s_1 - xs_5 - ys_7)\\
&= (540\ \text{R})\left[0.2418 - (0.6302)(0.2226) - (0.3698)(0.2295)\right] = \mathbf{9.00\ Btu/lbm}
\end{aligned}$$

For isentropic processes, the exergy destruction is zero:

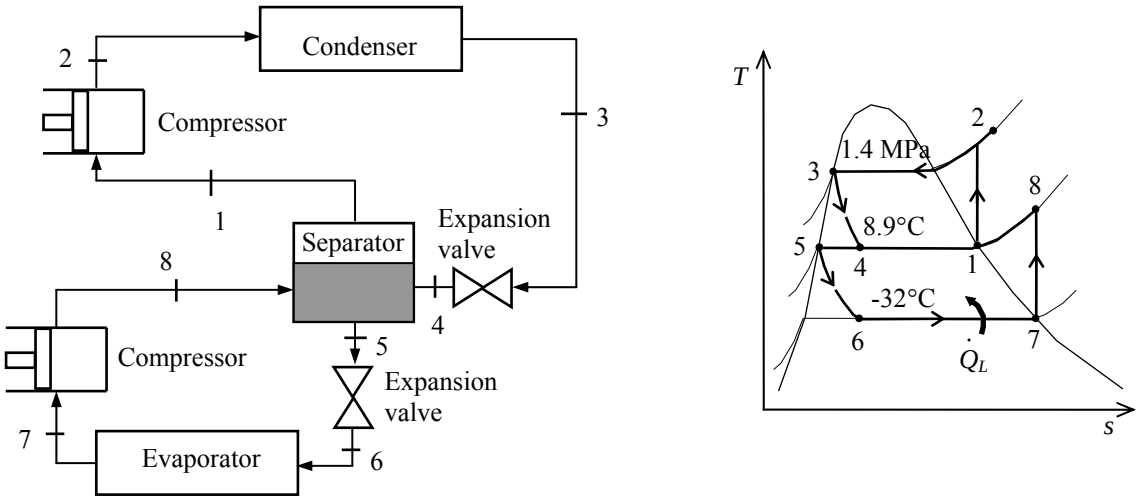
$$\dot{X}_{\text{destroyed},12} = 0$$

The greatest exergy destruction occurs during the mixing process. Note that heat is absorbed in evaporator 2 from a reservoir at -15°F (445 R), in evaporator 1 from a reservoir at 40°F (500 R), and rejected to a reservoir at 80°F (540 R), which is also taken as the dead state temperature.

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11-123 A two-stage compression refrigeration system with a separation unit is considered. The rate of cooling and the power requirement are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From the refrigerant tables (Tables A-11, A-12, and A-13),

$$\left.\begin{array}{l} T_1 = 8.9°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,8.9°\text{C}} = 255.55\ \text{kJ/kg} \\ s_1 = s_{g\,@\,8.9°\text{C}} = 0.92691\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1400\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 281.49\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1400\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\,@\,1400\,\text{kPa}} = 127.22\ \text{kJ/kg}$$

$$h_4 \cong h_3 = 127.22\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_5 = 8.9°\text{C} \\ \text{sat. liquid} \end{array}\right\} \quad h_5 = h_{f\,@\,8.9°\text{C}} = 63.94\ \text{kJ/kg}$$

$$h_6 \cong h_5 = 63.94\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_7 = -32°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_7 = h_{g\,@\,-32°\text{C}} = 230.91\ \text{kJ/kg} \\ s_7 = s_{g\,@\,-32°\text{C}} = 0.95813\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = P_{\text{sat}\,@\,8.9°\text{C}} = 400\ \text{kPa} \\ s_8 = s_7 \end{array}\right\} \quad h_8 = 264.51\ \text{kJ/kg}$$

An energy balance on the separator gives

$$\dot{m}_6(h_8 - h_5) = \dot{m}_2(h_1 - h_4) \longrightarrow \dot{m}_6 = \dot{m}_2\frac{h_1 - h_4}{h_8 - h_5} = (2\ \text{kg/s})\frac{255.55 - 127.22}{264.51 - 63.94} = 1.280\ \text{kg/s}$$

The rate of cooling produced by this system is then

$$\dot{Q}_L = \dot{m}_6(h_7 - h_6) = (1.280\ \text{kg/s})(230.91 - 63.94)\ \text{kJ/kg} = \mathbf{213.7\ kJ/s}$$

The total power input to the compressors is

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{m}_6(h_8 - h_7) + \dot{m}_2(h_2 - h_1) \\ &= (1.280\ \text{kg/s})(264.51 - 230.91)\ \text{kJ/kg} + (2\ \text{kg/s})(281.49 - 255.55)\ \text{kJ/kg} = \mathbf{94.89\ kW} \end{aligned}$$

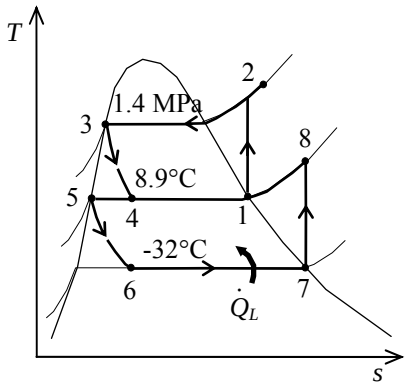
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11-124 A two-stage vapor-compression refrigeration system with refrigerant-134a as the working fluid is considered. The process with the greatest exergy destruction is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis From Prob. 11-109 and the refrigerant tables (Tables A-11, A-12, and A-13),

$$s_1 = s_2 = 0.92691\ \text{kJ/kg}\cdot\text{K}$$
$$s_3 = 0.45315\ \text{kJ/kg}\cdot\text{K}$$
$$s_4 = 0.4720\ \text{kJ/kg}\cdot\text{K}$$
$$s_5 = 0.24761\ \text{kJ/kg}\cdot\text{K}$$
$$s_6 = 0.2658\ \text{kJ/kg}\cdot\text{K}$$
$$s_7 = s_8 = 0.95813\ \text{kJ/kg}\cdot\text{K}$$
$$\dot{m}_{\text{upper}} = 2\ \text{kg/s}$$
$$\dot{m}_{\text{lower}} = 1.280\ \text{kg/s}$$
$$q_L = h_7 - h_6 = 166.97\ \text{kJ/kg}$$
$$q_H = h_2 - h_3 = 154.27\ \text{kJ/kg}$$
$$T_L = -18 + 273 = 255\ \text{K}$$
$$T_H = T_0 = 25 + 273 = 298\ \text{K}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$\dot{X}_{\text{destroyed},23} = \dot{m}_{\text{upper}} T_0\left(s_3 - s_2 + \frac{q_H}{T_H}\right)$$
$$= (2\ \text{kg/s})(298\ \text{K})\left(0.45315 - 0.92691 + \frac{154.27\ \text{kJ/kg}}{298\ \text{K}}\right) = \mathbf{26.18\ kW}$$
$$\dot{X}_{\text{destroyed},34} = \dot{m}_{\text{upper}} T_0 (s_4 - s_3) = (2\ \text{kg/s})(298\ \text{K})(0.4720 - 0.45315)\ \text{kJ/kg}\cdot\text{K} = 11.23\ \text{kW}$$
$$\dot{X}_{\text{destroyed},56} = \dot{m}_{\text{lower}} T_0 (s_6 - s_5) = (1.280\ \text{kg/s})(298\ \text{K})(0.2658 - 0.24761)\ \text{kJ/kg}\cdot\text{K} = 6.94\ \text{kW}$$
$$\dot{X}_{\text{destroyed},67} = \dot{m}_{\text{lower}} T_0\left(s_7 - s_6 - \frac{q_L}{T_L}\right)$$
$$= (1.280\ \text{kg/s})(298\ \text{K})\left(0.95813 - 0.2658 - \frac{166.97\ \text{kJ/kg}}{255\ \text{K}}\right) = 14.32\ \text{kW}$$
$$\dot{X}_{\text{destroyed, separator}} = T_0\left[\dot{m}_{\text{lower}}(s_5 - s_8) + \dot{m}_{\text{upper}}(s_1 - s_4)\right]$$
$$= (298\ \text{K})\left[(1.280\ \text{kg/s})(0.24761 - 0.95813) + (2\ \text{kg/s})(0.92691 - 0.4720)\right] = 0.11\ \text{kW}$$

For isentropic processes, the exergy destruction is zero:

$$\dot{X}_{\text{destroyed},12} = 0$$
$$\dot{X}_{\text{destroyed},78} = 0$$

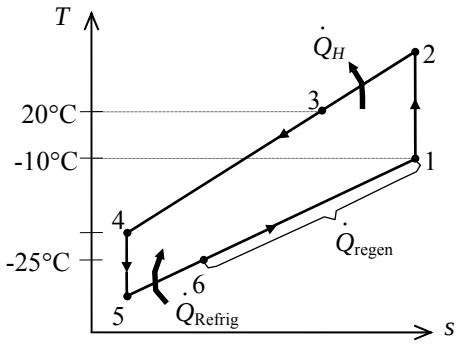
Note that heat is absorbed from a reservoir at 0°F (460 R) and rejected to the standard ambient air at 77°F (537 R), which is also taken as the dead state temperature. The greatest exergy destruction occurs during the condensation process.

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11-125 A regenerative gas refrigeration cycle with helium as the working fluid is considered. The temperature of the helium at the turbine inlet, the COP of the cycle, and the net power input required are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Helium is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of helium are $c_p = 5.1926$ kJ/kg·K and $k = 1.667$ (Table A-2).



Analysis (*a*) The temperature of the helium at the turbine inlet is determined from an energy balance on the regenerator,

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\nearrow 0\,(steady)} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}(h_3 - h_4) = \dot{m}(h_1 - h_6)$$

or,

$$\dot{m} c_p (T_3 - T_4) = \dot{m} c_p (T_1 - T_6) \longrightarrow T_3 - T_4 = T_1 - T_6$$

Thus,

$$T_4 = T_3 - T_1 + T_6 = 20°\text{C} - (-10°\text{C}) + (-25°\text{C}) = \mathbf{5°C} = 278\ \text{K}$$

(*b*) From the isentropic relations,

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (263\ \text{K})(3)^{0.667/1.667} = 408.2\ \text{K} = 135.2°\text{C}$$

$$T_5 = T_4 \left(\frac{P_5}{P_4} \right)^{(k-1)/k} = (278\ \text{K}) \left(\frac{1}{3} \right)^{0.667/1.667} = 179.1\ \text{K} = -93.9°\text{C}$$

Then the COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_R = \frac{q_L}{w_{net,in}} = \frac{q_L}{w_{comp,in} - w_{turb,out}} = \frac{h_6 - h_5}{(h_2 - h_1) - (h_4 - h_5)}$$

$$= \frac{T_6 - T_5}{(T_2 - T_1) - (T_4 - T_5)} = \frac{-25°\text{C} - (-93.9°\text{C})}{[135.2 - (-10)]°\text{C} - [5 - (-93.9)]°\text{C}} = \mathbf{1.49}$$

(*c*) The net power input is determined from

$$\dot{W}_{net,in} = \dot{W}_{comp,in} - \dot{W}_{turb,out} = \dot{m}[(h_2 - h_1) - (h_4 - h_5)]$$

$$= \dot{m} c_p [(T_2 - T_1) - (T_4 - T_5)]$$

$$= (0.45\ \text{kg/s})(5.1926\ \text{kJ/kg} \cdot °\text{C})([135.2 - (-10)] - [5 - (-93.9)])$$

$$= \mathbf{108.2\ kW}$$

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11-126 An absorption refrigeration system operating at specified conditions is considered. The minimum rate of heat supply required is to be determined.

Analysis The maximum COP that this refrigeration system can have is

$$\text{COP}_{\text{R,max}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{298\,\text{K}}{368\,\text{K}}\right)\left(\frac{275}{298 - 275}\right) = 2.274$$

Thus,

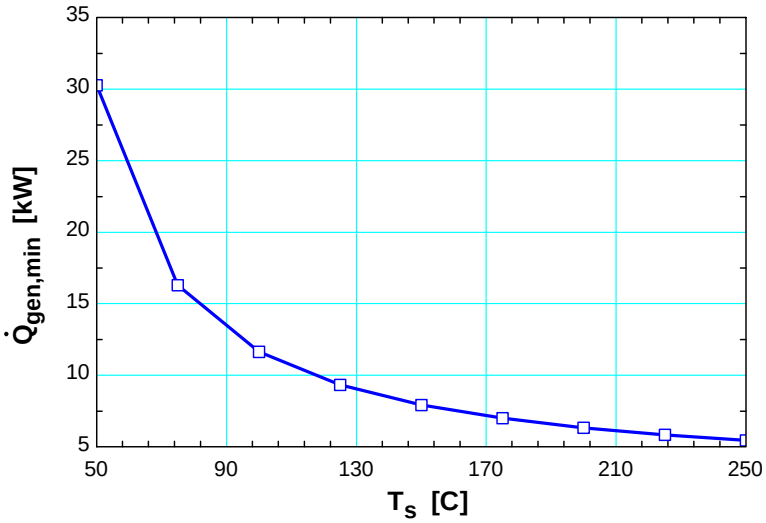
$$\dot{Q}_{\text{gen,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{28\,\text{kW}}{2.274} = \mathbf{12.3\,kW}$$

11-127 Problem 11-126 is reconsidered. The effect of the source temperature on the minimum rate of heat supply is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data:"
T_L = 2 [C]
T_0 = 25 [C]
T_s = 95 [C]
Q_dot_L = 28 [kW]
"The maximum COP that this refrigeration system can have is:"
COP_R_max = (1-(T_0+273)/(T_s+273))*((T_L+273)/(T_0 - T_L))
"The minimum rate of heat supply is:"
Q_dot_gen_min = Q_dot_L/COP_R_max
```

T_s [C]	Q_{genmin} [kW]
50	30.26
75	16.3
100	11.65
125	9.32
150	7.925
175	6.994
200	6.33
225	5.831
250	5.443



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11-128 A regenerative gas refrigeration cycle using air as the working fluid is considered. The effectiveness of the regenerator, the rate of heat removal from the refrigerated space, the COP of the cycle, and the refrigeration load and the COP if this system operated on the simple gas refrigeration cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) For this problem, we use the properties of air from EES. Note that for an ideal gas enthalpy is a function of temperature only while entropy is functions of both temperature and pressure.

$$T_1 = 0°\text{C} \longrightarrow h_1 = 273.40\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_1 = 100\ \text{kPa} \\ T_1 = 0°\text{C} \end{matrix}\right\} s_1 = 5.6110\ \text{kJ/kg.K}$$

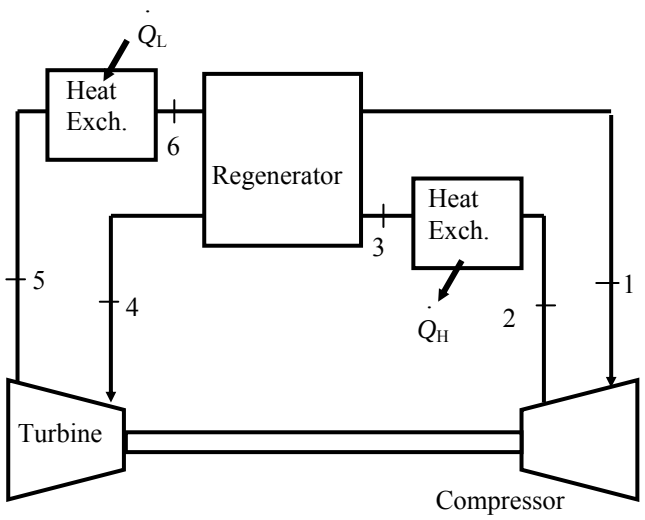
$$\left.\begin{matrix} P_2 = 500\ \text{kPa} \\ s_2 = s_1 \end{matrix}\right\} h_{2s} = 433.50\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.80 = \frac{433.50 - 273.40}{h_2 - 273.40}$$

$$h_2 = 473.52\ \text{kJ/kg}$$

$$T_3 = 35°\text{C} \longrightarrow h_3 = 308.63\ \text{kJ/kg}$$



For the turbine inlet and exit we have

$$T_5 = -80°\text{C} \longrightarrow h_5 = 193.45\ \text{kJ/kg}$$

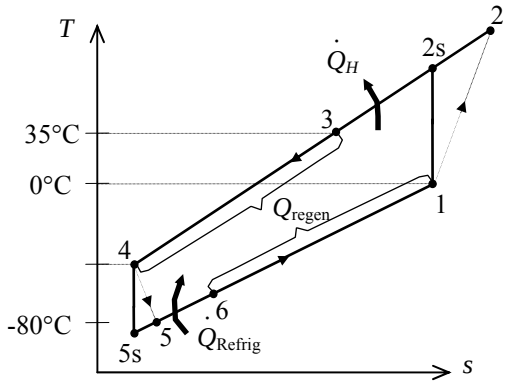
$$T_4 = ? \longrightarrow h_4 =$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}}$$

$$\left.\begin{matrix} P_1 = 100\ \text{kPa} \\ T_1 = 0°\text{C} \end{matrix}\right\} s_1 = 5.6110\ \text{kJ/kg.K}$$

$$\left.\begin{matrix} P_4 = 500\ \text{kPa} \\ T_4 = ? \end{matrix}\right\} s_4 =$$

$$\left.\begin{matrix} P_5 = 500\ \text{kPa} \\ s_5 = s_4 \end{matrix}\right\} h_{5s} =$$



We can determine the temperature at the turbine inlet from EES using the above relations. A hand solution would require a trial-error approach.

$$T_4 = 281.8\ \text{K},\ \ h_4 = 282.08\ \text{kJ/kg}$$

An energy balance on the regenerator gives

$$h_6 = h_1 - h_3 + h_4 = 273.40 - 308.63 + 282.08 = 246.85\ \text{kJ/kg}$$

The effectiveness of the regenerator is determined from

$$\varepsilon_{\text{regen}} = \frac{h_3 - h_4}{h_3 - h_6} = \frac{308.63 - 282.08}{308.63 - 246.85} = \mathbf{0.430}$$

(*b*) The refrigeration load is

$$\dot{Q}_L = \dot{m}(h_6 - h_5) = (0.4\ \text{kg/s})(246.85 - 193.45)\text{kJ/kg} = \mathbf{21.36\ kW}$$

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(*c*) The turbine and compressor powers and the COP of the cycle are

$$\dot{W}_{\text{C,in}} = \dot{m}(h_2 - h_1) = (0.4\ \text{kg/s})(473.52 - 273.40)\text{kJ/kg} = 80.05\ \text{kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}(h_4 - h_5) = (0.4\ \text{kg/s})(282.08 - 193.45)\text{kJ/kg} = 35.45\ \text{kW}$$

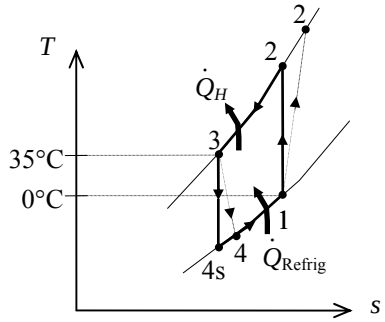
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{\dot{Q}_L}{\dot{W}_{\text{C,in}} - \dot{W}_{\text{T,out}}} = \frac{21.36}{80.05 - 35.45} = \mathbf{0.479}$$

(*d*) The simple gas refrigeration cycle analysis is as follows:

$$h_1 = 273.40\ \text{kJ/kg}$$
$$h_2 = 473.52\ \text{kJ/kg}$$
$$h_3 = 308.63\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 500\ \text{kPa} \\ T_3 = 35°\text{C} \end{matrix}\right\} s_3 = 5.2704\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_1 = 100\ \text{kPa} \\ s_4 = s_3 \end{matrix}\right\} h_{4s} = 194.52\ \text{kJ/kg.K}$$



$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.85 = \frac{308.63 - h_4}{308.63 - 194.52} \longrightarrow h_4 = 211.64\ \text{kJ/kg}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.4\ \text{kg/s})(273.40 - 211.64)\text{kJ/kg} = \mathbf{24.70\ kW}$$

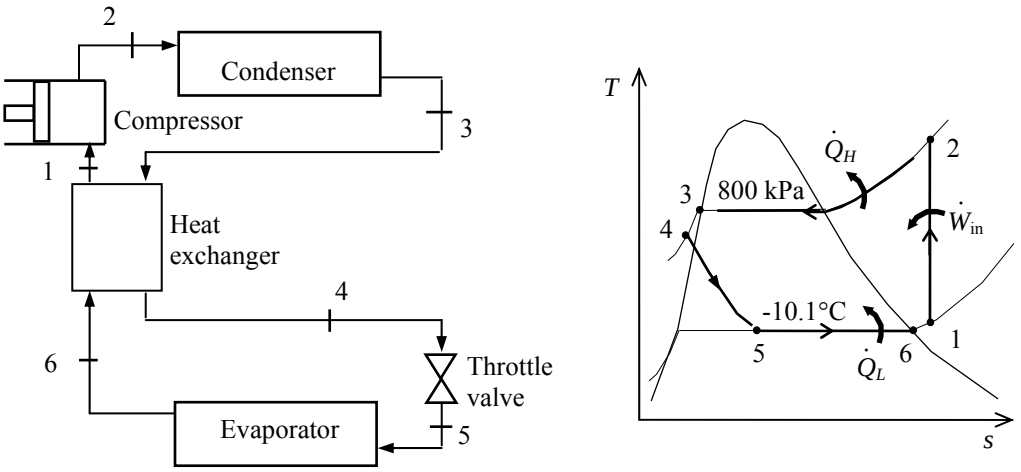
$$\dot{W}_{\text{net,in}} = \dot{m}(h_2 - h_1) - \dot{m}(h_3 - h_4) = (0.4\ \text{kg/s})\big[(473.52 - 273.40) - (308.63 - 211.64)\text{kJ/kg}\big] = 41.25\ \text{kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{24.70}{41.25} = \mathbf{0.599}$$

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11-129 An innovative vapor-compression refrigeration system with a heat exchanger is considered. The system's COP is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.



Analysis From the refrigerant tables (Tables A-11, A-12, and A-13),

$$\left.\begin{array}{l} P_3 = 800\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \ h_3 = h_{f\,@\,800\,\text{kPa}} = 95.47\ \text{kJ/kg}$$

$$\left.\begin{array}{l} T_4 = T_{\text{sat}\,@\,800\,\text{kPa}} - 11.3 \\ \quad = 31.3 - 11.3 = 20°\text{C} \\ P_4 = 800\ \text{kPa} \end{array}\right\} \ h_4 \cong h_{f\,@\,20°\text{C}} = 79.32\ \text{kJ/kg}$$

$$h_5 \cong h_4 = 79.32\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_6 = -10.1°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_6 = h_{g\,@\,-10.1°\text{C}} = 244.46\ \text{kJ/kg} \\ P_6 = P_{\text{sat}\,@\,-10.1°\text{C}} = 200\ \text{kPa} \end{array}$$

An energy balance on the heat exchanger gives

$$\dot{m}(h_1 - h_6) = \dot{m}(h_3 - h_4) \longrightarrow h_1 = h_3 - h_4 + h_6 = 95.47 - 79.32 + 244.46 = 260.61\ \text{kJ/kg}$$

Then,

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ h_1 = 260.61\ \text{kJ/kg} \end{array}\right\} \ s_1 = 0.9970\ \text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} \ h_2 = 292.17\ \text{kJ/kg}$$

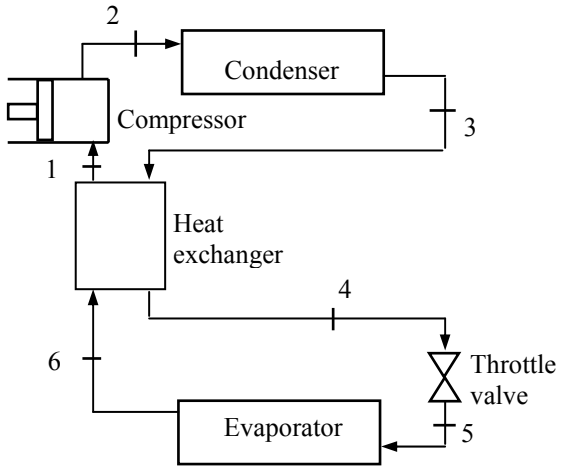
The COP of this refrigeration system is determined from its definition,

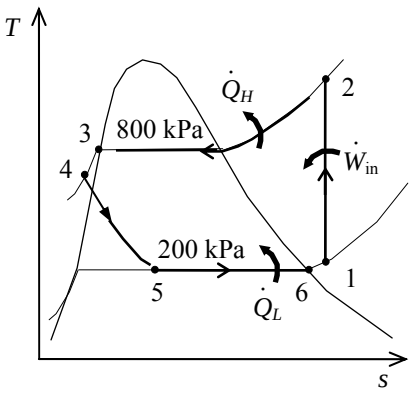
$$\text{COP}_R = \frac{q_L}{w_{\text{in}}} = \frac{h_6 - h_5}{h_2 - h_1} = \frac{244.46 - 79.32}{292.17 - 260.61} = \mathbf{5.23}$$

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11-130 An innovative vapor-compression refrigeration system with a heat exchanger is considered. The system's COP is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.





Analysis From the refrigerant tables (Tables A-11, A-12, and A-13),

$$\left.\begin{array}{l} P_3 = 800\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} \ h_3 = h_{f\ @\ 800\ \text{kPa}} = 95.47\ \text{kJ/kg}$$

$$\left.\begin{array}{l} T_4 = T_{\text{sat}\ @\ 800\ \text{kPa}} - 21.3 \\ \quad = 31.3 - 21.3 = 10°\text{C} \\ P_4 = 800\ \text{kPa} \end{array}\right\} \ h_4 \cong h_{f\ @\ 10°\text{C}} = 65.43\ \text{kJ/kg}$$

$$h_5 \cong h_4 = 65.43\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_6 = -10.1°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_6 = h_{g\ @\ -10.1°\text{C}} = 244.46\ \text{kJ/kg} \\ P_6 = P_{\text{sat}\ @\ -10.1°\text{C}} = 200\ \text{kPa} \end{array}$$

An energy balance on the heat exchanger gives

$$\dot{m}(h_1 - h_6) = \dot{m}(h_3 - h_4) \longrightarrow h_1 = h_3 - h_4 + h_6 = 95.47 - 65.43 + 244.46 = 274.50\ \text{kJ/kg}$$

Then,

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ h_1 = 274.50\ \text{kJ/kg} \end{array}\right\} \ s_1 = 1.0449\ \text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} \ h_2 = 308.28\ \text{kJ/kg}$$

The COP of this refrigeration system is determined from its definition,

$$\text{COP}_R = \frac{q_L}{w_{\text{in}}} = \frac{h_6 - h_5}{h_2 - h_1} = \frac{244.46 - 65.43}{308.28 - 274.50} = \mathbf{5.30}$$

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11-131 An ideal gas refrigeration cycle with with three stages of compression with intercooling using air as the working fluid is considered. The COP of this system is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

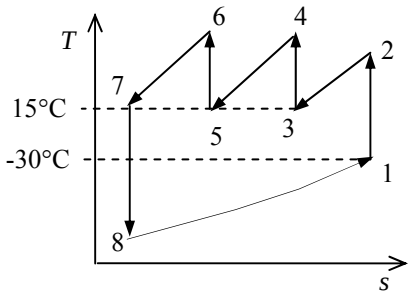
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis From the isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (243\text{ K})(7)^{0.4/1.4} = 423.7\text{ K}$$

$$T_4 = T_6 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (288\text{ K})(7)^{0.4/1.4} = 502.2\text{ K}$$

$$T_8 = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (288\text{ K})\left(\frac{1}{7\times 7\times 7}\right)^{0.4/1.4} = 54.3\text{ K}$$



The COP of this ideal gas refrigeration cycle is determined from

$$\begin{aligned}\text{COP}_\text{R} &= \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}} \\ &= \frac{h_1 - h_8}{(h_2 - h_1) + (h_4 - h_3) + (h_6 - h_5) - (h_7 - h_8)} \\ &= \frac{T_1 - T_8}{(T_2 - T_1) + 2(T_4 - T_3) - (T_7 - T_8)} \\ &= \frac{243 - 54.3}{(423.7 - 243) + 2(502.2 - 288) - (288 - 54.3)} \\ &= \mathbf{0.503}\end{aligned}$$

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11-132 A vortex tube receives compressed air at 500 kPa and 300 K, and supplies 25 percent of it as cold air and the rest as hot air. The COP of the vortex tube is to be compared to that of a reversed Brayton cycle for the same pressure ratio; the exit temperature of the hot fluid stream and the COP are to be determined; and it is to be shown if this process violates the second law.

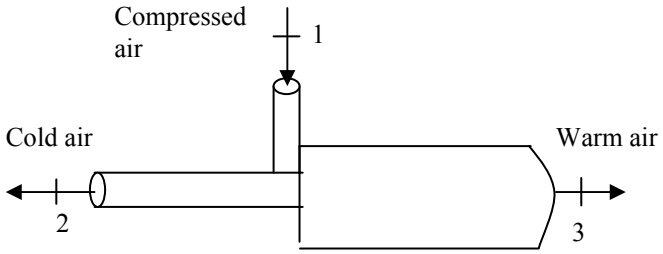
Assumptions **1** The vortex tube is adiabatic. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Steady operating conditions exist.

Properties The gas constant of air is 0.287 kJ/kg.K (Table A-1). The specific heat of air at room temperature is c_p = 1.005 kJ/kg.K (Table A-2). The enthalpy of air at absolute temperature T can be expressed in terms of specific heats as $h = c_pT$.

Analysis (*a*) The COP of the vortex tube is much lower than the COP of a reversed Brayton cycle of the same pressure ratio since the vortex tube involves *vortices*, which are highly irreversible. Owing to this irreversibility, the minimum temperature that can be obtained by the vortex tube is not as low as the one that can be obtained by the revered Brayton cycle.

(*b*) We take the vortex tube as the system. This is a steady flow system with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow energy balance equation for this system $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ for a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$ can be expressed as

$$\begin{aligned} \dot{m}_1 h_1 &= \dot{m}_2 h_2 + \dot{m}_3 h_3 \\ \dot{m}_1 c_p T_1 &= \dot{m}_2 c_p T_2 + \dot{m}_3 c_p T_3 \\ 1 c_p T_1 &= 0.25 c_p T_2 + 0.75 c_p T_3 \end{aligned}$$



Canceling c_p and solving for T_3 gives

$$T_3 = \frac{T_1 - 0.25T_2}{0.75} = \frac{300 - 0.25 \times 278}{0.75} = \mathbf{307.3\ K}$$

Therefore, the hot air stream will leave the vortex tube at an average temperature of 307.3 K.

(*c*) The entropy balance for this steady flow system $\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = 0$ can be expressed as with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow entropy balance equation for this system for a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$ can be expressed

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{S}_{\text{out}} - \dot{S}_{\text{in}} \\ &= \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_1 s_1 = \dot{m}_2 s_2 + \dot{m}_3 s_3 - (\dot{m}_2 + \dot{m}_3) s_1 \\ &= \dot{m}_2 (s_2 - s_1) + \dot{m}_3 (s_3 - s_1) \\ &= 0.25(s_2 - s_1) + 0.75(s_3 - s_1) \\ &= 0.25\left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) + 0.75\left(c_p \ln \frac{T_3}{T_1} - R \ln \frac{P_3}{P_1} \right) \end{aligned}$$

Substituting the known quantities, the rate of entropy generation is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= 0.25\left((1.005 \text{ kJ/kg.K}) \ln \frac{278 \text{ K}}{300 \text{ K}} - (0.287 \text{ kJ/kg.K}) \ln \frac{100 \text{ kPa}}{500 \text{ kPa}} \right) \\ &\quad + 0.75\left((1.005 \text{ kJ/kg.K}) \ln \frac{307.3 \text{ K}}{300 \text{ K}} - (0.287 \text{ kJ/kg.K}) \ln \frac{100 \text{ kPa}}{500 \text{ kPa}} \right) \\ &= 0.461 \text{ kW/K} > 0 \end{aligned}$$

which is a positive quantity. Therefore, this process **satisfies** the 2nd law of thermodynamics.

(*d*) For a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$, the cooling rate and the power input to the compressor are determined to

$$\begin{aligned} \dot{Q}_{\text{cooling}} &= \dot{m}_c (h_1 - h_c) = \dot{m}_c c_p (T_1 - T_c) \\ &= (0.25 \text{ kg/s})(1.005 \text{ kJ/kg.K})(300 - 278)\text{K} = 5.53 \text{ kW} \end{aligned}$$

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$$\dot{W}_{\text{comp,in}} = \frac{\dot{m}_0 R T_0}{(k-1)\eta_{\text{comp}}}\left[\left(\frac{P_1}{P_0}\right)^{(k-1)/k} - 1\right]$$

$$= \frac{(1\,\text{kg/s})(0.287\,\text{kJ/kg.K})(300\,\text{K})}{(1.4-1)0.80}\left[\left(\frac{500\,\text{kPa}}{100\,\text{kPa}}\right)^{(1.4-1)/1.4} - 1\right] = 157.1\,\text{kW}$$

Then the COP of the vortex refrigerator becomes

$$\text{COP} = \frac{\dot{Q}_{\text{cooling}}}{\dot{W}_{\text{comp, in}}} = \frac{5.53\,\text{kW}}{157.1\,\text{kW}} = \mathbf{0.035}$$

The COP of a Carnot refrigerator operating between the same temperature limits of 300 K and 278 K is

$$\text{COP}_{\text{Carnot}} = \frac{T_L}{T_H - T_L} = \frac{278\ \text{K}}{(300-278)\ \text{K}} = \mathbf{12.6}$$

Discussion Note that the COP of the vortex refrigerator is a small fraction of the COP of a Carnot refrigerator operating between the same temperature limits.

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11-133 A vortex tube receives compressed air at 600 kPa and 300 K, and supplies 25 percent of it as cold air and the rest as hot air. The COP of the vortex tube is to be compared to that of a reversed Brayton cycle for the same pressure ratio; the exit temperature of the hot fluid stream and the COP are to be determined; and it is to be shown if this process violates the second law.

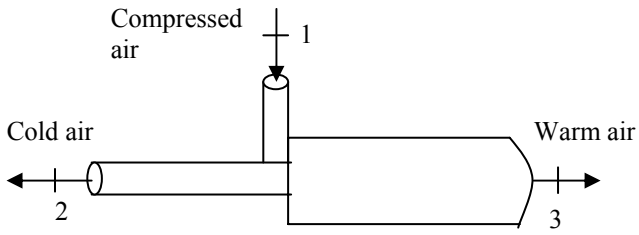
Assumptions **1** The vortex tube is adiabatic. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Steady operating conditions exist.

Properties The gas constant of air is 0.287 kJ/kg.K (Table A-1). The specific heat of air at room temperature is c_p = 1.005 kJ/kg.K (Table A-2). The enthalpy of air at absolute temperature T can be expressed in terms of specific heats as $h = c_p T$.

Analysis (*a*) The COP of the vortex tube is much lower than the COP of a reversed Brayton cycle of the same pressure ratio since the vortex tube involves *vortices*, which are highly irreversible. Owing to this irreversibility, the minimum temperature that can be obtained by the vortex tube is not as low as the one that can be obtained by the revered Brayton cycle.

(*b*) We take the vortex tube as the system. This is a steady flow system with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow entropy balance equation for this system $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ for a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$ can be expressed as

$$\begin{aligned} \dot{m}_1 h_1 &= \dot{m}_2 h_2 + \dot{m}_3 h_3 \\ \dot{m}_1 c_p T_1 &= \dot{m}_2 c_p T_2 + \dot{m}_3 c_p T_3 \\ 1 c_p T_1 &= 0.25 c_p T_2 + 0.75 c_p T_3 \end{aligned}$$



Canceling c_p and solving for T_3 gives

$$T_3 = \frac{T_1 - 0.25T_2}{0.75} = \frac{300 - 0.25 \times 278}{0.75} = \mathbf{307.3\ K}$$

Therefore, the hot air stream will leave the vortex tube at an average temperature of 307.3 K.

(*c*) The entropy balance for this steady flow system $\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = 0$ can be expressed as with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow energy balance equation for this system for a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$ can be expressed

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{S}_{\text{out}} - \dot{S}_{\text{in}} \\ &= \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_1 s_1 = \dot{m}_2 s_2 + \dot{m}_3 s_3 - (\dot{m}_2 + \dot{m}_3) s_1 \\ &= \dot{m}_2 (s_2 - s_1) + \dot{m}_3 (s_3 - s_1) \\ &= 0.25(s_2 - s_1) + 0.75(s_3 - s_1) \\ &= 0.25\left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) + 0.75\left(c_p \ln \frac{T_3}{T_1} - R \ln \frac{P_3}{P_1} \right) \end{aligned}$$

Substituting the known quantities, the rate of entropy generation is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= 0.25\left((1.005 \text{ kJ/kg.K}) \ln \frac{278 \text{ K}}{300 \text{ K}} - (0.287 \text{ kJ/kg.K}) \ln \frac{100 \text{ kPa}}{600 \text{ kPa}} \right) \\ &\quad + 0.75\left((1.005 \text{ kJ/kg.K}) \ln \frac{307.3 \text{ K}}{300 \text{ K}} - (0.287 \text{ kJ/kg.K}) \ln \frac{100 \text{ kPa}}{600 \text{ kPa}} \right) \\ &= 0.513 \text{ kW/K} > 0 \end{aligned}$$

which is a positive quantity. Therefore, this process **satisfies** the 2nd law of thermodynamics.

(*d*) For a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$, the cooling rate and the power input to the compressor are determined to

$$\begin{aligned} \dot{Q}_{\text{cooling}} &= \dot{m}_c (h_1 - h_{\text{c}}) = \dot{m}_c c_p (T_1 - T_{\text{c}}) \\ &= (0.25 \text{ kg/s})(1.005 \text{ kJ/kg.K})(300 - 278)\text{K} = 5.53 \text{ kW} \end{aligned}$$

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$$\dot{W}_{\text{comp,in}} = \frac{\dot{m}_0 RT_0}{(k-1)\eta_{\text{comp}}}\left[\left(\frac{P_1}{P_0}\right)^{(k-1)/k} - 1\right]$$

$$= \frac{(1\,\text{kg/s})(0.287\,\text{kJ/kg.K})(300\,\text{K})}{(1.4-1)0.80}\left[\left(\frac{600\,\text{kPa}}{100\,\text{kPa}}\right)^{(1.4-1)/1.4} - 1\right] = 179.9\,\text{kW}$$

Then the COP of the vortex refrigerator becomes

$$\text{COP} = \frac{\dot{Q}_{\text{cooling}}}{\dot{W}_{\text{comp, in}}} = \frac{5.53\,\text{kW}}{179.9\,\text{kW}} = \mathbf{0.031}$$

The COP of a Carnot refrigerator operating between the same temperature limits of 300 K and 278 K is

$$\text{COP}_{\text{Carnot}} = \frac{T_L}{T_H - T_L} = \frac{278\,\text{K}}{(300-278)\,\text{K}} = \mathbf{12.6}$$

Discussion Note that the COP of the vortex refrigerator is a small fraction of the COP of a Carnot refrigerator operating between the same temperature limits.

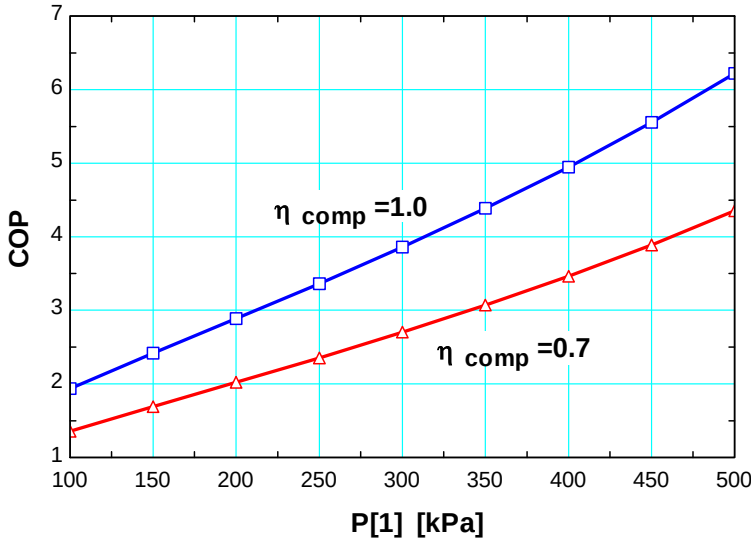
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11-134 The effect of the evaporator pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[1]=100 [kPa]
P[2] = 1400 [kPa]
Fluid$='R134a'
Eta_c=1.0 "Compressor isentropic efficiency"
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
W_c=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+W_c=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])
"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
"Coefficient of Performance:"
COP=Q_in/W_c "definition of COP"
```

P_1 [kPa]	COP	η_c
100	1.937	1
150	2.417	1
200	2.886	1
250	3.363	1
300	3.859	1
350	4.384	1
400	4.946	1
450	5.555	1
500	6.22	1



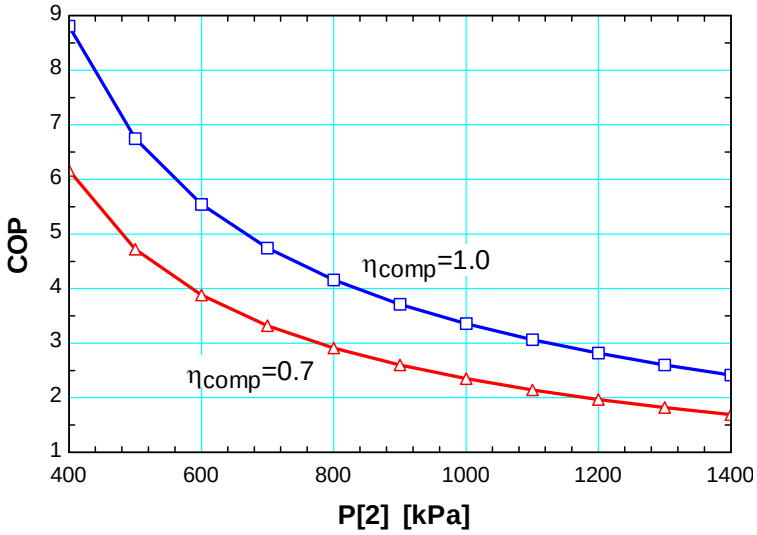
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11-135 The effect of the condenser pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[1]=150 [kPa]
P[2] = 400 [kPa]
Fluid$='R134a'
Eta_c=0.7 "Compressor isentropic efficiency"
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
W_c=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+W_c=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])
"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
"Coefficient of Performance:"
COP=Q_in/W_c "definition of COP"
```

P_2 [kPa]	COP	η_c
400	6.162	0.7
500	4.722	0.7
600	3.881	0.7
700	3.32	0.7
800	2.913	0.7
900	2.6	0.7
1000	2.351	0.7
1100	2.145	0.7
1200	1.971	0.7
1300	1.822	0.7
1400	1.692	0.7



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11-136 A relation for the COP of the two-stage refrigeration system with a flash chamber shown in Fig. 11-12 is to be derived.

Analysis The coefficient of performance is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{in}}$$

where

$$q_L = (1 - x_6)(h_1 - h_8) \quad \text{with} \quad x_6 = \frac{h_6 - h_f}{h_{fg}}$$

$$w_\text{in} = w_\text{compI,in} + w_\text{compII,in} = (1 - x_6)(h_2 - h_1) + (1)(h_4 - h_9)$$

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Fundamentals of Engineering (FE) Exam Problems

11-137 Consider a heat pump that operates on the reversed Carnot cycle with R-134a as the working fluid executed under the saturation dome between the pressure limits of 140 kPa and 800 kPa. R-134a changes from saturated vapor to saturated liquid during the heat rejection process. The net work input for this cycle is

(a) 28 kJ/kg (b) 34 kJ/kg (c) 49 kJ/kg (d) 144 kJ/kg (e) 275 kJ/kg

Answer (a) 28 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=800 "kPa"
P2=140 "kPa"
h_fg=ENTHALPY(R134a,x=1,P=P1)-ENTHALPY(R134a,x=0,P=P1)
TH=TEMPERATURE(R134a,x=0,P=P1)+273
TL=TEMPERATURE(R134a,x=0,P=P2)+273
q_H=h_fg
COP=TH/(TH-TL)
w_net=q_H/COP

"Some Wrong Solutions with Common Mistakes:"
W1_work = q_H/COP1; COP1=TL/(TH-TL) "Using COP of regrigerator"
W2_work = q_H/COP2; COP2=(TH-273)/(TH-TL)  "Using C instead of K"
W3_work = h_fg3/COP; h_fg3= ENTHALPY(R134a,x=1,P=P2)-ENTHALPY(R134a,x=0,P=P2) "Using h_fg at P2"
W4_work = q_H*TL/TH "Using the wrong relation"
```

11-138 A refrigerator removes heat from a refrigerated space at 0°C at a rate of 2.2 kJ/s and rejects it to an environment at 20°C. The minimum required power input is

(a) 89 W (b) 150 W (c) 161 W (d) 557 W (e) 2200 W

Answer (c) 161 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TH=20+273
TL=0+273
Q_L=2.2 "kJ/s"
COP_max=TL/(TH-TL)
w_min=Q_L/COP_max

"Some Wrong Solutions with Common Mistakes:"
W1_work = Q_L/COP1; COP1=TH/(TH-TL) "Using COP of heat pump"
W2_work = Q_L/COP2; COP2=(TH-273)/(TH-TL)  "Using C instead of K"
W3_work = Q_L*TL/TH "Using the wrong relation"
W4_work = Q_L "Taking the rate of refrigeration as power input"
```

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11-139 A refrigerator operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 120 kPa and 800 kPa. If the rate of heat removal from the refrigerated space is 32 kJ/s, the mass flow rate of the refrigerant is

(a) 0.19 kg/s (b) 0.15 kg/s (c) 0.23 kg/s (d) 0.28 kg/s (e) 0.81 kg/s

Answer (c) 0.23 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=120 "kPa"
P2=800 "kPa"
P3=P2
P4=P1
s2=s1
Q_refrig=32 "kJ/s"
m=Q_refrig/(h1-h4)
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,s=s2,P=P2)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3

"Some Wrong Solutions with Common Mistakes:"
W1_mass = Q_refrig/(h2-h1) "Using wrong enthalpies, for W_in"
W2_mass = Q_refrig/(h2-h3) "Using wrong enthalpies, for Q_H"
W3_mass = Q_refrig/(h1-h44); h44=ENTHALPY(R134a,x=0,P=P4)  "Using wrong enthalpy h4 (at P4)"
W4_mass = Q_refrig/h_fg; h_fg=ENTHALPY(R134a,x=1,P=P2) - ENTHALPY(R134a,x=0,P=P2)  "Using h_fg at P2"
```

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11-140 A heat pump operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 0.32 MPa and 1.2 MPa. If the mass flow rate of the refrigerant is 0.193 kg/s, the rate of heat supply by the heat pump to the heated space is

(a) 3.3 kW (b) 23 kW (c) 26 kW (d) 31 kW (e) 45 kW

Answer (d) 31 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=320 "kPa"
P2=1200 "kPa"
P3=P2
P4=P1
s2=s1
m=0.193 "kg/s"
Q_supply=m*(h2-h3) "kJ/s"
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,s=s2,P=P2)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3

"Some Wrong Solutions with Common Mistakes:"
W1_Qh = m*(h2-h1) "Using wrong enthalpies, for W_in"
W2_Qh = m*(h1-h4) "Using wrong enthalpies, for Q_L"
W3_Qh = m*(h22-h4); h22=ENTHALPY(R134a,x=1,P=P2)  "Using wrong enthalpy h2 (hg at P2)"
W4_Qh = m*h_fg; h_fg=ENTHALPY(R134a,x=1,P=P1) - ENTHALPY(R134a,x=0,P=P1)  "Using h_fg at P1"
```

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11-141 An ideal vapor compression refrigeration cycle with R-134a as the working fluid operates between the pressure limits of 120 kPa and 700 kPa. The mass fraction of the refrigerant that is in the liquid phase at the inlet of the evaporator is

(a) 0.69 (b) 0.63 (c) 0.58 (d) 0.43 (e) 0.35

Answer (a) 0.69

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=120 "kPa"
P2=700 "kPa"
P3=P2
P4=P1
h1=ENTHALPY(R134a,x=1,P=P1)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3
x4=QUALITY(R134a,h=h4,P=P4)
liquid=1-x4

"Some Wrong Solutions with Common Mistakes:"
W1_liquid = x4 "Taking quality as liquid content"
W2_liquid = 0 "Assuming superheated vapor"
W3_liquid = 1-x4s; x4s=QUALITY(R134a,s=s3,P=P4) "Assuming isentropic expansion"
s3=ENTROPY(R134a,x=0,P=P3)
```

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11-142 Consider a heat pump that operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 0.32 MPa and 1.2 MPa. The coefficient of performance of this heat pump is

(a) 0.17 (b) 1.2 (c) 3.1 (d) 4.9 (e) 5.9

Answer (e) 5.9

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=320 "kPa"
P2=1200 "kPa"
P3=P2
P4=P1
s2=s1
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,s=s2,P=P2)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3
COP_HP=qH/Win
Win=h2-h1
qH=h2-h3

"Some Wrong Solutions with Common Mistakes:"
W1_COP = (h1-h4)/(h2-h1) "COP of refrigerator"
W2_COP = (h1-h4)/(h2-h3)  "Using wrong enthalpies, QL/QH"
W3_COP = (h22-h3)/(h22-h1); h22=ENTHALPY(R134a,x=1,P=P2)  "Using wrong enthalpy h2 (hg at P2)"
```

11-143 An ideal gas refrigeration cycle using air as the working fluid operates between the pressure limits of 80 kPa and 280 kPa. Air is cooled to 35°C before entering the turbine. The lowest temperature of this cycle is

(a) –58°C (b) -26°C (c) 0°C (d) 11°C (e) 24°C

Answer (a) –58°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
P1= 80 "kPa"
P2=280 "kPa"
T3=35+273 "K"
"Mimimum temperature is the turbine exit temperature"
T4=T3*(P1/P2)^((k-1)/k) - 273

"Some Wrong Solutions with Common Mistakes:"
W1_Tmin = (T3-273)*(P1/P2)^((k-1)/k) "Using C instead of K"
W2_Tmin = T3*(P1/P2)^((k-1)) - 273  "Using wrong exponent"
W3_Tmin = T3*(P1/P2)^k - 273  "Using wrong exponent"
```

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11-144 Consider an ideal gas refrigeration cycle using helium as the working fluid. Helium enters the compressor at 100 kPa and 17°C and is compressed to 400 kPa. Helium is then cooled to 20°C before it enters the turbine. For a mass flow rate of 0.2 kg/s, the net power input required is

(a) 28.3 kW (b) 40.5 kW (c) 64.7 kW (d) 93.7 kW (e) 113 kW

Answer (d) 93.7 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=5.1926 "kJ/kg.K"
P1= 100 "kPa"
T1=17+273 "K"
P2=400 "kPa"
T3=20+273 "K"
m=0.2 "kg/s"
"Mimimum temperature is the turbine exit temperature"
T2=T1*(P2/P1)^((k-1)/k)
T4=T3*(P1/P2)^((k-1)/k)
W_netin=m*Cp*((T2-T1)-(T3-T4))

"Some Wrong Solutions with Common Mistakes:"
W1_Win = m*Cp*((T22-T1)-(T3-T44));  T22=T1*P2/P1; T44=T3*P1/P2  "Using wrong relations for temps"
W2_Win =  m*Cp*(T2-T1) "Ignoring turbine work"
W3_Win=m*1.005*((T2B-T1)-(T3-T4B)); T2B=T1*(P2/P1)^((kB-1)/kB); T4B=T3*(P1/P2)^((kB-1)/kB); kB=1.4
"Using air properties"
W4_Win=m*Cp*((T2A-(T1-273))-(T3-273-T4A)); T2A=(T1-273)*(P2/P1)^((k-1)/k); T4A=(T3-273)*(P1/P2)^((k-1)/k) "Using C instead of K"
```

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11-145 An absorption air-conditioning system is to remove heat from the conditioned space at 20°C at a rate of 150 kJ/s while operating in an environment at 35°C. Heat is to be supplied from a geothermal source at 140°C. The minimum rate of heat supply required is

(a) 86 kJ/s (b) 21 kJ/s (c) 30 kJ/s (d) 61 kJ/s (e) 150 kJ/s

Answer (c) 30 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=20+273 "K"
Q_refrig=150 "kJ/s"
To=35+273 "K"
Ts=140+273 "K"
COP_max=(1-To/Ts)*(TL/(To-TL))
Q_in=Q_refrig/COP_max

"Some Wrong Solutions with Common Mistakes:"
W1_Qin = Q_refrig "Taking COP = 1"
W2_Qin = Q_refrig/COP2; COP2=TL/(Ts-TL)  "Wrong COP expression"
W3_Qin = Q_refrig/COP3; COP3=(1-To/Ts)*(Ts/(To-TL))  "Wrong COP expression, COP_HP"
W4_Qin = Q_refrig*COP_max "Multiplying by COP instead of dividing"
```

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11-146 Consider a refrigerator that operates on the vapor compression refrigeration cycle with R-134a as the working fluid. The refrigerant enters the compressor as saturated vapor at 160 kPa, and exits at 800 kPa and 50°C, and leaves the condenser as saturated liquid at 800 kPa. The coefficient of performance of this refrigerator is

(a) 2.6 (b) 1.0 (c) 4.2 (d) 3.2 (e) 4.4

Answer (d) 3.2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=160 "kPa"
P2=800 "kPa"
T2=50 "C"
P3=P2
P4=P1
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3
COP_R=qL/Win
Win=h2-h1
qL=h1-h4

"Some Wrong Solutions with Common Mistakes:"
W1_COP = (h2-h3)/(h2-h1) "COP of heat pump"
W2_COP = (h1-h4)/(h2-h3)  "Using wrong enthalpies, QL/QH"
W3_COP = (h1-h4)/(h2s-h1); h2s=ENTHALPY(R134a,s=s1,P=P2)  "Assuming isentropic compression"
```

11-147 ··· 11-155 Design and Essay Problems



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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 12
THERMODYNAMIC PROPERTY RELATIONS

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Partial Derivatives and Associated Relations

12-1C For functions that depend on one variable, they are identical. For functions that depend on two or more variable, the partial differential represents the change in the function with one of the variables as the other variables are held constant. The ordinary differential for such functions represents the total change as a result of differential changes in all variables.

12-2C (*a*) $(\partial x)_v = dx$; (*b*) $(\partial z)_v \le dz$; and (*c*) $dz = (\partial z)_x + (\partial z)_v$

12-3C Yes.

12-4C Yes.

12-5 Air at a specified temperature and specific volume is considered. The changes in pressure corresponding to a certain increase of different properties are to be determined.

Assumptions Air is an ideal gas.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1).

Analysis An ideal gas equation can be expressed as $P = RT/v$. Noting that R is a constant and $P = P(T, v)$,

$$dP = \left(\frac{\partial P}{\partial T}\right)_v dT + \left(\frac{\partial P}{\partial v}\right)_T dv = \frac{R\,dT}{v} - \frac{RT\,dv}{v^2}$$

(*a*) The change in T can be expressed as $dT \cong \Delta T = 300 \times 0.01 = 3.0$ K. At v = constant,

$$(dP)_v = \frac{R\,dT}{v} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(3.0\ \text{K})}{1.2\ \text{m}^3/\text{kg}} = \mathbf{0.7175\ kPa}$$

(*b*) The change in v can be expressed as $dv \cong \Delta v = 1.2 \times 0.01 = 0.012$ m^3/kg. At T = constant,

$$(dP)_T = -\frac{RT\,dv}{v^2} = -\frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\ \text{K})(0.012\ \text{m}^3/\text{kg})}{(1.2\ \text{m}^3/\text{kg})^2} = \mathbf{-0.7175\ kPa}$$

(*c*) When both v and T increases by 1%, the change in P becomes

$$dP = (dP)_v + (dP)_T = 0.7175 + (-0.7175) = \mathbf{0}$$

Thus the changes in T and v balance each other.

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12-6 Helium at a specified temperature and specific volume is considered. The changes in pressure corresponding to a certain increase of different properties are to be determined.

Assumptions Helium is an ideal gas

Properties The gas constant of helium is $R = 2.0769$ kPa·m^3/kg·K (Table A-1).

Analysis An ideal gas equation can be expressed as $P = RT/\nu$. Noting that R is a constant and $P = P(T, \nu)$,

$$dP = \left(\frac{\partial P}{\partial T}\right)_{\nu} dT + \left(\frac{\partial P}{\partial \nu}\right)_{T} d\nu = \frac{R\,dT}{\nu} - \frac{RT\,d\nu}{\nu^2}$$

(*a*) The change in T can be expressed as $dT \cong \Delta T = 300 \times 0.01 = 3.0$ K. At ν = constant,

$$(dP)_{\nu} = \frac{R\,dT}{\nu} = \frac{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(3.0\ \text{K})}{1.2\ \text{m}^3/\text{kg}} = \mathbf{5.192\ kPa}$$

(*b*) The change in ν can be expressed as $d\nu \cong \Delta\nu = 1.2 \times 0.01 = 0.012$ m^3/kg. At T = constant,

$$(dP)_T = -\frac{RT\,d\nu}{\nu^2} = \frac{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\ \text{K})(0.012\ \text{m}^3)}{(1.2\ \text{m}^3/\text{kg})^2} = \mathbf{-5.192\ kPa}$$

(*c*) When both ν and T increases by 1%, the change in P becomes

$$dP = (dP)_{\nu} + (dP)_T = 5.192 + (-5.192) = \mathbf{0}$$

Thus the changes in T and ν balance each other.

12-7 Nitrogen gas at a specified state is considered. The c_p and c_ν of the nitrogen are to be determined using Table A-18, and to be compared to the values listed in Table A-2*b*.

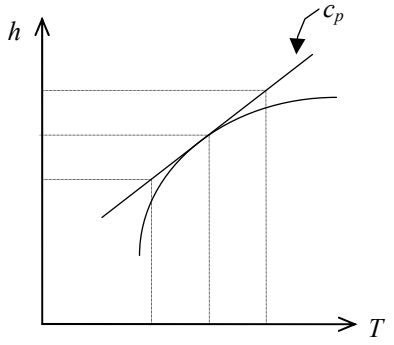
Analysis The c_p and c_ν of ideal gases depends on temperature only, and are expressed as $c_p(T) = dh(T)/dT$ and $c_\nu(T) = du(T)/dT$. Approximating the differentials as differences about 400 K, the c_p and c_ν values are determined to be

$$c_p(400\ \text{K}) = \left(\frac{dh(T)}{dT}\right)_{T=400\ \text{K}} \cong \left(\frac{\Delta h(T)}{\Delta T}\right)_{T\cong 400\ \text{K}}$$

$$= \frac{h(410\ \text{K}) - h(390\ \text{K})}{(410-390)\text{K}}$$

$$= \frac{(11{,}932 - 11{,}347)/28.0\ \text{kJ/kg}}{(410-390)\text{K}}$$

$$= \mathbf{1.045\ kJ/kg\cdot K}$$



(Compare: Table A-2b at 400 K $\rightarrow$ $c_p = 1.044$ kJ/kg·K)

$$c_\nu(400\text{K}) = \left(\frac{du(T)}{dT}\right)_{T=400\ \text{K}} \cong \left(\frac{\Delta u(T)}{\Delta T}\right)_{T\cong 400\ \text{K}}$$

$$= \frac{u(410\ \text{K}) - u(390\ \text{K})}{(410-390)\text{K}}$$

$$= \frac{(8{,}523 - 8{,}104)/28.0\ \text{kJ/kg}}{(410-390)\text{K}} = \mathbf{0.748\ kJ/kg\cdot K}$$

(Compare: Table A-2b at 400 K $\rightarrow$ $c_\nu = 0.747$ kJ/kg·K)

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12-8E Nitrogen gas at a specified state is considered. The c_p and c_v of the nitrogen are to be determined using Table A-18E, and to be compared to the values listed in Table A-2E*b*.

Analysis The c_p and c_v of ideal gases depends on temperature only, and are expressed as $c_p(T) = dh(T)/dT$ and $c_v(T) = du(T)/dT$. Approximating the differentials as differences about 600 R, the c_p and c_v values are determined to be

$$c_p(800\text{ R}) = \left(\frac{dh(T)}{dT}\right)_{T=800\text{ R}} \cong \left(\frac{\Delta h(T)}{\Delta T}\right)_{T\cong 800\text{ R}}$$

$$= \frac{h(820\text{ R}) - h(780\text{ R})}{(820 - 780)\text{R}}$$

$$= \frac{(5704.7 - 5424.2)/28.013\text{ Btu/lbm}}{(820 - 780)\text{R}} = \mathbf{0.250\ Btu/lbm \cdot R}$$

(Compare: Table A-2E*b* at 800 R = 340°F $\rightarrow$ c_p = 0.250 Btu/lbm·R)

$$c_v(800\text{ R}) = \left(\frac{du(T)}{dT}\right)_{T=800\text{ R}} \cong \left(\frac{\Delta u(T)}{\Delta T}\right)_{T\cong 800\text{ R}}$$

$$= \frac{u(820\text{ R}) - u(780\text{ R})}{(820 - 780)\text{R}}$$

$$= \frac{(4076.3 - 3875.2)/28.013\text{ Btu/lbm}}{(820 - 780)\text{R}} = \mathbf{0.179\ Btu/lbm \cdot R}$$

(Compare: Table A-2Eb at 800 R = 340°F $\rightarrow$ c_v = 0.179 Btu/lbm·R)

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12-9 The state of an ideal gas is altered slightly. The change in the specific volume of the gas is to be determined using differential relations and the ideal-gas relation at each state.

Assumptions The gas is air and air is an ideal gas.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1).

Analysis (*a*) The changes in T and P can be expressed as

$$dT \cong \Delta T = (404 - 400)\text{K} = 4 \text{ K}$$
$$dP \cong \Delta P = (96 - 100)\text{kPa} = -4 \text{ kPa}$$

The ideal gas relation $P\nu = RT$ can be expressed as $\nu = RT/P$. Note that R is a constant and $\nu = \nu(T, P)$. Applying the total differential relation and using average values for T and P,

$$d\nu = \left(\frac{\partial \nu}{\partial T}\right)_P dT + \left(\frac{\partial \nu}{\partial P}\right)_T dP = \frac{R\,dT}{P} - \frac{RT\,dP}{P^2}$$
$$= (0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})\left(\frac{4 \text{ K}}{98 \text{ kPa}} - \frac{(402 \text{ K})(-4 \text{ kPa})}{(98 \text{ kPa})^2}\right)$$
$$= (0.0117 \text{ m}^3/\text{kg}) + (0.04805 \text{ m}^3/\text{kg}) = \mathbf{0.0598 \text{ m}^3/\text{kg}}$$

(*b*) Using the ideal gas relation at each state,

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(400 \text{ K})}{100 \text{ kPa}} = 1.1480 \text{ m}^3/\text{kg}$$
$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(404 \text{ K})}{96 \text{ kPa}} = 1.2078 \text{ m}^3/\text{kg}$$

Thus,

$$\Delta\nu = \nu_2 - \nu_1 = 1.2078 - 1.1480 = \mathbf{0.0598 \text{ m}^3/\text{kg}}$$

The two results are identical.

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12-10 Using the equation of state $P(\nu - a) = RT$, the cyclic relation, and the reciprocity relation at constant ν are to be verified.

Analysis (*a*) This equation of state involves three variables P, ν, and T. Any two of these can be taken as the independent variables, with the remaining one being the dependent variable. Replacing x, y, and z by P, ν, and T, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial \nu}\right)_T \left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial T}{\partial P}\right)_\nu = -1$$

where

$$P = \frac{RT}{\nu - a} \longrightarrow \left(\frac{\partial P}{\partial \nu}\right)_T = \frac{-RT}{(\nu - a)^2} = -\frac{P}{\nu - a}$$

$$\nu = \frac{RT}{P} + a \longrightarrow \left(\frac{\partial \nu}{\partial T}\right)_P = \frac{R}{P}$$

$$T = \frac{P(\nu - a)}{R} \longrightarrow \left(\frac{\partial T}{\partial P}\right)_\nu = \frac{\nu - a}{R}$$

Substituting,

$$\left(\frac{\partial P}{\partial \nu}\right)_T \left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial T}{\partial P}\right)_\nu = \left(-\frac{P}{\nu - a}\right)\left(\frac{R}{P}\right)\left(\frac{\nu - a}{R}\right) = -1$$

which is the desired result.

(*b*) The reciprocity rule for this gas at ν = constant can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_\nu = \frac{1}{(\partial T / \partial P)_\nu}$$

$$T = \frac{P(\nu - a)}{R} \longrightarrow \left(\frac{\partial T}{\partial P}\right)_\nu = \frac{\nu - a}{R}$$

$$P = \frac{RT}{\nu - a} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_\nu = \frac{R}{\nu - a}$$

We observe that the first differential is the inverse of the second one. Thus the proof is complete.

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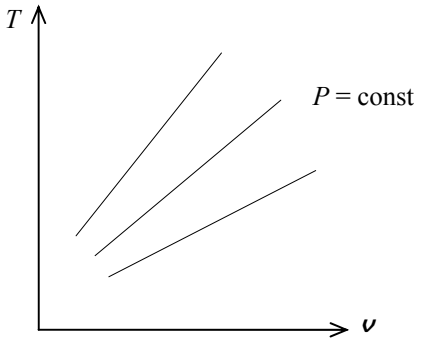
12-11 It is to be proven for an ideal gas that the P = constant lines on a T-v diagram are straight lines and that the high pressure lines are steeper than the low-pressure lines.

Analysis (*a*) For an ideal gas $Pv = RT$ or $T = Pv/R$. Taking the partial derivative of T with respect to v holding P constant yields

$$\left(\frac{\partial T}{\partial v}\right)_P = \frac{P}{R}$$

which remains constant at P = constant. Thus the derivative $(\partial T/\partial v)_P$, which represents the slope of the P = const. lines on a T-v diagram, remains constant. That is, the P = const. lines are straight lines on a T-v diagram.

(*b*) The slope of the P = const. lines on a T-v diagram is equal to P/R, which is proportional to P. Therefore, the high pressure lines are steeper than low pressure lines on the T-v diagram.



12-12 A relation is to be derived for the slope of the v = constant lines on a T-P diagram for a gas that obeys the van der Waals equation of state.

Analysis The van der Waals equation of state can be expressed as

$$T = \frac{1}{R}\left(P + \frac{a}{v^2}\right)(v - b)$$

Taking the derivative of T with respect to P holding v constant,

$$\left(\frac{\partial T}{\partial P}\right)_v = \frac{1}{R}(1 + 0)(v - b) = \frac{v - b}{R}$$

which is the slope of the v = constant lines on a T-P diagram.

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The Maxwell Relations

12-13 The validity of the last Maxwell relation for refrigerant-134a at a specified state is to be verified.

Analysis We do not have exact analytical property relations for refrigerant-134a, and thus we need to replace the differential quantities in the last Maxwell relation with the corresponding finite quantities. Using property values from the tables about the specified state,

$$\left(\frac{\partial s}{\partial P}\right)_T \stackrel{?}{=} -\left(\frac{\partial v}{\partial T}\right)_P$$

$$\left(\frac{\Delta s}{\Delta P}\right)_{T=50°\text{C}} \stackrel{?}{\cong} -\left(\frac{\Delta v}{\Delta T}\right)_{P=700\text{kPa}}$$

$$\left(\frac{s_{900\text{ kPa}} - s_{500\text{ kPa}}}{(900-500)\text{kPa}}\right)_{T=50°\text{C}} \stackrel{?}{\cong} -\left(\frac{v_{70°\text{C}} - v_{30°\text{C}}}{(70-30)°\text{C}}\right)_{P=700\text{kPa}}$$

$$\frac{(0.9660-1.0309)\text{kJ/kg}\cdot\text{K}}{(900-500)\text{kPa}} \stackrel{?}{\cong} -\frac{(0.036373-0.029966)\text{m}^3/\text{kg}}{(70-30)°\text{C}}$$

$$-1.621\times10^{-4}\ \text{m}^3/\text{kg}\cdot\text{K} \cong -1.602\times10^{-4}\ \text{m}^3/\text{kg}\cdot\text{K}$$

since kJ ≡ kPa·m³, and K ≡ °C for temperature differences. Thus the last Maxwell relation is satisfied.

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12-14 Problem 12-13 is reconsidered. The validity of the last Maxwell relation for refrigerant 134a at the specified state is to be verified.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data:"
T=50 [C]
P=700 [kPa]
P_increment = 200 [kPa]
T_increment = 20 [C]
P[2]=P+P_increment
P[1]=P-P_increment
T[2]=T+T_increment
T[1]=T-T_increment

DELTAP = P[2]-P[1]
DELTAT = T[2]-T[1]

v[1]=volume(R134a,T=T[1],P=P)
v[2]=volume(R134a,T=T[2],P=P)
s[1]=entropy(R134a,T=T,P=P[1])
s[2]=entropy(R134a,T=T,P=P[2])

DELTAs=s[2] - s[1]
DELTAv=v[2] - v[1]

"The partial derivatives in the last Maxwell relation (Eq. 12-19) is associated with the Gibbs function and are approximated by the ratio of ordinary differentials:"

LeftSide =DELTAs/DELTAP*Convert(kJ,m^3-kPa) "[m^3/kg-K]"   "at T = Const."
RightSide=-DELTAv/DELTAT  "[m^3/kg-K]"     "at P = Const."
```

SOLUTION

```
DELTAP=400 [kPa]
DELTAs=-0.06484 [kJ/kg-K]
DELTAT=40 [C]
DELTAv=0.006407 [m^3/kg]
LeftSide=-0.0001621 [m^3/kg-K]
P=700 [kPa]
P[1]=500 [kPa]
P[2]=900 [kPa]
P_increment=200 [kPa]
RightSide=-0.0001602 [m^3/kg-K]
s[1]=1.0309 [kJ/kg-K]
s[2]=0.9660 [kJ/kg-K]
T=50 [C]
T[1]=30 [C]
T[2]=70 [C]
T_increment=20 [C]
v[1]=0.02997 [m^3/kg]
v[2]=0.03637 [m^3/kg]
```

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12-15E The validity of the last Maxwell relation for steam at a specified state is to be verified.

Analysis We do not have exact analytical property relations for steam, and thus we need to replace the differential quantities in the last Maxwell relation with the corresponding finite quantities. Using property values from the tables about the specified state,

$$\left(\frac{\partial s}{\partial P}\right)_T \stackrel{?}{=} -\left(\frac{\partial v}{\partial T}\right)_P$$

$$\left(\frac{\Delta s}{\Delta P}\right)_{T=800°\text{F}} \stackrel{?}{\cong} -\left(\frac{\Delta v}{\Delta T}\right)_{P=400\text{psia}}$$

$$\left(\frac{s_{450\text{ psia}} - s_{350\text{ psia}}}{(450-350)\text{psia}}\right)_{T=800°\text{F}} \stackrel{?}{\cong} -\left(\frac{v_{900°\text{F}} - v_{700°\text{F}}}{(900-700)°\text{F}}\right)_{P=400\text{psia}}$$

$$\frac{(1.6706-1.7009)\text{Btu/lbm}\cdot\text{R}}{(450-350)\text{psia}} \stackrel{?}{\cong} -\frac{(1.9777-1.6507)\text{ft}^3/\text{lbm}}{(900-700)°\text{F}}$$

$$-1.639\times10^{-3}\ \text{ft}^3/\text{lbm}\cdot\text{R} \cong -1.635\times10^{-3}\ \text{ft}^3/\text{lbm}\cdot\text{R}$$

since 1 Btu ≡ 5.4039 psia·ft^3, and R ≡ °F for temperature differences. Thus the fourth Maxwell relation is satisfied.

12-16 Using the Maxwell relations, a relation for $(\partial s/\partial P)_T$ for a gas whose equation of state is $P(v-b) = RT$ is to be obtained.

Analysis This equation of state can be expressed as $v = \frac{RT}{P} + b$. Then,

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

From the fourth Maxwell relation,

$$\left(\frac{\partial s}{\partial P}\right)_T = -\left(\frac{\partial v}{\partial T}\right)_P = -\frac{\boldsymbol{R}}{\boldsymbol{P}}$$

12-17 Using the Maxwell relations, a relation for $(\partial s/\partial v)_T$ for a gas whose equation of state is $(P-a/v^2)(v-b) = RT$ is to be obtained.

Analysis This equation of state can be expressed as $P = \frac{RT}{v-b} + \frac{a}{v^2}$. Then,

$$\left(\frac{\partial P}{\partial T}\right)_v = \frac{R}{v-b}$$

From the third Maxwell relation,

$$\left(\frac{\partial s}{\partial v}\right)_T = \left(\frac{\partial P}{\partial T}\right)_v = \frac{\boldsymbol{R}}{\boldsymbol{v-b}}$$

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12-18 Using the Maxwell relations and the ideal-gas equation of state, a relation for $(\partial s/\partial \boldsymbol{v})_T$ for an ideal gas is to be obtained.

Analysis The ideal gas equation of state can be expressed as $P = \dfrac{RT}{\boldsymbol{v}}$. Then,

$$\left(\frac{\partial P}{\partial T}\right)_{\boldsymbol{v}} = \frac{R}{\boldsymbol{v}}$$

From the third Maxwell relation,

$$\left(\frac{\partial s}{\partial \boldsymbol{v}}\right)_T = \left(\frac{\partial P}{\partial T}\right)_{\boldsymbol{v}} = \frac{\boldsymbol{R}}{\boldsymbol{v}}$$

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12-19 It is to be proven that $\left(\frac{\partial P}{\partial T}\right)_s = \frac{k}{k-1}\left(\frac{\partial P}{\partial T}\right)_v$

Analysis Using the definition of c_v,

$$c_v = T\left(\frac{\partial s}{\partial T}\right)_v = T\left(\frac{\partial s}{\partial P}\right)_v\left(\frac{\partial P}{\partial T}\right)_v$$

Substituting the first Maxwell relation $\left(\frac{\partial s}{\partial P}\right)_v = -\left(\frac{\partial v}{\partial T}\right)_s$,

$$c_v = -T\left(\frac{\partial v}{\partial T}\right)_s\left(\frac{\partial P}{\partial T}\right)_v$$

Using the definition of c_p,

$$c_p = T\left(\frac{\partial s}{\partial T}\right)_P = T\left(\frac{\partial s}{\partial v}\right)_P\left(\frac{\partial v}{\partial T}\right)_P$$

Substituting the second Maxwell relation $\left(\frac{\partial s}{\partial v}\right)_P = \left(\frac{\partial P}{\partial T}\right)_s$,

$$c_p = T\left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial v}{\partial T}\right)_P$$

From Eq. 12-46,

$$c_p - c_v = -T\left(\frac{\partial v}{\partial T}\right)_P^2\left(\frac{\partial P}{\partial v}\right)_T$$

Also,

$$\frac{k}{k-1} = \frac{c_p}{c_p - c_v}$$

Then,

$$\frac{k}{k-1} = -\frac{\left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial v}{\partial T}\right)_P}{\left(\frac{\partial v}{\partial T}\right)_P^2\left(\frac{\partial P}{\partial v}\right)_T} = -\left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_T$$

Substituting this into the original equation in the problem statement produces

$$\left(\frac{\partial P}{\partial T}\right)_s = -\left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_T\left(\frac{\partial P}{\partial T}\right)_v$$

But, according to the cyclic relation, the last three terms are equal to −1. Then,

$$\left(\frac{\partial P}{\partial T}\right)_s = \left(\frac{\partial P}{\partial T}\right)_s$$

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12-20 It is to be shown how T, ν, u, a, and g could be evaluated from the thermodynamic function $h = h(s, P)$.

Analysis Forming the differential of the given expression for h produces

$$dh = \left(\frac{\partial h}{\partial s}\right)_P ds + \left(\frac{\partial h}{\partial P}\right)_s dP$$

Solving the dh Gibbs equation gives

$$dh = Tds + \nu dP$$

Comparing the coefficient of these two expressions

$$T = \left(\frac{\partial h}{\partial s}\right)_P$$

$$\nu = \left(\frac{\partial h}{\partial P}\right)_s$$

both of which can be evaluated for a given P and s.

From the definition of the enthalpy,

$$u = h - P\nu = h\text{-}P\left(\frac{\partial h}{\partial P}\right)_s$$

Similarly, the definition of the Helmholtz function,

$$a = u - Ts = h - P\left(\frac{\partial h}{\partial P}\right)_s - s\left(\frac{\partial h}{\partial s}\right)_P$$

while the definition of the Gibbs function gives

$$q = h - Ts = h - s\left(\frac{\partial h}{\partial s}\right)_P$$

All of these can be evaluated for a given P and s and the fundamental $h(s,P)$ equation.

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The Clapeyron Equation

12-21C It enables us to determine the enthalpy of vaporization from h_{fg} at a given temperature from the P, v, T data alone.

12-22C It is assumed that $v_{fg} \cong v_g \cong RT/P$, and $h_{fg} \cong$ constant for small temperature intervals.

12-23 Using the Clapeyron equation, the enthalpy of vaporization of steam at a specified pressure is to be estimated and to be compared to the tabulated data.

Analysis From the Clapeyron equation,

$$\begin{aligned} h_{fg} &= Tv_{fg}\left(\frac{dP}{dT}\right)_{\text{sat}} \\ &\cong T(v_g - v_f)_{@300\text{ kPa}}\left(\frac{\Delta P}{\Delta T}\right)_{\text{sat, 300 kPa}} \\ &= T_{\text{sat@300 kPa}}(v_g - v_f)_{@300\text{ kPa}}\left(\frac{(325-275)\text{kPa}}{T_{\text{sat@325 kPa}} - T_{\text{sat@275 kPa}}}\right) \\ &= (133.52 + 273.15\text{ K})(0.60582 - 0.001073\text{ m}^3/\text{kg})\left(\frac{50\text{ kPa}}{(136.27-130.58)^\circ\text{C}}\right) \\ &= \mathbf{2159.9\ kJ/kg} \end{aligned}$$

The tabulated value of h_{fg} at 300 kPa is **2163.5 kJ/kg**.

12-24 The h_{fg} and s_{fg} of steam at a specified temperature are to be calculated using the Clapeyron equation and to be compared to the tabulated data.

Analysis From the Clapeyron equation,

$$\begin{aligned} h_{fg} &= Tv_{fg}\left(\frac{dP}{dT}\right)_{\text{sat}} \\ &\cong T(v_g - v_f)_{@120^\circ\text{C}}\left(\frac{\Delta P}{\Delta T}\right)_{\text{sat,120°C}} \\ &= T(v_g - v_f)_{@120^\circ\text{C}}\left(\frac{P_{\text{sat@125°C}} - P_{\text{sat@115°C}}}{125^\circ\text{C} - 115^\circ\text{C}}\right) \\ &= (120 + 273.15\text{ K})(0.89133 - 0.001060\text{ m}^3/\text{kg})\left(\frac{(232.23-169.18)\text{kPa}}{10\text{ K}}\right) \\ &= \mathbf{2206.8\ kJ/kg} \end{aligned}$$

Also, $$s_{fg} = \frac{h_{fg}}{T} = \frac{2206.8\text{ kJ/kg}}{(120 + 273.15)\text{K}} = \mathbf{5.6131\ kJ/kg \cdot K}$$

The tabulated values at 120°C are h_{fg} = **2202.1 kJ/kg** and s_{fg} = **5.6013 kJ/kg·K**.

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12-25E The h_{fg} of refrigerant-134a at a specified temperature is to be calculated using the Clapeyron equation and Clapeyron-Clausius equation and to be compared to the tabulated data.

Analysis (*a*) From the Clapeyron equation,

$$\begin{aligned} h_{fg} &= T\nu_{fg}\left(\frac{dP}{dT}\right)_{\text{sat}} \\ &\cong T(\nu_g - \nu_f)_{@\,10°\text{F}}\left(\frac{\Delta P}{\Delta T}\right)_{\text{sat, }10°\text{F}} \\ &= T(\nu_g - \nu_f)_{@\,10°\text{F}}\left(\frac{P_{\text{sat@ }15°\text{F}} - P_{\text{sat@}5°\text{F}}}{15°\text{F} - 5°\text{F}}\right) \\ &= (10 + 459.67\text{ R})(1.7345 - 0.01201\,\text{ft}^3/\text{lbm})\left(\frac{(29.759 - 23.793)\text{ psia}}{10\text{ R}}\right) \\ &= 482.6\,\text{psia}\cdot\text{ft}^3/\text{lbm} = \mathbf{89.31\,Btu/lbm} \quad (0.1\%\text{ error}) \end{aligned}$$

since 1 Btu = 5.4039 psia·ft^3.

(*b*) From the Clapeyron-Clausius equation,

$$\ln\left(\frac{P_2}{P_1}\right)_{\text{sat}} \cong \frac{h_{fg}}{R}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)_{\text{sat}}$$

$$\ln\left(\frac{23.793\,\text{psia}}{29.759\,\text{psia}}\right) \cong \frac{h_{fg}}{0.01946\,\text{Btu/lbm}\cdot\text{R}}\left(\frac{1}{15 + 459.67\text{ R}} - \frac{1}{5 + 459.67\text{ R}}\right)$$

$$h_{fg} = \mathbf{96.04\,Btu/lbm} \quad (7.6\%\text{ error})$$

The tabulated value of h_{fg} at 10°F is **89.23 Btu/lbm**.

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12-26 The enthalpy of vaporization of steam as a function of temperature using Clapeyron equation and steam data in EES is to be plotted.

Analysis The enthalpy of vaporization is determined using Clapeyron equation from

$$h_{fg,\text{Clapeyron}} = T\nu_{fg}\frac{\Delta P}{\Delta T}$$

At 100ºC, for an increment of 5ºC, we obtain

$$\begin{aligned}
T_1 &= T - T_{\text{increment}} = 100 - 5 = 95°\text{C} \\
T_2 &= T + T_{\text{increment}} = 100 + 5 = 105°\text{C} \\
P_1 &= P_{\text{sat@ }95°\text{C}} = 84.61\,\text{kPa} \\
P_2 &= P_{\text{sat@ }105°\text{C}} = 120.90\,\text{kPa} \\
\Delta T &= T_2 - T_1 = 105 - 95 = 10°\text{C} \\
\Delta P &= P_2 - P_1 = 120.90 - 84.61 = 36.29\,\text{kPa} \\
\nu_{f@\,100°\text{C}} &= 0.001043\,\text{m}^3/\text{kg} \\
\nu_{g@\,100°\text{C}} &= 1.6720\,\text{m}^3/\text{kg} \\
\nu_{fg} &= \nu_g - \nu_f = 1.6720 - 0.001043 = 1.6710\,\text{m}^3/\text{kg}
\end{aligned}$$

Substituting,

$$h_{fg,\text{Clapeyron}} = T\nu_{fg}\frac{\Delta P}{\Delta T} = (100 + 273.15\,\text{K})(1.6710\,\text{m}^3/\text{kg})\frac{36.29\,\text{kPa}}{10\,\text{K}} = \mathbf{2262.8\ kJ/kg}$$

The enthalpy of vaporization from steam table is

$$h_{fg@\,100°\text{C}} = \mathbf{2256.4\ m^3/kg}$$

The percent error in using Clapeyron equation is

$$\text{PercentError} = \frac{2262.8 - 2256.4}{2256.4} \times 100 = \mathbf{0.28\%}$$

We repeat the analysis over the temperature range 10 to 200ºC using EES. Below, the copy of EES solution is provided:

```
"Input Data:"
"T=100" "[C]"
T_increment = 5"[C]"
T[2]=T+T_increment"[C]"
T[1]=T-T_increment"[C]"
P[1] = pressure(Steam_iapws,T=T[1],x=0)"[kPa]"
P[2] = pressure(Steam_iapws,T=T[2],x=0)"[kPa]"
DELTAP = P[2]-P[1]"[kPa]"
DELTAT = T[2]-T[1]"[C]"

v_f=volume(Steam_iapws,T=T,x=0)"[m^3/kg]"
v_g=volume(Steam_iapws,T=T,x=1)"[m^3/kg]"
h_f=enthalpy(Steam_iapws,T=T,x=0)"[kJ/kg]"
h_g=enthalpy(Steam_iapws,T=T,x=1)"[kJ/kg]"

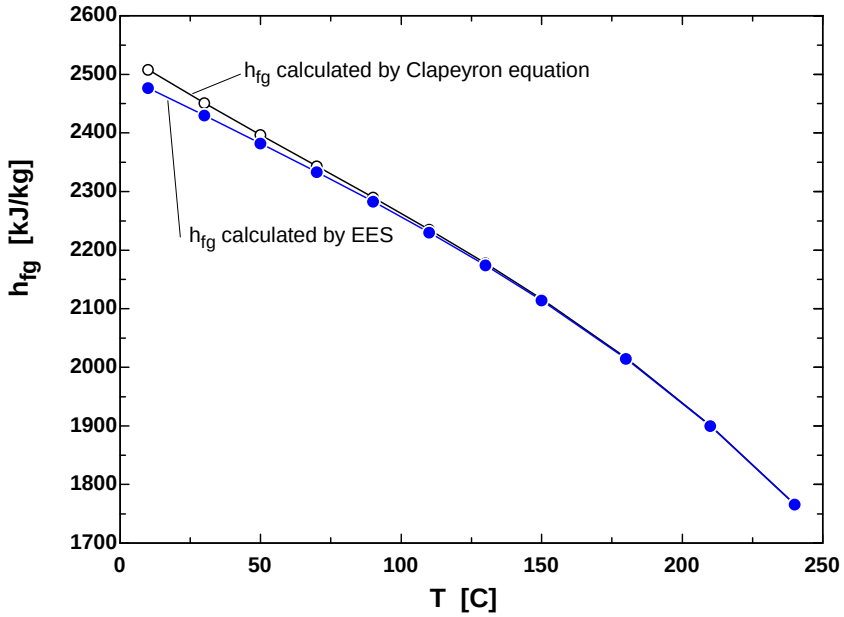
h_fg=h_g - h_f"[kJ/kg-K]"
v_fg=v_g - v_f"[m^3/kg]"

"The Clapeyron equation (Eq. 11-22) provides a means to calculate the enthalpy of vaporization, h_fg at a given temperature by determining the slope of the saturation curve on a P-T diagram and the specific volume of the saturated liquid and satruated vapor at the temperature."
```

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h_fg_Clapeyron=(T+273.15)*v_fg*DELTAP/DELTAT*Convert(m^3-kPa,kJ)"[kJ/kg]"
PercentError=ABS(h_fg_Clapeyron-h_fg)/h_fg*100"[%]"

h_{fg} [kJ/kg]	$h_{fg,Clapeyron}$ [kJ/kg]	PercentError [%]	T [C]
2477.20	2508.09	1.247	10
2429.82	2451.09	0.8756	30
2381.95	2396.69	0.6188	50
2333.04	2343.47	0.4469	70
2282.51	2290.07	0.3311	90
2229.68	2235.25	0.25	110
2173.73	2177.86	0.1903	130
2113.77	2116.84	0.1454	150
2014.17	2016.15	0.09829	180
1899.67	1900.98	0.06915	210
1765.50	1766.38	0.05015	240

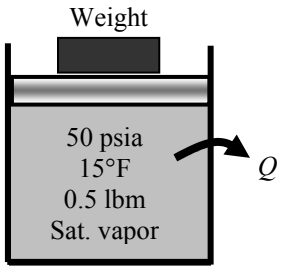


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12-27E A substance is cooled in a piston-cylinder device until it turns from saturated vapor to saturated liquid at a constant pressure and temperature. The boiling temperature of this substance at a different pressure is to be estimated.

Analysis From the Clapeyron equation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} = \frac{h_{fg}}{T v_{fg}} = \frac{(250\ \text{Btu})\left(\frac{5.404\ \text{psia}\cdot\text{ft}^3}{1\ \text{Btu}}\right)/(0.5\ \text{lbm})}{(475\ \text{R})(1.5\ \text{ft}^3)/(0.5\ \text{lbm})} = 1.896\ \text{psia/R}$$



Using the finite difference approximation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} \approx \left(\frac{P_2 - P_1}{T_2 - T_1}\right)_{\text{sat}}$$

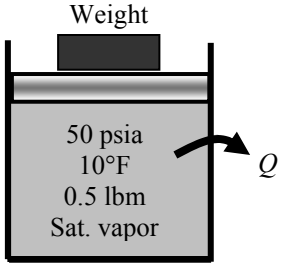
Solving for T_2,

$$T_2 = T_1 + \frac{P_2 - P_1}{dP/dT} = 475\ \text{R} + \frac{(60-50)\text{psia}}{1.896\ \text{psia/R}} = \mathbf{480.3\ R}$$

12-28E A substance is cooled in a piston-cylinder device until it turns from saturated vapor to saturated liquid at a constant pressure and temperature. The saturation pressure of this substance at a different temperature is to be estimated.

Analysis From the Clapeyron equation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} = \frac{h_{fg}}{T v_{fg}} = \frac{(250\ \text{Btu})\left(\frac{5.404\ \text{psia}\cdot\text{ft}^3}{1\ \text{Btu}}\right)/(0.5\ \text{lbm})}{(475\ \text{R})(1.5\ \text{ft}^3)/(0.5\ \text{lbm})} = 1.896\ \text{psia/R}$$



Using the finite difference approximation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} \approx \left(\frac{P_2 - P_1}{T_2 - T_1}\right)_{\text{sat}}$$

Solving for P_2,

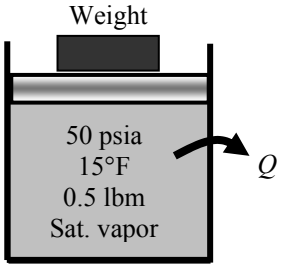
$$P_2 = P_1 + \frac{dP}{dT}(T_2 - T_1) = 50\ \text{psia} + (1.896\ \text{psia/R})(470 - 475)\text{R} = \mathbf{40.52\ psia}$$

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12-29E A substance is cooled in a piston-cylinder device until it turns from saturated vapor to saturated liquid at a constant pressure and temperature. The s_{fg} of this substance at the given temperature is to be estimated.

Analysis From the Clapeyron equation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} = \frac{h_{fg}}{T\nu_{fg}} = \frac{s_{fg}}{\nu_{fg}}$$



Solving for s_{fg},

$$s_{fg} = \frac{h_{fg}}{T} = \frac{(250\ \text{Btu})/(0.5\ \text{lbm})}{475\ \text{R}} = \mathbf{1.053\ Btu/lbm \cdot R}$$

Alternatively,

$$s_{fg} = \left(\frac{dP}{dT}\right)_{\text{sat}} \nu_{fg} = (1.896\ \text{psia/R})\frac{1.5\ \text{ft}^3}{0.5\ \text{lbm}}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia} \cdot \text{ft}^3}\right) = 1.053\ \text{Btu/lbm} \cdot \text{R}$$

12-30E Saturation properties for R-134a at a specified temperature are given. The saturation pressure is to be estimated at two different temperatures.

Analysis From the Clapeyron equation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} = \frac{h_{fg}}{T\nu_{fg}} = \frac{90.886\ \text{Btu/lbm}}{(460\ \text{R})(2.1446\ \text{ft}^3/\text{lbm})}\left(\frac{5.404\ \text{psia} \cdot \text{ft}^3}{1\ \text{Btu}}\right) = 0.4979\ \text{psia/R}$$

Using the finite difference approximation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} \approx \left(\frac{P_2 - P_1}{T_2 - T_1}\right)_{\text{sat}}$$

Solving for P_2 at −15°F

$$P_2 = P_1 + \frac{dP}{dT}(T_2 - T_1) = 21.185\ \text{psia} + (0.4979\ \text{psia/R})(445 - 460)\text{R} = \mathbf{13.72\ psia}$$

Solving for P_2 at −30°F

$$P_2 = P_1 + \frac{dP}{dT}(T_2 - T_1) = 21.185\ \text{psia} + (0.4979\ \text{psia/R})(430 - 460)\text{R} = \mathbf{6.25\ psia}$$

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12-31E A table of properties for methyl chloride is given. The saturation pressure is to be estimated at two different temperatures.

Analysis The Clapeyron equation is

$$\left(\frac{dP}{dT}\right)_{\text{sat}} = \frac{h_{fg}}{T\nu_{fg}}$$

Using the finite difference approximation,

$$\left(\frac{dP}{dT}\right)_{\text{sat}} \approx \left(\frac{P_2 - P_1}{T_2 - T_1}\right)_{\text{sat}} = \frac{h_{fg}}{T\nu_{fg}}$$

Solving this for the second pressure gives for $T_2 = 110°\text{F}$

$$\begin{aligned} P_2 &= P_1 + \frac{h_{fg}}{T\nu_{fg}}(T_2 - T_1) \\ &= 116.7 \text{ psia} + \frac{154.85 \text{ Btu/lbm}}{(560 \text{ R})(0.86332 \text{ ft}^3/\text{lbm})}\left(\frac{5.404 \text{ psia} \cdot \text{ft}^3}{1 \text{ Btu}}\right)(110 - 100)\text{R} \\ &= \mathbf{134.0 \text{ psia}} \end{aligned}$$

When $T_2 = 90°\text{F}$

$$\begin{aligned} P_2 &= P_1 + \frac{h_{fg}}{T\nu_{fg}}(T_2 - T_1) \\ &= 116.7 \text{ psia} + \frac{154.85 \text{ Btu/lbm}}{(560 \text{ R})(0.86332 \text{ ft}^3/\text{lbm})}\left(\frac{5.404 \text{ psia} \cdot \text{ft}^3}{1 \text{ Btu}}\right)(90 - 100)\text{R} \\ &= \mathbf{99.4 \text{ psia}} \end{aligned}$$

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12-32 It is to be shown that $c_{p,g} - c_{p,f} = T\left(\frac{\partial(h_{fg}/T)}{\partial T}\right)_P + v_{fg}\left(\frac{\partial P}{\partial T}\right)_{\text{sat}}$.

Analysis The definition of specific heat and Clapeyron equation are

$$c_p = \left(\frac{\partial h}{\partial T}\right)_P$$

$$\left(\frac{dP}{dT}\right)_{\text{sat}} = \frac{h_{fg}}{T v_{fg}}$$

According to the definition of the enthalpy of vaporization,

$$\frac{h_{fg}}{T} = \frac{h_g}{T} - \frac{h_f}{T}$$

Differentiating this expression gives

$$\begin{aligned}\left(\frac{\partial h_{fg}/T}{\partial T}\right)_P &= \left(\frac{\partial h_g/T}{\partial T}\right)_P - \left(\frac{\partial h_f/T}{\partial T}\right)_P \\ &= \frac{1}{T}\left(\frac{\partial h_g}{\partial T}\right)_P - \frac{h_g}{T^2} - \frac{1}{T}\left(\frac{\partial h_f}{\partial T}\right)_P + \frac{h_f}{T^2} \\ &= \frac{c_{p,g}}{T} - \frac{c_{p,f}}{T} - \frac{h_g - h_f}{T^2}\end{aligned}$$

Using Clasius-Clapeyron to replace the last term of this expression and solving for the specific heat difference gives

$$c_{p,g} - c_{p,f} = T\left(\frac{\partial(h_{fg}/T)}{\partial T}\right)_P + v_{fg}\left(\frac{\partial P}{\partial T}\right)_{\text{sat}}$$

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General Relations for *du*, *dh*, *ds*, c_v, and c_p

12-33C Yes, through the relation

$$\left(\frac{\partial c_p}{\partial P}\right)_T = -T\left(\frac{\partial^2 v}{\partial T^2}\right)_P$$

12-34E The specific heat difference $c_p - c_v$ for liquid water at 1000 psia and 300°F is to be estimated.

Analysis The specific heat difference $c_p - c_v$ is given as

$$c_p - c_v = -T\left(\frac{\partial v}{\partial T}\right)_P^2 \left(\frac{\partial P}{\partial v}\right)_T$$

Approximating differentials by differences about the specified state,

$$\begin{aligned} c_p - c_v &\cong -T\left(\frac{\Delta v}{\Delta T}\right)_{P=1000\text{psia}}^2 \left(\frac{\Delta P}{\Delta v}\right)_{T=300°\text{F}} \\ &= -(300 + 459.67\ \text{R})\left(\frac{v_{325°\text{F}} - v_{275°\text{F}}}{(325-275)°\text{F}}\right)_{P=1000\text{psia}}^2 \left(\frac{(1500-500)\text{psia}}{v_{1500\text{psia}} - v_{500\text{psia}}}\right)_{T=300°\text{F}} \\ &= -(609.67\ \text{R})\left(\frac{(0.017633-0.017151)\text{ft}^3/\text{lbm}}{50\ \text{R}}\right)^2 \left(\frac{1000\ \text{psia}}{(0.017345-0.017417)\text{ft}^3/\text{lbm}}\right) \\ &= 0.986\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R} \\ &= \mathbf{0.183\ Btu/lbm\cdot R}\ \ (1\ \text{Btu} = 5.4039\ \text{psia}\cdot\text{ft}^3) \end{aligned}$$

Properties are obtained from Table A-7E.

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12-35 The volume expansivity β and the isothermal compressibility α of refrigerant-134a at 200 kPa and 30°C are to be estimated.

Analysis The volume expansivity and isothermal compressibility are expressed as

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_P \text{ and } \alpha = -\frac{1}{v}\left(\frac{\partial v}{\partial P}\right)_T$$

Approximating differentials by differences about the specified state,

$$\beta \cong \frac{1}{v}\left(\frac{\Delta v}{\Delta T}\right)_{P=200\text{kPa}} = \frac{1}{v}\left(\frac{v_{40°\text{C}} - v_{20°\text{C}}}{(40-20)°\text{C}}\right)_{P=200\text{kPa}}$$

$$= \frac{1}{0.11874\ \text{m}^3/\text{kg}}\left(\frac{(0.12322-0.11418)\text{m}^3/\text{kg}}{20\ \text{K}}\right)$$

$$= \mathbf{0.00381\ K^{-1}}$$

and

$$\alpha \cong -\frac{1}{v}\left(\frac{\Delta v}{\Delta P}\right)_{T=30°\text{C}} = -\frac{1}{v}\left(\frac{v_{240\text{kPa}} - v_{180\text{kPa}}}{(240-180)\text{kPa}}\right)_{T=30°\text{C}}$$

$$= -\frac{1}{0.11874\ \text{m}^3/\text{kg}}\left(\frac{(0.09812-0.13248)\text{m}^3/\text{kg}}{60\ \text{kPa}}\right)$$

$$= \mathbf{0.00482\ kPa^{-1}}$$

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12-36 The internal energy change of air between two specified states is to be compared for two equations of states.

Assumptions Constant specific heats for air can be used.

Properties For air at the average temperature (20+300)/2=160°C=433 K, c_v = 0.731 kJ/kg·K (Table A-2*b*).

Analysis Solving the equation of state for *P* gives

$$P = \frac{RT}{v - a}$$

Then,

$$\left(\frac{\partial P}{\partial T}\right)_v = \frac{R}{v - a}$$

Using equation 12-29,

$$du = c_v dT + \left[T\left(\frac{\partial P}{\partial T}\right)_v - P\right]dv$$

Substituting,

$$du = c_v dT + \left(\frac{RT}{v - a} - \frac{RT}{v - a}\right)dv$$
$$= c_v dT$$

Integrating this result between the two states with constant specific heats gives

$$u_2 - u_1 = c_v(T_2 - T_1) = (0.731\,\text{kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{205\ kJ/kg}$$

The ideal gas model for the air gives

$$du = c_v dT$$

which gives the same answer.

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12-37 The enthalpy change of air between two specified states is to be compared for two equations of states.

Assumptions Constant specific heats for air can be used.

Properties For air at the average temperature (20+300)/2=160°C=433 K, c_p = 1.018 kJ/kg·K (Table A-2*b*).

Analysis Solving the equation of state for $\boldsymbol{v}$ gives

$$\boldsymbol{v} = \frac{RT}{P} + a$$

Then,

$$\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P = \frac{R}{P}$$

Using equation 12-35,

$$dh = c_p dT + \left[\boldsymbol{v} - T\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P\right] dP$$

Substituting,

$$dh = c_p dT + \left(\frac{RT}{P} + a - \frac{RT}{P}\right) dP$$
$$= c_p dT + a dP$$

Integrating this result between the two states with constant specific heats gives

$$h_2 - h_1 = c_p(T_2 - T_1) + a(P_2 - P_1)$$
$$= (1.018\,\text{kJ/kg}\cdot\text{K})(300 - 20)\text{K} + (0.01\,\text{m}^3/\text{kg})(600 - 100)\text{kPa}$$
$$= \mathbf{290.0\ kJ/kg}$$

For an ideal gas,

$$dh = c_p dT$$

which when integrated gives

$$h_2 - h_1 = c_p(T_2 - T_1) = (1.018\,\text{kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{285.0\ kJ/kg}$$

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12-38 The entropy change of air between two specified states is to be compared for two equations of states.

Assumptions Constant specific heats for air can be used.

Properties For air at the average temperature (20+300)/2=160°C=433 K, c_p = 1.018 kJ/kg·K (Table A-2*b*) and R = 0.287 kJ/kg·K (Table A-1).

Analysis Solving the equation of state for v gives

$$v = \frac{RT}{P} + a$$

Then,

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

The entropy differential is

$$ds = c_p \frac{dT}{T} - \left(\frac{\partial v}{\partial T}\right)_P dP$$
$$= c_p \frac{dT}{T} - R\frac{dP}{P}$$

which is the same as that of an ideal gas. Integrating this result between the two states with constant specific heats gives

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$
$$= (1.018\ \text{kJ/kg}\cdot\text{K})\ln\frac{573\ \text{K}}{293\ \text{K}} - (0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{600\ \text{kPa}}{100\ \text{kPa}}$$
$$= \mathbf{0.1686\ kJ/kg\cdot K}$$

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12-39 The internal energy change of helium between two specified states is to be compared for two equations of states.

Properties For helium, $c_v = 3.1156$ kJ/kg·K (Table A-2a).

Analysis Solving the equation of state for P gives

$$P = \frac{RT}{v - a}$$

Then,

$$\left(\frac{\partial P}{\partial T}\right)_v = \frac{R}{v - a}$$

Using equation 12-29,

$$du = c_v dT + \left[T\left(\frac{\partial P}{\partial T}\right)_v - P\right] dv$$

Substituting,

$$du = c_v dT + \left(\frac{RT}{v - a} - \frac{RT}{v - a}\right) dv$$
$$= c_v dT$$

Integrating this result between the two states gives

$$u_2 - u_1 = c_v (T_2 - T_1) = (3.1156 \text{ kJ/kg} \cdot \text{K})(300 - 20)\text{K} = \mathbf{872.4\ kJ/kg}$$

The ideal gas model for the helium gives

$$du = c_v dT$$

which gives the same answer.

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12-40 The enthalpy change of helium between two specified states is to be compared for two equations of states.

Properties For helium, c_p = 5.1926 kJ/kg·K (Table A-2*a*).

Analysis Solving the equation of state for ν gives

$$\nu = \frac{RT}{P} + a$$

Then,

$$\left(\frac{\partial \nu}{\partial T}\right)_P = \frac{R}{P}$$

Using equation 12-35,

$$dh = c_p dT + \left[\nu - T\left(\frac{\partial \nu}{\partial T}\right)_P\right] dP$$

Substituting,

$$\begin{aligned} dh &= c_p dT + \left(\frac{RT}{P} + a - \frac{RT}{P}\right) dP \\ &= c_p dT + a dP \end{aligned}$$

Integrating this result between the two states gives

$$\begin{aligned} h_2 - h_1 &= c_p (T_2 - T_1) + a(P_2 - P_1) \\ &= (5.1926\ \text{kJ/kg} \cdot \text{K})(300 - 20)\text{K} + (0.01\ \text{m}^3/\text{kg})(600 - 100)\text{kPa} \\ &= \mathbf{1459\ kJ/kg} \end{aligned}$$

For an ideal gas,

$$dh = c_p dT$$

which when integrated gives

$$h_2 - h_1 = c_p (T_2 - T_1) = (5.1926\ \text{kJ/kg} \cdot \text{K})(300 - 20)\text{K} = \mathbf{1454\ kJ/kg}$$

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12-41 The entropy change of helium between two specified states is to be compared for two equations of states.

Properties For helium, $c_p = 5.1926$ kJ/kg·K and $R = 2.0769$ kJ/kg·K (Table A-2*a*).

Analysis Solving the equation of state for v gives

$$v = \frac{RT}{P} + a$$

Then,

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

The entropy differential is

$$\begin{aligned} ds &= c_p \frac{dT}{T} - \left(\frac{\partial v}{\partial T}\right)_P dP \\ &= c_p \frac{dT}{T} - R\frac{dP}{P} \end{aligned}$$

which is the same as that of an ideal gas. Integrating this result between the two states gives

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (5.1926\ \text{kJ/kg}\cdot\text{K})\ln\frac{573\ \text{K}}{293\ \text{K}} - (2.0769\ \text{kJ/kg}\cdot\text{K})\ln\frac{600\ \text{kPa}}{100\ \text{kPa}} \\ &= \mathbf{-0.2386\ kJ/kg\cdot K} \end{aligned}$$

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12-42 General expressions for Δu, Δh, and Δs for a gas whose equation of state is $P(\upsilon - a) = RT$ for an isothermal process are to be derived.

Analysis (*a*) A relation for Δu is obtained from the general relation

$$\Delta u = u_2 - u_1 = \int_{T_1}^{T_2} c_\upsilon dT + \int_{\upsilon_1}^{\upsilon_2} \left(T\left(\frac{\partial P}{\partial T}\right)_\upsilon - P \right) d\upsilon$$

The equation of state for the specified gas can be expressed as

$$P = \frac{RT}{\upsilon - a} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_\upsilon = \frac{R}{\upsilon - a}$$

Thus,

$$T\left(\frac{\partial P}{\partial T}\right)_\upsilon - P = \frac{RT}{\upsilon - a} - P = P - P = 0$$

Substituting, $\Delta u = \int_{T_1}^{T_2} c_\upsilon dT$

(*b*) A relation for Δh is obtained from the general relation

$$\Delta h = h_2 - h_1 = \int_{T_1}^{T_2} c_P dT + \int_{P_1}^{P_2} \left(\upsilon - T\left(\frac{\partial v}{\partial T}\right)_P \right) dP$$

The equation of state for the specified gas can be expressed as

$$\upsilon = \frac{RT}{P} + a \longrightarrow \left(\frac{\partial \upsilon}{\partial T}\right)_P = \frac{R}{P}$$

Thus,

$$\upsilon - T\left(\frac{\partial v}{\partial T}\right)_P = \upsilon - T\frac{R}{P} = \upsilon - (\upsilon - a) = a$$

Substituting,

$$\Delta h = \int_{T_1}^{T_2} c_p dT + \int_{P_1}^{P_2} a dP = \int_{T_1}^{T_2} c_p dT + a\left(P_2 - P_1\right)$$

(*c*) A relation for Δs is obtained from the general relation

$$\Delta s = s_2 - s_1 = \int_{T_1}^{T_2} \frac{c_p}{T} dT - \int_{P_1}^{P_2} \left(\frac{\partial \upsilon}{\partial T}\right)_P dP$$

Substituting $(\partial \upsilon / \partial T)_P = R/T$,

$$\Delta s = \int_{T_1}^{T_2} \frac{c_p}{T} dT - \int_{P_1}^{P_2} \left(\frac{R}{P}\right)_P dP = \int_{T_1}^{T_2} \frac{c_p}{T} dT - R\ln\frac{P_2}{P_1}$$

For an isothermal process $dT = 0$ and these relations reduce to

$$\Delta u = 0, \quad \Delta h = a\left(P_2 - P_1\right), \quad \text{and} \quad \Delta s = -R\ln\frac{P_2}{P_1}$$

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12-43 General expressions for $(\partial u/\partial P)_T$ and $(\partial h/\partial \nu)_T$ in terms of P, ν, and T only are to be derived.

Analysis The general relation for du is

$$du = c_\nu dT + \left(T\left(\frac{\partial P}{\partial T}\right)_\nu - P \right) d\nu$$

Differentiating each term in this equation with respect to P at T = constant yields

$$\left(\frac{\partial u}{\partial P}\right)_T = 0 + \left(T\left(\frac{\partial P}{\partial T}\right)_\nu - P \right)\left(\frac{\partial \nu}{\partial P}\right)_T = T\left(\frac{\partial P}{\partial T}\right)_\nu \left(\frac{\partial \nu}{\partial P}\right)_T - P\left(\frac{\partial \nu}{\partial P}\right)_T$$

Using the properties P, T, ν, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_\nu \left(\frac{\partial T}{\partial \nu}\right)_P \left(\frac{\partial \nu}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_\nu \left(\frac{\partial \nu}{\partial P}\right)_T = -\left(\frac{\partial \nu}{\partial T}\right)_P$$

Substituting, we get

$$\left(\frac{\partial u}{\partial P}\right)_T = -T\left(\frac{\partial \nu}{\partial T}\right)_P - P\left(\frac{\partial \nu}{\partial P}\right)_T$$

The general relation for dh is

$$dh = c_p dT + \left(\nu - T\left(\frac{\partial \nu}{\partial T}\right)_P \right) dP$$

Differentiating each term in this equation with respect to ν at T = constant yields

$$\left(\frac{\partial h}{\partial \nu}\right)_T = 0 + \left(\nu - T\left(\frac{\partial \nu}{\partial T}\right)_P \right)\left(\frac{\partial P}{\partial \nu}\right)_T = \nu\left(\frac{\partial P}{\partial \nu}\right)_T - T\left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial P}{\partial \nu}\right)_T$$

Using the properties ν, T, P, the cyclic relation can be expressed as

$$\left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial T}{\partial P}\right)_\nu \left(\frac{\partial P}{\partial \nu}\right)_T = -1 \longrightarrow \left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial P}{\partial \nu}\right)_T = -\left(\frac{\partial T}{\partial P}\right)_\nu$$

Substituting, we get

$$\left(\frac{\partial h}{\partial \nu}\right)_T = \nu\left(\frac{\partial P}{\partial \nu}\right)_T + T\left(\frac{\partial T}{\partial P}\right)_\nu$$

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12-44 It is to be shown that $c_p - c_v = T\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial T}\right)_P$.

Analysis We begin by taking the entropy to be a function of specific volume and temperature. The differential of the entropy is then

$$ds = \left(\frac{\partial s}{\partial T}\right)_v dT + \left(\frac{\partial s}{\partial v}\right)_T dv$$

Substituting $\left(\frac{\partial s}{\partial T}\right)_v = \frac{c_v}{T}$ from Eq. 12-28 and the third Maxwell equation changes this to

$$ds = \frac{c_v}{T} dT + \left(\frac{\partial P}{\partial T}\right)_v dv$$

Taking the entropy to be a function of pressure and temperature,

$$ds = \left(\frac{\partial s}{\partial T}\right)_P dT + \left(\frac{\partial s}{\partial P}\right)_T dP$$

Combining this result with $\left(\frac{\partial s}{\partial T}\right)_P = \frac{c_p}{T}$ from Eq. 12-34 and the fourth Maxwell equation produces

$$ds = \frac{c_p}{T} dT - \left(\frac{\partial v}{\partial T}\right)_P dP$$

Equating the two previous *ds* expressions and solving the result for the specific heat difference,

$$(c_p - c_v)dT = T\left(\frac{\partial v}{\partial T}\right)_P dP + \left(\frac{\partial P}{\partial T}\right)_v dv$$

Taking the pressure to be a function of temperature and volume,

$$dP = \left(\frac{\partial P}{\partial T}\right)_v dT + \left(\frac{\partial P}{\partial v}\right)_T dv$$

When this is substituted into the previous expression, the result is

$$(c_p - c_v)dT = T\left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial T}\right)_v dT + T\left[\left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial v}\right)_T \left(\frac{\partial P}{\partial T}\right)_v\right] dv$$

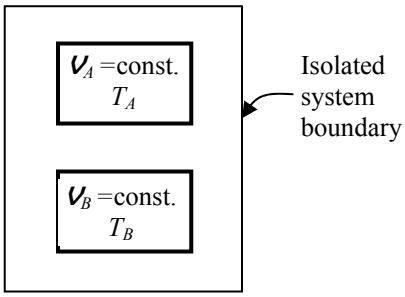
According to the cyclic relation, the term in the bracket is zero. Then, canceling the common *dT* term,

$$c_p - c_v = T\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial T}\right)_P$$

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12-45 It is to be proven that the definition for temperature $T = (\partial u / \partial s)_{v}$ reduces the net entropy change of two constant-volume systems filled with simple compressible substances to zero as the two systems approach thermal equilibrium.

Analysis The two constant-volume systems form an isolated system shown here



For the isolated system

$$dS_{\text{tot}} = dS_{\text{A}} + dS_{\text{B}} \geq 0$$

Assume $S = S(u, v)$

Then,

$$ds = \left(\frac{\partial s}{\partial u}\right)_{v} du + \left(\frac{\partial s}{\partial v}\right)_{u} dv$$

Since $v = \text{const.}$ and $dv = 0$,

$$ds = \left(\frac{\partial s}{\partial u}\right)_{v} du$$

and from the definition of temperature from the problem statement,

$$\frac{du}{(\partial u / \partial s)_{v}} = \frac{du}{T}$$

Then,

$$dS_{\text{tot}} = m_A \frac{du_A}{T_A} + m_B \frac{du_B}{T_B}$$

The first law applied to the isolated system yields

$$E_{\text{in}} - E_{\text{out}} = dU$$

$$0 = dU \longrightarrow m_A du_A + m_B du_B = 0 \longrightarrow m_B du_B = -m_A du_A$$

Now, the entropy change may be expressed as

$$dS_{\text{tot}} = m_A du_A \left(\frac{1}{T_A} - \frac{1}{T_B}\right) = m_A du_A \left(\frac{T_B - T_A}{T_A T_B}\right)$$

As the two systems approach thermal equilibrium,

$$\lim_{T_A \to T_B} dS_{\text{tot}} = 0$$

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12-46 An expression for the volume expansivity of a substance whose equation of state is $P(\nu - a) = RT$ is to be derived.

Analysis Solving the equation of state for *v* gives

$$\nu = \frac{RT}{P} + a$$

The specific volume derivative is then

$$\left(\frac{\partial \nu}{\partial T}\right)_P = \frac{R}{P}$$

The definition for volume expansivity is

$$\beta = \frac{1}{\nu}\left(\frac{\partial \nu}{\partial T}\right)_P$$

Combining these two equations gives

$$\beta = \frac{R}{RT + aP}$$

12-47 An expression for the specific heat difference of a substance whose equation of state is $P(\nu - a) = RT$ is to be derived.

Analysis The specific heat difference is expressed by

$$c_p - c_\nu = -T\left(\frac{\partial \nu}{\partial T}\right)_P^2 \left(\frac{\partial P}{\partial \nu}\right)_T$$

Solving the equation of state for specific volume,

$$\nu = \frac{RT}{P} + a$$

The specific volume derivatives are then

$$\left(\frac{\partial \nu}{\partial P}\right)_T = -\frac{RT}{P^2} \longrightarrow \left(\frac{\partial P}{\partial \nu}\right)_T = -\frac{P^2}{RT}$$

$$\left(\frac{\partial \nu}{\partial T}\right)_P = \frac{R}{P}$$

Substituting,

$$c_p - c_\nu = -T\left(\frac{R}{P}\right)^2\left(-\frac{P^2}{RT}\right) = -T\left(\frac{R^2}{P^2}\right)\left(-\frac{P^2}{RT}\right) = R$$

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12-48 An expression for the isothermal compressibility of a substance whose equation of state is $P(\nu - a) = RT$ is to be derived.

Analysis The definition for the isothermal compressibility is

$$\alpha = -\frac{1}{\nu}\left(\frac{\partial \nu}{\partial P}\right)_T$$

Solving the equation of state for specific volume,

$$\nu = \frac{RT}{P} + a$$

The specific volume derivative is then

$$\left(\frac{\partial \nu}{\partial P}\right)_T = -\frac{RT}{P^2}$$

Substituting these into the isothermal compressibility equation gives

$$\alpha = \frac{RT}{P^2}\left(\frac{P}{RT + aP}\right) = \frac{RT}{P(RT + aP)}$$

12-49 An expression for the isothermal compressibility of a substance whose equation of state is $P = \dfrac{RT}{\nu - b} - \dfrac{a}{\nu(\nu + b)T^{1/2}}$ is to be derived.

Analysis The definition for the isothermal compressibility is

$$\alpha = -\frac{1}{\nu}\left(\frac{\partial \nu}{\partial P}\right)_T$$

The derivative is

$$\left(\frac{\partial P}{\partial \nu}\right)_T = -\frac{RT}{(\nu - b)^2} + \frac{a}{T^{1/2}}\frac{2\nu + b}{\nu^2(\nu + b)^2}$$

Substituting,

$$\alpha = -\frac{1}{\nu}\left(\frac{\partial \nu}{\partial P}\right)_T = -\frac{1}{\nu}\left(\frac{1}{-\dfrac{RT}{(\nu - b)^2} + \dfrac{a}{T^{1/2}}\dfrac{2\nu + b}{\nu^2(\nu + b)^2}}\right) = -\frac{1}{-\dfrac{RT\nu}{(\nu - b)^2} + \dfrac{a}{T^{1/2}}\dfrac{2\nu + b}{(\nu + b)^2}}$$

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12-50 An expression for the volume expansivity of a substance whose equation of state is $P = \dfrac{RT}{\nu - b} - \dfrac{a}{\nu(\nu + b)T^{1/2}}$ is to be derived.

Analysis The definition for volume expansivity is

$$\beta = \frac{1}{\nu}\left(\frac{\partial \nu}{\partial T}\right)_P$$

According to the cyclic relation,

$$\left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial P}{\partial \nu}\right)_T \left(\frac{\partial T}{\partial P}\right)_\nu = -1$$

which on rearrangement becomes

$$\left(\frac{\partial \nu}{\partial T}\right)_P = -\frac{\left(\dfrac{\partial P}{\partial T}\right)_\nu}{\left(\dfrac{\partial P}{\partial \nu}\right)_T}$$

Proceeding to perform the differentiations gives

$$\left(\frac{\partial P}{\partial T}\right)_\nu = \frac{R}{\nu - b} + \frac{a}{2\nu(\nu + b)T^{3/2}}$$

and

$$\begin{aligned}\left(\frac{\partial P}{\partial \nu}\right)_T &= -\frac{RT}{(\nu - b)^2} + \frac{a}{bT^{1/2}}\left[\frac{1}{\nu^2} - \frac{1}{(\nu + b)^2}\right] \\ &= -\frac{RT}{(\nu - b)^2} + \frac{a}{T^{1/2}}\frac{2\nu + b}{\nu^2(\nu + b)^2}\end{aligned}$$

Substituting these results into the definition of the volume expansivity produces

$$\beta = -\frac{1}{\nu}\frac{\dfrac{R}{\nu - b} + \dfrac{a}{2\nu(\nu + b)T^{3/2}}}{\dfrac{-RT}{(\nu - b)^2} + \dfrac{a}{T^{1/2}}\dfrac{2\nu + b}{\nu^2(\nu + b)^2}}$$

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12-51 An expression for the volume expansivity of a substance whose equation of state is $P = \dfrac{RT}{\nu - b} - \dfrac{a}{\nu^2 T}$ is to be derived.

Analysis The definition for volume expansivity is

$$\beta = \frac{1}{\nu}\left(\frac{\partial \nu}{\partial T}\right)_P$$

According to the cyclic relation,

$$\left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial P}{\partial \nu}\right)_T \left(\frac{\partial T}{\partial P}\right)_\nu = -1$$

which on rearrangement becomes

$$\left(\frac{\partial \nu}{\partial T}\right)_P = -\frac{\left(\dfrac{\partial P}{\partial T}\right)_\nu}{\left(\dfrac{\partial P}{\partial \nu}\right)_T}$$

Proceeding to perform the differentiations gives

$$\left(\frac{\partial P}{\partial T}\right)_\nu = \frac{R}{\nu - b} + \frac{a}{\nu^2 T^2}$$

and

$$\left(\frac{\partial P}{\partial \nu}\right)_T = -\frac{RT}{(\nu - b)^2} + \frac{2a}{\nu^3 T}$$

Substituting these results into the definition of the volume expansivity produces

$$\beta = -\frac{1}{\nu}\frac{\dfrac{R}{\nu - b} + \dfrac{a}{\nu^2 T^2}}{\dfrac{-RT}{(\nu - b)^2} + \dfrac{2a}{\nu^3 T}}$$

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12-52 It is to be shown that $\beta = \alpha\left(\frac{\partial P}{\partial T}\right)_{\nu}$.

Analysis The definition for the volume expansivity is

$$\beta = \frac{1}{\nu}\left(\frac{\partial \nu}{\partial T}\right)_P$$

The definition for the isothermal compressibility is

$$\alpha = -\frac{1}{\nu}\left(\frac{\partial \nu}{\partial P}\right)_T$$

According to the cyclic relation,

$$\left(\frac{\partial \nu}{\partial T}\right)_P \left(\frac{\partial P}{\partial \nu}\right)_T \left(\frac{\partial T}{\partial P}\right)_{\nu} = -1$$

which on rearrangement becomes

$$\left(\frac{\partial \nu}{\partial T}\right)_P = -\left(\frac{\partial \nu}{\partial P}\right)_T \left(\frac{\partial P}{\partial T}\right)_{\nu}$$

When this is substituted into the definition of the volume expansivity, the result is

$$\beta = -\frac{1}{\nu}\left(\frac{\partial \nu}{\partial P}\right)_T \left(\frac{\partial P}{\partial T}\right)_{\nu}$$
$$= -\alpha\left(\frac{\partial P}{\partial T}\right)_{\nu}$$

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12-53 It is to be demonstrated that $k = \dfrac{c_p}{c_v} = -\dfrac{v\alpha}{(\partial v / \partial P)_s}$.

Analysis The relations for entropy differential are

$$ds = c_v \frac{dT}{T} + \left(\frac{\partial P}{\partial T}\right)_v dv$$

$$ds = c_p \frac{dT}{T} - \left(\frac{\partial v}{\partial T}\right)_P dP$$

For fixed *s*, these basic equations reduce to

$$c_v \frac{dT}{T} = -\left(\frac{\partial P}{\partial T}\right)_v dv$$

$$c_p \frac{dT}{T} = \left(\frac{\partial v}{\partial T}\right)_P dP$$

Also, when *s* is fixed,

$$\frac{\partial v}{\partial P} = \left(\frac{\partial v}{\partial P}\right)_s$$

Forming the specific heat ratio from these expressions gives

$$k = -\frac{\left(\dfrac{\partial v}{\partial T}\right)_P \left(\dfrac{\partial T}{\partial P}\right)_v}{\left(\dfrac{\partial v}{\partial P}\right)_s}$$

The cyclic relation is

$$\left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial v}\right)_T \left(\frac{\partial T}{\partial P}\right)_v = -1$$

Solving this for the numerator of the specific heat ratio expression and substituting the result into this numerator produces

$$k = \frac{\left(\dfrac{\partial v}{\partial P}\right)_T}{\left(\dfrac{\partial v}{\partial P}\right)_s} = -\frac{v\alpha}{\left(\dfrac{\partial v}{\partial P}\right)_s}$$

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12-54 The Helmholtz function of a substance has the form $a = -RT\ln\frac{v}{v_0} - cT_0\left(1 - \frac{T}{T_0} + \frac{T}{T_0}\ln\frac{T}{T_0}\right)$. It is to be shown how to obtain *P, h, s,* c_v, and c_p from this expression.

Analysis Taking the Helmholtz function to be a function of temperature and specific volume yields

$$da = \left(\frac{\partial a}{\partial T}\right)_v dT + \left(\frac{\partial a}{\partial v}\right)_T dv$$

while the applicable Helmholtz equation is

$$da = -Pdv - sdT$$

Equating the coefficients of the two results produces

$$P = -\left(\frac{\partial a}{\partial v}\right)_T$$

$$s = -\left(\frac{\partial a}{\partial T}\right)_v$$

Taking the indicated partial derivatives of the Helmholtz function given in the problem statement reduces these expressions to

$$P = \frac{RT}{v}$$

$$s = R\ln\frac{v}{v_0} + c\ln\frac{T}{T_0}$$

The definition of the enthalpy ($h = u + Pv$) and Helmholtz function ($a = u - Ts$) may be combined to give

$$\begin{aligned} h &= u + Pv \\ &= a + Ts + Pv \\ &= a - T\left(\frac{\partial a}{\partial T}\right)_v - v\left(\frac{\partial a}{\partial v}\right)_T \\ &= -RT\ln\frac{v}{v_0} - cT_0\left(1 - \frac{T}{T_0} + \frac{T}{T_0}\ln\frac{T}{T_0}\right) + RT\ln\frac{v}{v_0} - cT\ln\frac{T}{T_0} + RT \\ &= cT_0 + cT + RT \end{aligned}$$

According to $\left(\frac{\partial s}{\partial T}\right)_v = \frac{c_v}{T}$ given in the text (Eq. 12-28),

$$c_v = T\left(\frac{\partial s}{\partial T}\right)_v = T\frac{c}{T} = c$$

The preceding expression for the temperature indicates that the equation of state for the substance is the same as that of an ideal gas. Then,

$$c_p = R + c_v = R + c$$

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The Joule-Thomson Coefficient

12-55C It represents the variation of temperature with pressure during a throttling process.

12-56C The line that passes through the peak points of the constant enthalpy lines on a *T-P* diagram is called the inversion line. The maximum inversion temperature is the highest temperature a fluid can be cooled by throttling.

12-57C No. The temperature may even increase as a result of throttling.

12-58C Yes.

12-59C No. Helium is an ideal gas and $h = h(T)$ for ideal gases. Therefore, the temperature of an ideal gas remains constant during a throttling (h = constant) process.



12-60E The Joule-Thompson coefficient of nitrogen at two states is to be estimated.

Analysis (*a*) The enthalpy of nitrogen at 120 psia and 350 R is, from EES, h = 84.88 Btu/lbm. Approximating differentials by differences about the specified state, the Joule-Thomson coefficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=84.88\ \text{Btu/lbm}}$$

Considering a throttling process from 130 psia to 110 psia at h = 84.88 Btu/lbm, the Joule-Thomson coefficient is determined to be

$$\mu = \left(\frac{T_{110\ \text{psia}} - T_{130\ \text{psia}}}{(110-130)\ \text{psia}}\right)_{h=84.88\ \text{Btu/lbm}} = \frac{(349.40-350.60)\ \text{R}}{(110-130)\ \text{psia}} = \mathbf{0.0599\ R/psia}$$

(*b*) The enthalpy of nitrogen at 1200 psia and 700 R is, from EES, h = 170.14 Btu/lbm. Approximating differentials by differences about the specified state, the Joule-Thomson coefficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=170.14\text{Btu/lbm}}$$

Considering a throttling process from 1210 psia to 1190 psia at h = 170.14 Btu/lbm, the Joule-Thomson coefficient is determined to be

$$\mu = \left(\frac{T_{1190\ \text{psia}} - T_{1210\ \text{psia}}}{(1190-1210)\ \text{psia}}\right)_{h=170.14\ \text{Btu/lbm}} = \frac{(699.91-700.09)\ \text{R}}{(1190-1210)\ \text{psia}} = \mathbf{0.00929\ R/psia}$$

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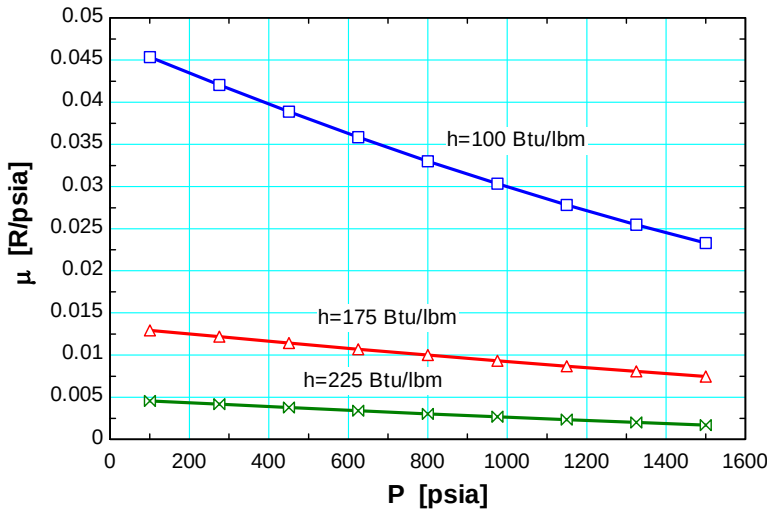
12-61E Problem 12-60E is reconsidered. The Joule-Thompson coefficient for nitrogen over the pressure range 100 to 1500 psia at the enthalpy values 100, 175, and 225 Btu/lbm is to be plotted.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
Gas$ = 'Nitrogen'
{P_ref=120 [psia]
T_ref=350 [R]
P= P_ref}
h=100 [Btu/lbm]
{h=enthalpy(Gas$, T=T_ref, P=P_ref)}
dP = 10 [psia]
T = temperature(Gas$, P=P, h=h)
P[1] = P + dP
P[2] = P - dP
T[1] = temperature(Gas$, P=P[1], h=h)
T[2] = temperature(Gas$, P=P[2], h=h)
Mu = DELTAT/DELTAP  "Approximate the differential by differences about the state at h=const."
DELTAT=T[2]-T[1]
DELTAP=P[2]-P[1]
```

h = 225 Btu/lbm

P [psia]	μ [R/psia]
100	0.004573
275	0.00417
450	0.003781
625	0.003405
800	0.003041
975	0.002688
1150	0.002347
1325	0.002015
1500	0.001694



12-62 Steam is throttled slightly from 1 MPa and 300°C. It is to be determined if the temperature of the steam will increase, decrease, or remain the same during this process.

Analysis The enthalpy of steam at 1 MPa and $T = 300$°C is $h = 3051.6$ kJ/kg. Now consider a throttling process from this state to 0.8 MPa, which is the next lowest pressure listed in the tables. The temperature of the steam at the end of this throttling process will be

$$\left.\begin{array}{l} P = 0.8 \text{ MPa} \\ h = 3051.6 \text{ kJ/kg} \end{array}\right\} T_2 = 297.52°\text{C}$$

Therefore, the temperature will **decrease**.

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12-63E The Joule-Thomson coefficient of refrigerant-134a at a given state is to be estimated.

Analysis The Joule-Thomson coefficient is defined as

$$\mu = \left(\frac{\partial T}{\partial P} \right)_h$$

We use a finite difference approximation as

$$\mu \cong \frac{T_2 - T_1}{P_2 - P_1} \quad \text{(at constant enthalpy)}$$

At the given state (we call it state 1), the enthalpy of R-134a is

$$\left.\begin{array}{l} P_1 = 40 \text{ psia} \\ T_1 = 60°\text{F} \end{array}\right\} h_1 = 113.79 \text{ Btu/lbm} \quad \text{(Table A - 13E)}$$

The second state will be selected for a pressure of 30 psia. At this pressure and the same enthalpy, we have

$$\left.\begin{array}{l} P_2 = 30 \text{ psia} \\ h_2 = h_1 = 113.79 \text{ kJ/kg} \end{array}\right\} T_2 = 56.78°\text{F} \quad \text{(Table A - 13E)}$$

Substituting,

$$\mu \cong \frac{T_2 - T_1}{P_2 - P_1} = \frac{(56.78 - 60)\text{R}}{(30 - 40)\text{psia}} = \mathbf{0.322 \text{ R/psia}}$$

12-64 The Joule-Thomson coefficient of refrigerant-134a at a given state is to be estimated.

Analysis The Joule-Thomson coefficient is defined as

$$\mu = \left(\frac{\partial T}{\partial P} \right)_h$$

We use a finite difference approximation as

$$\mu \cong \frac{T_2 - T_1}{P_2 - P_1} \quad \text{(at constant enthalpy)}$$

At the given state (we call it state 1), the enthalpy of R-134a is

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 90°\text{C} \end{array}\right\} h_1 = 333.93 \text{ kJ/kg} \quad \text{(Table A - 13)}$$

The second state will be selected for a pressure of 180 kPa. At this pressure and the same enthalpy, we have

$$\left.\begin{array}{l} P_2 = 180 \text{ kPa} \\ h_2 = h_1 = 333.93 \text{ kJ/kg} \end{array}\right\} T_2 = 89.78°\text{C} \quad \text{(Table A - 13)}$$

Substituting,

$$\mu \cong \frac{T_2 - T_1}{P_2 - P_1} = \frac{(89.78 - 90)\text{K}}{(180 - 200)\text{kPa}} = \mathbf{0.0110 \text{ K/kPa}}$$

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12-65 The equation of state of a gas is given by $\nu = \frac{RT}{P} - \frac{bP}{T^2}$. An equation for the Joule-Thomson coefficient inversion line using this equation is to be derived.

Analysis From Eq. 12-52 of the text,

$$c_p = \frac{1}{\mu}\left[T\left(\frac{\partial \nu}{\partial T}\right)_P - \nu\right]$$

When $\mu = 0$ as it does on the inversion line, this equation becomes

$$T\left(\frac{\partial \nu}{\partial T}\right)_P = \nu$$

Using the equation of state to evaluate the partial derivative,

$$\left(\frac{\partial \nu}{\partial T}\right)_P = \frac{R}{P} + 2\frac{bP}{T^3}$$

Substituting this result into the previous expression produces

$$\frac{RT}{P} + 2\frac{bP}{T^2} = \frac{RT}{P} - \frac{bP}{T^2} \longrightarrow 3\frac{bP}{T^2} = 0$$

The condition along the inversion line is then

$$P = 0$$

12-66 It is to be demonstrated that the Joule-Thomson coefficient is given by $\mu = \frac{T^2}{c_p}\left(\frac{\partial(\nu/T)}{\partial T}\right)_P$.

Analysis From Eq. 12-52 of the text,

$$c_p = \frac{1}{\mu}\left[T\left(\frac{\partial \nu}{\partial T}\right)_P - \nu\right]$$

Expanding the partial derivative of ν/T produces

$$\left(\frac{\partial \nu/T}{\partial T}\right)_P = \frac{1}{T}\left(\frac{\partial \nu}{\partial T}\right)_P - \frac{\nu}{T^2}$$

When this is multiplied by T^2, the right-hand side becomes the same as the bracketed quantity above. Then,

$$\mu = \frac{T^2}{c_p}\left(\frac{\partial(\nu/T)}{\partial T}\right)_P$$

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12-67 The most general equation of state for which the Joule-Thomson coefficient is always zero is to be determined.

Analysis From Eq. 12-52 of the text,

$$c_p = \frac{1}{\mu}\left[T\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P - \boldsymbol{v}\right]$$

When $\mu = 0$, this equation becomes

$$\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P = \frac{\boldsymbol{v}}{T}$$

This can only be satisfied by an equation of state of the form

$$\frac{\boldsymbol{v}}{T} = f(P)$$

where $f(P)$ is an arbitrary function of the pressure.

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The *dh*, *du*, and *ds* of Real Gases

12-68C It is the variation of enthalpy with pressure at a fixed temperature.

12-69C As P_R approaches zero, the gas approaches ideal gas behavior. As a result, the deviation from ideal gas behavior diminishes.

12-70C So that a single chart can be used for all gases instead of a single particular gas.

12-71 The errors involved in the enthalpy and internal energy of CO_2 at 350 K and 10 MPa if it is assumed to be an ideal gas are to be determined.

Analysis (*a*) The enthalpy departure of CO_2 at the specified state is determined from the generalized chart to be (Fig. A-29)

and
$$\left.\begin{aligned} T_R &= \frac{T}{T_{\text{cr}}} = \frac{350}{304.2} = 1.151 \\ P_R &= \frac{P}{P_{\text{cr}}} = \frac{10}{7.39} = 1.353 \end{aligned}\right\} \longrightarrow Z_h = \frac{(\bar{h}_{\text{ideal}} - \bar{h})_{T,P}}{R_u T_{\text{cr}}} = 1.5$$

CO_2
350 K
10 MPa

Thus,

$$\bar{h} = \bar{h}_{\text{ideal}} - Z_h R_u T_{\text{cr}} = 11{,}351 - [(1.5)(8.314)(304.2)] = 7{,}557 \text{kJ/kmol}$$

and,

$$\text{Error} = \frac{(\bar{h}_{\text{ideal}} - \bar{h})_{T,P}}{\bar{h}} = \frac{11{,}351 - 7{,}557}{7{,}557} = \mathbf{50.2}\%$$

(*b*) At the calculated T_R and P_R the compressibility factor is determined from the compressibility chart to be Z = 0.65. Then using the definition of enthalpy, the internal energy is determined to be

$$\bar{u} = \bar{h} - P\bar{v} = \bar{h} - ZR_u T = 7557 - [(0.65)(8.314)(350)] = 5{,}666 \text{kJ/kmol}$$

and,

$$\text{Error} = \frac{\bar{u}_{\text{ideal}} - \bar{u}}{\bar{u}} = \frac{8{,}439 - 5{,}666}{5{,}666} = \mathbf{48.9\%}$$

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12-72 The enthalpy and entropy changes of nitrogen during a process are to be determined assuming ideal gas behavior and using generalized charts.

Analysis (*a*) Using data from the ideal gas property table of nitrogen (Table A-18),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 9306 - 6{,}537 = \mathbf{2769\ kJ/kmol}$$

and

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln\frac{P_2}{P_1} = 193.562 - 183.289 - 8.314 \times \ln\frac{12}{6} = \mathbf{4.510\ kJ/kmol \cdot K}$$

(*b*) The enthalpy and entropy departures of nitrogen at the specified states are determined from the generalized charts to be (Figs. A-29, A-30)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{225}{126.2} = 1.783 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{6}{3.39} = 1.770 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.6 \text{ and } Z_{s1} = 0.25$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{320}{126.2} = 2.536 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{12}{3.39} = 2.540 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.4 \text{ and } Z_{s2} = 0.15$$

Substituting,

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= R_u T_{cr}(Z_{h1} - Z_{h2}) + (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} \\ &= (8.314)(126.2)(0.6 - 0.4) + 2769 \\ &= \mathbf{2979\ kJ/kmol} \end{aligned}$$

$$\begin{aligned} \bar{s}_2 - \bar{s}_1 &= R_u(Z_{s1} - Z_{s2}) + (\bar{s}_2 - \bar{s}_1)_{\text{ideal}} \\ &= (8.314)(0.25 - 0.15) + 4.510 \\ &= \mathbf{5.341\ kJ/kmol \cdot K} \end{aligned}$$

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12-73E The enthalpy and entropy changes of water vapor during a change of state are to be determined using the departure charts and the property tables.

Properties The properties of water are (Table A-1E)

$$M = 18.015 \text{ lbm/lbmol},\ T_{cr} = 1164.8 \text{ R},\ P_{cr} = 3200 \text{ psia}$$

Analysis (*a*) The pressure of water vapor during this process is

$$P_1 = P_2 = P_{\text{sat @ 500°F}} = 680.56 \text{ psia}$$

Using data from the ideal gas property table of water vapor (Table A-23),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 12{,}178.8 - 7738.0 = 4440.8 \text{ Btu/lbmol}$$

and

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln \frac{P_2}{P_1} = 53.556 - 49.843 - 0 = 3.713 \text{ Btu/lbmol} \cdot \text{R}$$

The enthalpy and entropy departures of water vapor at the specified states are determined from the generalized charts to be (Figs. A-29, A-30 or from EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{960}{1164.8} = 0.824 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{680.56}{3200} = 0.213 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.340 \text{ and } Z_{s1} = 0.277$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{cr}} = \frac{1460}{1164.8} = 1.253 \\ P_{R2} &= \frac{P_2}{P_{cr}} = \frac{680.56}{3200} = 0.213 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.157 \text{ and } Z_{s2} = 0.0903$$

The enthalpy and entropy changes per mole basis are

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} - R_u T_{cr} (Z_{h2} - Z_{h1}) \\ &= 4440.8 - (1.9858)(1164.8)(0.157 - 0.340) = 4864 \text{ kJ/kmol} \end{aligned}$$

$$\begin{aligned} \bar{s}_2 - \bar{s}_1 &= (\bar{s}_2 - \bar{s}_1)_{\text{ideal}} - R_u (Z_{s2} - Z_{s1}) \\ &= 3.713 - (1.9858)(0.0903 - 0.277) = 4.084 \text{ Btu/lbmol} \cdot \text{R} \end{aligned}$$

The enthalpy and entropy changes per mass basis are

$$h_2 - h_1 = \frac{\bar{h}_2 - \bar{h}_1}{M} = \frac{4864 \text{ Btu/lbmol}}{18.015 \text{ lbm/lbmol}} = \mathbf{270.0 \text{ Btu/lbm}}$$

$$s_2 - s_1 = \frac{\bar{s}_2 - \bar{s}_1}{M} = \frac{4.084 \text{ Btu/lbmol} \cdot \text{K}}{18.015 \text{ lbm/lbmol}} = \mathbf{0.2267 \text{ Btu/lbm} \cdot \text{R}}$$

(*b*) The inlet and exit state properties of water are

$$\left.\begin{aligned} T_1 &= 500°\text{F} \\ x_1 &= 1 \end{aligned}\right\} \begin{aligned} h_1 &= 1202.3 \text{ Btu/lbm} \\ s_1 &= 1.4334 \text{ Btu/lbm} \cdot \text{R} \end{aligned} \quad \text{(Table A-4E)}$$

$$\left.\begin{aligned} P_2 &= 680.56 \text{ psia} \\ T_2 &= 1000°\text{F} \end{aligned}\right\} \begin{aligned} h_2 &= 1515.7 \text{ Btu/lbm} \\ s_2 &= 1.7008 \text{ Btu/lbm} \cdot \text{R} \end{aligned} \quad \text{(from EES)}$$

The enthalpy and entropy changes are

$$h_2 - h_1 = 1515.7 - 1202.3 = \mathbf{313.4 \text{ Btu/lbm}}$$

$$s_2 - s_1 = 1.7008 - 1.4334 = \mathbf{0.2674 \text{ Btu/lbm} \cdot \text{R}}$$

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12-74E The enthalpy and entropy changes of water vapor during a change of state are to be determined using the departure charts and the property tables.

Properties The properties of water are (Table A-1E)

$$M = 18.015 \text{ lbm/lbmol},\ T_{\text{cr}} = 1164.8 \text{ R},\ P_{\text{cr}} = 3200 \text{ psia}$$

Analysis (*a*) Using data from the ideal gas property table of water vapor (Table A-23E),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 12{,}178.8 - 17{,}032.5 = -4853.7 \text{ Btu/lbmol}$$

and

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln \frac{P_2}{P_1} = 53.556 - 56.411 - 1.9858 \times \ln \frac{1000}{3000} = -0.6734 \text{ Btu/lbmol} \cdot \text{R}$$

The enthalpy and entropy departures of water vapor at the specified states are determined from the generalized charts to be (Figs. A-29, A-30 or from EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{1960}{1164.8} = 1.683 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{3000}{3200} = 0.9375 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.387 \text{ and } Z_{s1} = 0.188$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{1460}{1164.8} = 1.253 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{1000}{3200} = 0.3125 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.233 \text{ and } Z_{s2} = 0.134$$

The enthalpy and entropy changes per mole basis are

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} - R_u T_{\text{cr}} (Z_{h2} - Z_{h1}) \\ &= -4853.7 - (1.9858)(1164.8)(0.233 - 0.387) = -4497.5 \text{ Btu/lbmol} \end{aligned}$$

$$\begin{aligned} \bar{s}_2 - \bar{s}_1 &= (\bar{s}_2 - \bar{s}_1)_{\text{ideal}} - R_u (Z_{s2} - Z_{s1}) \\ &= -0.6734 - (1.9858)(0.134 - 0.188) = -0.5662 \text{ Btu/lbmol} \cdot \text{R} \end{aligned}$$

The enthalpy and entropy changes per mass basis are

$$h_2 - h_1 = \frac{\bar{h}_2 - \bar{h}_1}{M} = \frac{-4497.5 \text{ Btu/lbmol}}{18.015 \text{ lbm/lbmol}} = \mathbf{-249.7 \text{ Btu/lbm}}$$

$$s_2 - s_1 = \frac{\bar{s}_2 - \bar{s}_1}{M} = \frac{-0.5662 \text{ Btu/lbmol} \cdot \text{R}}{18.015 \text{ lbm/lbmol}} = \mathbf{-0.0314 \text{ Btu/lbm} \cdot \text{R}}$$

(*b*) Using water tables (Table A-6E)

$$\left.\begin{aligned} P_1 &= 3000 \text{ psia} \\ T_1 &= 1500°\text{F} \end{aligned}\right\} \begin{aligned} h_1 &= 1764.6 \text{ Btu/lbm} \\ s_1 &= 1.6883 \text{ Btu/lbm} \cdot \text{R} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 1000 \text{ psia} \\ T_2 &= 1000°\text{F} \end{aligned}\right\} \begin{aligned} h_2 &= 1506.2 \text{ Btu/lbm} \\ s_2 &= 1.6535 \text{ Btu/lbm} \cdot \text{R} \end{aligned}$$

The enthalpy and entropy changes are

$$h_2 - h_1 = 1506.2 - 1764.6 = \mathbf{-258.4 \text{ Btu/lbm}}$$

$$s_2 - s_1 = 1.6535 - 1.6883 = \mathbf{-0.0348 \text{ Btu/lbm} \cdot \text{R}}$$

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12-75 The enthalpy and entropy changes of water vapor during a change of state are to be determined using the departure charts and the property tables.

Properties The properties of water are (Table A-1)

$$M = 18.015\ \text{kg/kmol},\ T_{\text{cr}} = 647.1\ \text{K},\ P_{\text{cr}} = 22.06\ \text{MPa}$$

Analysis Using data from the ideal gas property table of water vapor (Table A-23),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 23{,}082 - 30{,}754 = -7672\ \text{kJ/kmol}$$

and

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln\frac{P_2}{P_1} = 217.141 - 227.109 - 8.314 \times \ln\frac{500}{1000} = -4.2052\ \text{kJ/kmol}\cdot\text{K}$$

The enthalpy and entropy departures of water vapor at the specified states are determined from the generalized charts to be (Figs. A-29, A-30 or from EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{873}{647.1} = 1.349 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{1}{22.06} = 0.0453 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.0288 \text{ and } Z_{s1} = 0.0157$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{673}{647.1} = 1.040 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{0.5}{22.06} = 0.0227 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.0223 \text{ and } Z_{s2} = 0.0146$$

The enthalpy and entropy changes per mole basis are

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} - R_u T_{\text{cr}} (Z_{h2} - Z_{h1}) \\ &= -7672 - (8.314)(647.1)(0.0223 - 0.0288) = -7637\ \text{kJ/kmol} \end{aligned}$$

$$\begin{aligned} \bar{s}_2 - \bar{s}_1 &= (\bar{s}_2 - \bar{s}_1)_{\text{ideal}} - R_u (Z_{s2} - Z_{s1}) \\ &= -4.2052 - (8.314)(0.0146 - 0.0157) = -4.1961\ \text{kJ/kmol}\cdot\text{K} \end{aligned}$$

The enthalpy and entropy changes per mass basis are

$$h_2 - h_1 = \frac{\bar{h}_2 - \bar{h}_1}{M} = \frac{-7637\ \text{kJ/kmol}}{18.015\ \text{kg/kmol}} = \mathbf{-423.9\ kJ/kg}$$

$$s_2 - s_1 = \frac{\bar{s}_2 - \bar{s}_1}{M} = \frac{-4.1961\ \text{kJ/kmol}\cdot\text{K}}{18.015\ \text{kg/kmol}} = \mathbf{-0.2329\ kJ/kg}\cdot\mathbf{K}$$

The inlet and exit state properties of water vapor from Table A-6 are

$$\left.\begin{aligned} P_1 &= 1000\ \text{kPa} \\ T_1 &= 600°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 3698.6\ \text{kJ/kg} \\ s_1 &= 8.0311\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 500\ \text{kPa} \\ T_2 &= 400°\text{C} \end{aligned}\right\} \begin{aligned} h_2 &= 3272.4\ \text{kJ/kg} \\ s_2 &= 7.7956\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

The enthalpy and entropy changes are

$$h_2 - h_1 = 3272.4 - 3698.6 = \mathbf{-426.2\ kJ/kg}$$

$$s_2 - s_1 = 7.7956 - 8.0311 = \mathbf{-0.2355\ kJ/kg}\cdot\mathbf{K}$$

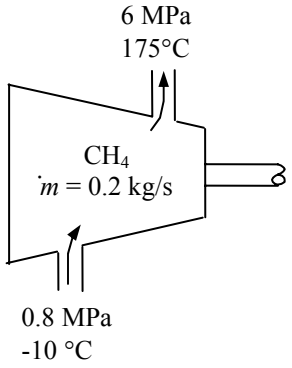
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12-76 Methane is compressed adiabatically by a steady-flow compressor. The required power input to the compressor is to be determined using the generalized charts.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The steady-flow energy balance equation for this compressor can be expressed as

$$\begin{aligned}\dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0\\ \dot{E}_{\text{in}} &= \dot{E}_{\text{out}}\\ \dot{W}_{\text{C,in}} + \dot{m}h_1 &= \dot{m}h_2\\ \dot{W}_{\text{C,in}} &= \dot{m}(h_2 - h_1)\end{aligned}$$



The enthalpy departures of CH_4 at the specified states are determined from the generalized charts to be (Fig. A-29)

$$\left.\begin{aligned}T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{263}{191.1} = 1.376\\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{0.8}{4.64} = 0.172\end{aligned}\right\} \longrightarrow Z_{h1} = 0.075$$

and

$$\left.\begin{aligned}T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{448}{191.1} = 2.34\\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{6}{4.64} = 1.29\end{aligned}\right\} \longrightarrow Z_{h2} = 0.25$$

Thus,

$$\begin{aligned}h_2 - h_1 &= RT_{\text{cr}}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{\text{ideal}}\\ &= (0.5182)(191.1)(0.075 - 0.25) + 2.2537(175 - (-10))\\ &= 399.6 \text{ kJ/kg}\end{aligned}$$

Substituting,

$$\dot{W}_{\text{C,in}} = (0.2 \text{ kg/s})(399.6 \text{ kJ/kg}) = \mathbf{79.9\ kW}$$

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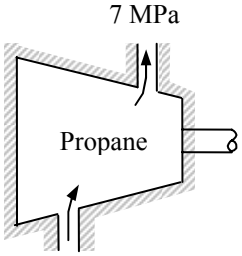
12-77 Propane is to be adiabatically and reversibly compressed in a steady-flow device. The specific work required for this compression is to be determined using the departure charts and treating the propane as an ideal gas with temperature variable specific heats.

Properties The properties of propane are (Table A-1)

$$M = 44.097 \text{ kg/kmol},\ R = 0.1885 \text{ kJ/kg}\cdot\text{K},\ T_{cr} = 370 \text{ K},\ P_{cr} = 4.26 \text{ MPa}$$

Analysis The temperature at the exit state may be determined by the fact that the process is isentropic and the molar entropy change between the inlet and exit is zero. When the entropy change equation is integrated with variable specific heats, it becomes

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = \int_1^2 \frac{c_p}{T}\,dT - R_u \ln\frac{P_2}{P_1}$$



When the expression of Table A-2c is substituted for c_p and the integration performed, we obtain

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = \int_1^2 \frac{c_p}{T}\,dT - R_u \ln\frac{P_2}{P_1} = \int_1^2 \left(\frac{a}{T} + b + cT + dT^2\right)dT - R_u \ln\frac{P_2}{P_1}$$

$$= a\ln\frac{T_2}{T_1} + b(T_2 - T_1) + \frac{c}{2}(T_2^2 - T_1^2) + \frac{d}{3}(T_2^3 - T_1^3) - R_u \ln\frac{P_2}{P_1}$$

$$0 = -4.04\ln\frac{T_2}{450} + 30.48\left(\frac{T_2}{100} - 4.50\right) - 0.786\left[\left(\frac{T_2}{100}\right)^2 - 4.50^2\right] + 0.01058\left[\left(\frac{T_2}{100}\right)^3 - 4.50^3\right]$$

$$- (8.314)\ln\frac{7000}{750}$$

Solving this equation by EES or an iterative solution by hand gives

$$T_2 = 532 \text{ K}$$

When en energy balance is applied to the compressor, it becomes

$$\overline{w}_{\text{in}} = (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \int_1^2 c_p\,dT = \int_1^2 (a + bT + cT^2 + dT^3)\,dT$$

$$= a(T_2 - T_1) + \frac{b}{2}(T_2^2 - T_1^2) + \frac{c}{3}(T_2^3 - T_1^3) + \frac{d}{4}(T_2^4 - T_1^4)$$

$$= -4.04(532 - 450) + 0.1524(532^2 - 450^2) - 52.4(5.32^3 - 4.50^3) + 0.7935(5.32^4 - 4.50^4)$$

$$= 9111 \text{ kJ/kmol}$$

The work input per unit mass basis is

$$w_{\text{in}} = \frac{\overline{w}_{\text{in}}}{M} = \frac{9111 \text{ kJ/kmol}}{44.097 \text{ kg/kmol}} = \mathbf{207 \text{ kJ/kg}}$$

The enthalpy departures of propane at the specified states are determined from the generalized charts to be (Fig. A-29 or from EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{450}{370} = 1.22 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{0.5}{4.26} = 0.176 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.136$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{cr}} = \frac{532}{370} = 1.44 \\ P_{R2} &= \frac{P_2}{P_{cr}} = \frac{7}{4.26} = 1.64 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.971$$

The work input (i.e., enthalpy change) is determined to be

$$w_{\text{in}} = h_2 - h_1 = (h_2 - h_1)_{\text{ideal}} - RT_{cr}(Z_{h2} - Z_{h1})$$

$$= 207 - (0.1885)(370)(0.971 - 0.136)$$

$$= \mathbf{148 \text{ kJ/kg}}$$

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12-78E Oxygen is to be adiabatically and reversibly expanded in a nozzle. The exit velocity is to be determined using the departure charts and treating the oxygen as an ideal gas with temperature variable specific heats.

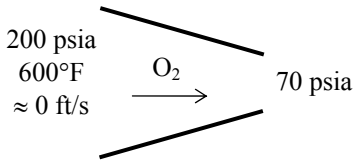
Properties The properties of oxygen are (Table A-1)

$$M = 31.999 \text{ lbm/lbmol},\ R = 0.06206 \text{ Btu/lbm} \cdot \text{R},\ T_{\text{cr}} = 278.6 \text{ R},\ P_{\text{cr}} = 736 \text{ psia}$$

Analysis The temperature at the exit state may be determined by the fact that the process is isentropic and the molar entropy change between the inlet and exit is zero. From the entropy change equation for an ideal gas with variable specific heats:

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = 0$$

$$s_2^\circ - s_1^\circ = R_u \ln \frac{P_2}{P_1} = (1.9858) \ln \frac{70}{200} = -2.085 \text{ Btu/lbmol} \cdot \text{R}$$



Then from Table A-19E,

$$T_1 = 1060 \text{ R} \longrightarrow \bar{h}_{1,\text{ideal}} = 7543.6 \text{ Btu/lbmol},\ s_1^\circ = 53.921 \text{ Btu/lbmol} \cdot \text{R}$$

$$s_2^\circ = s_1^\circ - 2.085 = 53.921 - 2.085 = 51.836 \text{ Btu/lbmol} \cdot \text{R}$$

$$s_2^\circ = 51.836 \text{ Btu/lbmol} \cdot \text{R} \longrightarrow T_2 = 802 \text{ R},\ \bar{h}_{2,\text{ideal}} = 5614.1 \text{ Btu/lbmol}$$

The enthalpy change per mole basis is

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 5614.1 - 7543.6 = -1929.5 \text{ Btu/lbmol}$$

The enthalpy change per mass basis is

$$(h_2 - h_1)_{\text{ideal}} = \frac{(\bar{h}_2 - \bar{h}_1)_{\text{ideal}}}{M} = \frac{-1929.5 \text{ Btu/lbmol}}{31.999 \text{ lbm/lbmol}} = -60.30 \text{ Btu/lbm}$$

An energy balance on the nozzle gives

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

Solving for the exit velocity,

$$V_2 = \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} = \left[(0 \text{ ft/s})^2 + 2(60.30 \text{ Btu/lbm})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)\right]^{0.5} = \mathbf{1738 \text{ ft/s}}$$

The enthalpy departures of oxygen at the specified states are determined from the generalized charts to be (Fig. A-29 or from EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{1060}{278.6} = 3.805 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{200}{736} = 0.272 \end{aligned}\right\} Z_{h1} = 0.000759 \qquad \left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{802}{278.6} = 2.879 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{70}{736} = 0.0951 \end{aligned}\right\} Z_{h2} = 0.00894$$

The enthalpy change is

$$\begin{aligned} h_2 - h_1 &= (h_2 - h_1)_{\text{ideal}} - RT_{\text{cr}}(Z_{h2} - Z_{h1}) \\ &= -60.30 \text{ Btu/lbm} - (0.06206 \text{ Btu/lbm} \cdot \text{R})(278.6 \text{ R})(0.00894 - 0.000759) \\ &= -60.44 \text{ Btu/lbm} \end{aligned}$$

The exit velocity is

$$V_2 = \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} = \left[(0 \text{ ft/s})^2 + 2(60.44 \text{ Btu/lbm})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)\right]^{0.5} = \mathbf{1740 \text{ ft/s}}$$

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12-79 Propane is compressed isothermally by a piston-cylinder device. The work done and the heat transfer are to be determined using the generalized charts.

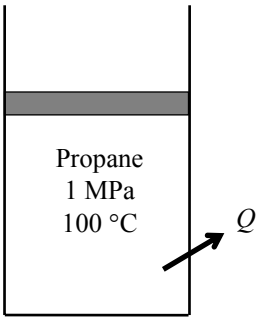
Assumptions **1** The compression process is quasi-equilibrium. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) The enthalpy departure and the compressibility factors of propane at the initial and the final states are determined from the generalized charts to be (Figs. A-29, A-15)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{373}{370} = 1.008 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{1}{4.26} = 0.235 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.28 \text{ and } Z_1 = 0.92$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{373}{370} = 1.008 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{4}{4.26} = 0.939 \end{aligned}\right\} \longrightarrow Z_{h2} = 1.8 \text{ and } Z_2 = 0.50$$



Treating propane as a real gas with $Z_{\text{avg}} = (Z_1+Z_2)/2 = (0.92 + 0.50)/2 = 0.71$,

$$P\nu = ZRT \cong Z_{\text{avg}}RT = C = \text{constant}$$

Then the boundary work becomes

$$w_{b,\text{in}} = -\int_1^2 P\,d\nu = -\int_1^2 \frac{C}{\nu}\,d\nu = -C\ln\frac{\nu_2}{\nu_1} = Z_{\text{avg}}RT\ln\frac{Z_2RT/P_2}{Z_1RT/P_1} = -Z_{\text{ave}}RT\ln\frac{Z_2P_1}{Z_1P_2}$$

$$= -(0.71)(0.1885 \text{ kJ/kg}\cdot\text{K})(373 \text{ K})\ln\frac{(0.50)(1)}{(0.92)(4)} = \mathbf{99.6\ kJ/kg}$$

Also,

$$h_2 - h_1 = RT_{cr}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{\text{ideal}} = (0.1885)(370)(0.28 - 1.8) + 0 = -106 \text{ kJ/kg}$$

$$u_2 - u_1 = (h_2 - h_1) - R(Z_2T_2 - Z_1T_1) = -106 - (0.1885)[(0.5)(373) - (0.92)(373)] = -76.5 \text{ kJ/kg}$$

Then the heat transfer for this process is determined from the closed system energy balance to be

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$q_{\text{in}} + w_{b,\text{in}} = \Delta u = u_2 - u_1$$

$$q_{\text{in}} = (u_2 - u_1) - w_{b,\text{in}} = -76.5 - 99.6 = -176.1 \text{ kJ/kg} \quad \rightarrow \quad q_{\text{out}} = 176.1 \text{ kJ/kg}$$

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12-80 Problem 12-79 is reconsidered. This problem is to be extended to compare the solutions based on the ideal gas assumption, generalized chart data and real fluid (EES) data. Also, the solution is to be extended to carbon dioxide, nitrogen and methane.

Analysis The problem is solved using EES, and the solution is given below.

```
Procedure INFO(Name$, T[1] : Fluid$, T_critical, p_critical)
If Name$='Propane' then
        T_critical=370 ; p_critical=4620 ; Fluid$='C3H8'; goto 10
endif
If Name$='Methane' then
        T_critical=191.1 ; p_critical=4640 ; Fluid$='CH4'; goto 10
endif
If Name$='Nitrogen' then
        T_critical=126.2 ; p_critical=3390 ; Fluid$='N2'; goto 10
endif
If Name$='Oxygen' then
        T_critical=154.8 ; p_critical=5080 ; Fluid$='O2'; goto 10
endif
If Name$='CarbonDioxide' then
        T_critical=304.2 ; p_critical=7390 ; Fluid$='CO2' ; goto 10
endif
If Name$='n-Butane' then
        T_critical=425.2 ; p_critical=3800 ; Fluid$='C4H10' ; goto 10
endif

10:
  If T[1]<=T_critical then
CALL ERROR('The supplied temperature must be greater than the critical temperature for the fluid.  A value of
XXXF1 K was supplied',T[1])
endif

end

{"Data from the Diagram Window"
T[1]=100+273.15
p[1]=1000
p[2]=4000
Name$='Propane'
Fluid$='C3H8' }

Call INFO(Name$, T[1] : Fluid$, T_critical, p_critical)
R_u=8.314
M=molarmass(Fluid$)
R=R_u/M

"****** IDEAL GAS SOLUTION ******"
"State 1"
h_ideal[1]=enthalpy(Fluid$, T=T[1])   "Enthalpy of ideal gas"
s_ideal[1]=entropy(Fluid$, T=T[1], p=p[1]) "Entropy of ideal gas"
u_ideal[1]=h_ideal[1]-R*T[1]          "Internal energy of ideal gas"
"State 2"
h_ideal[2]=enthalpy(Fluid$, T=T[2])   "Enthalpy of ideal gas"
s_ideal[2]=entropy(Fluid$, T=T[2], p=p[2]) "Entropy of ideal gas"
u_ideal[2]=h_ideal[2]-R*T[2]          "Internal energy of ideal gas"

"Work is the integral of p dv, which can be done analytically."
w_ideal=R*T[1]*Ln(p[1]/p[2])
```

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```
"First Law - note that u_ideal[2] is equal to u_ideal[1]"
q_ideal-w_ideal=u_ideal[2]-u_ideal[1]

"Entropy change"
DELTAs_ideal=s_ideal[2]-s_ideal[1]

"****** COMPRESSABILITY CHART  SOLUTION ******"
"State 1"
Tr[1]=T[1]/T_critical
pr[1]=p[1]/p_critical
Z[1]=COMPRESS(Tr[1], Pr[1])
DELTAh[1]=ENTHDEP(Tr[1], Pr[1])*R*T_critical "Enthalpy departure"
h[1]=h_ideal[1]-DELTAh[1]              "Enthalpy of real gas using charts"
u[1]=h[1]-Z[1]*R*T[1]
        "Internal energy of gas using charts"
DELTAs[1]=ENTRDEP(Tr[1], Pr[1])*R "Entropy departure"
s[1]=s_ideal[1]-DELTAs[1]              "Entropy of real gas using charts"
"State 2"
T[2]=T[1]
Tr[2]=Tr[1]
pr[2]=p[2]/p_critical
Z[2]=COMPRESS(Tr[2], Pr[2])
DELTAh[2]=ENTHDEP(Tr[2], Pr[2])*R*T_critical "Enthalpy departure"
DELTAs[2]=ENTRDEP(Tr[2], Pr[2])*R "Entropy departure"
h[2]=h_ideal[2]-DELTAh[2]              "Enthalpy of real gas using charts"
s[2]=s_ideal[2]-DELTAs[2]              "Entropy of real gas using charts"
u[2]=h[2]-Z[2]*R*T[2]                  "Internal energy of gas using charts"

"Work using charts - note use of EES integral function to evaluate the integral of p dv."
w_chart=Integral(p,v,v[1],v[2])
"We need an equation to relate p and v in the above INTEGRAL function.  "
p*v=COMPRESS(Tr[2],p/p_critical)*R*T[1]  "To specify relationship between p and v"
"Find the limits of integration"
p[1]*v[1]=Z[1]*R*T[1]                  "to get v[1], the lower bound"
p[2]*v[2]=Z[2]*R*T[2]                  "to get v[2], the upper bound"

"First Law  - note that u[2] is not equal to u[1]"
q_chart-w_chart=u[2]-u[1]

"Entropy Change"
DELTAs_chart=s[2]-s[1]

"***** SOLUTION USING EES BUILT-IN PROPERTY DATA *****"
"At state 1"
u_ees[1]=intEnergy(Name$,T=T[1],p=p[1])
s_ees[1]=entropy(Name$,T=T[1],p=p[1])
"At state 2"
u_ees[2]=IntEnergy(Name$,T=T[2],p=p[2])
s_ees[2]=entropy(Name$,T=T[2],p=p[2])

"Work using EES built-in properties- note use of EES Integral funcion to evaluate the integral of pdv."
w_ees=integral(p_ees, v_ees, v_ees[1],v_ees[2])
"The following equation relates p and v in the above INTEGRAL"
p_ees=pressure(Name$,T=T[1], v=v_ees) "To specify relationship between p and v"

"Find the limits of integration"
v_ees[1]=volume(Name$, T=T[1],p=p[1]) "to get lower bound"
v_ees[2]=volume(Name$, T=T[2],p=p[2]) "to get upper bound"

"First law - note that u_ees[2] is not equal to u_ees[1]"
q_ees-w_ees=u_ees[2]-u_ees[1]
```

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"Entropy change"
DELTAs_ees=s_ees[2]-s_ees[1]

"Note: In all three solutions to this problem we could have calculated the heat transfer by q/T=DELTA_s since T is constant. Then the first law could have been used to find the work. The use of integral of p dv to find the work is a more fundamental approach and can be used if T is not constant."

SOLUTION

DELTAh[1]=16.48 [kJ/kg]
DELTAh[2]=91.96 [kJ/kg]
DELTAs[1]=0.03029 [kJ/kg-K]
DELTAs[2]=0.1851 [kJ/kg-K]
DELTAs_chart=-0.4162 [kJ/kg-K]
DELTAs_ees=-0.4711 [kJ/kg-K]
DELTAs_ideal=-0.2614 [kJ/kg-K]
Fluid$='C3H8'
h[1]=-2232 [kJ/kg]
h[2]=-2308 [kJ/kg]
h_ideal[1]=-2216 [kJ/kg]
h_ideal[2]=-2216 [kJ/kg]
M=44.1
Name$='Propane'
p=4000
p[1]=1000 [kPa]
p[2]=4000 [kPa]
pr[1]=0.2165
pr[2]=0.8658
p_critical=4620 [kPa]
p_ees=4000
q_chart=-155.3 [kJ/kg]
q_ees=-175.8 [kJ/kg]
q_ideal=-97.54 [kJ/kg]
R=0.1885 [kJ/kg-K]
R_u=8.314 [kJ/mole-K]
s[1]=6.073 [kJ/kg-K]
s[2]=5.657 [kJ/kg-K]
s_ees[1]=2.797 [kJ/kg-K]
s_ees[2]=2.326 [kJ/kg-K]
s_ideal[1]=6.103 [kJ/kg-K]
s_ideal[2]=5.842 [kJ/kg-K]
T[1]=373.2 [K]
T[2]=373.2 [K]
Tr[1]=1.009
Tr[2]=1.009
T_critical=370 [K]
u[1]=-2298 [kJ/kg]
u[2]=-2351 [kJ/kg]
u_ees[1]=688.4 [kJ/kg]
u_ees[2]=617.1 [kJ/kg]
u_ideal[1]=-2286 [kJ/kg]
u_ideal[2]=-2286 [kJ/kg]
v=0.01074
v[1]=0.06506 [m^3/kg]
v[2]=0.01074 [m^3/kg]
v_ees=0.009426
v_ees[1]=0.0646 [m^3/kg]
v_ees[2]=0.009426 [m^3/kg]
w_chart=-101.9 [kJ/kg]
w_ees=-104.5 [kJ/kg]
w_ideal=-97.54 [kJ/kg]
Z[1]=0.9246
Z[2]=0.6104

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12-81 Propane is compressed isothermally by a piston-cylinder device. The exergy destruction associated with this process is to be determined.

Assumptions **1**The compression process is quasi-equilibrium. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of propane is $R = 0.1885$ kJ/kg.K (Table A-1).

Analysis The exergy destruction is determined from its definition $x_{\text{destroyed}} = T_0 s_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the contents of the cylinder. It gives

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$

$$-\frac{Q_{\text{out}}}{T_{b,\text{surr}}} + S_{\text{gen}} = m(s_2 - s_1) \quad \rightarrow s_{\text{gen}} = (s_2 - s_1) + \frac{q_{\text{out}}}{T_{\text{surr}}}$$

where

$$\Delta s_{\text{sys}} = s_2 - s_1 = R(Z_{s1} - Z_{s2}) + (s_2 - s_1)_{\text{ideal}}$$

$$(s_2 - s_1)_{\text{ideal}} = c_p \ln \frac{T_2}{T_1}^{\nearrow 0} - R\ln \frac{P_2}{P_1} = 0 - 0.1885 \ln \frac{4}{1} = -0.261 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{373}{370} = 1.008 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{1}{4.26} = 0.235 \end{aligned}\right\} \longrightarrow Z_{s1} = 0.21$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{373}{370} = 1.008 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{4}{4.26} = 0.939 \end{aligned}\right\} \longrightarrow Z_{s2} = 1.5$$

Thus,

$$\Delta s_{\text{sys}} = s_2 - s_1 = R(Z_{s1} - Z_{s2}) + (s_2 - s_1)_{\text{ideal}} = (0.1885)(0.21 - 1.5) - 0.261 = -0.504 \text{ kJ/kg} \cdot \text{K}$$

and

$$\begin{aligned} x_{\text{destroyed}} &= T_0 s_{\text{gen}} = T_0 \left((s_2 - s_1) + \frac{q_{\text{out}}}{T_{\text{surr}}} \right) \\ &= (303 \text{ K}) \left(-0.504 + \frac{176.1 \text{ kJ/kg}}{303 \text{ K}} \right) \text{kJ/kg} \cdot \text{K} \\ &= \mathbf{23.4 \text{ kJ/kg}} \end{aligned}$$

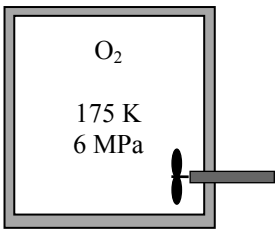
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12-82 A paddle-wheel placed in a well-insulated rigid tank containing oxygen is turned on. The final pressure in the tank and the paddle-wheel work done during this process are to be determined.

Assumptions **1**The tank is well-insulated and thus heat transfer is negligible. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of O_2 is $R = 0.2598$ kJ/kg.K (Table A-1).

Analysis (*a*) For this problem, we use critical properties, compressibility factor, and enthalpy departure factors in EES. The compressibility factor of oxygen at the initial state is determined from EES to be



$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{175}{154.6} = 1.13 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{6}{5.043} = 1.19 \end{aligned}\right\} \longrightarrow Z_1 = 0.682 \quad \text{and} \quad Z_{h1} = 1.33$$

Then,

$$P\nu = ZRT \longrightarrow \nu_1 = \frac{(0.682)(0.2598\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(175\ \text{K})}{6000\ \text{kPa}} = 0.00516\ \text{m}^3/\text{kg}$$

$$m = \frac{V}{\nu_1} = \frac{0.05\ \text{m}^3}{0.00516\ \text{m}^3/\text{kg}} = 9.68\ \text{kg}$$

The specific volume of oxygen remains constant during this process, $\nu_2 = \nu_1$. Thus,

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{cr}} = \frac{225}{154.8} = 1.46 \\ \nu_{R2} &= \frac{\nu_2}{RT_{cr}/P_{cr}} = \frac{0.00516\ \text{m}^3/\text{kg}}{(0.2598\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(154.6\ \text{K})/(5043\ \text{kPa})} = 0.649 \end{aligned}\right\} \begin{aligned} Z_2 &= 0.853 \\ Z_{h2} &= 1.09 \\ P_{R2} &= 1.91 \end{aligned}$$

$$P_2 = P_{R2}P_{cr} = (1.91)(5043) = \mathbf{9652\ kPa}$$

(*b*) The energy balance relation for this closed system can be expressed as

$$\begin{aligned} E_{in} - E_{out} &= \Delta E_{system} \\ W_{in} &= \Delta U = m(u_2 - u_1) \\ W_{in} &= m[h_2 - h_1 - (P_2\nu_2 - P_1\nu_1)] = m[h_2 - h_1 - R(Z_2T_2 - Z_1T_1)] \end{aligned}$$

where

$$\begin{aligned} h_2 - h_1 &= RT_{cr}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{ideal} \\ &= (0.2598)(154.6)(1.33 - 1.09) + 52.96 \\ &= 62.51\ \text{kJ/kg} \end{aligned}$$

Substituting,

$$W_{in} = (9.68\ \text{kg})[62.51 - (0.2598\ \text{kJ/kg}\cdot\text{K})\{(0.853)(225) - (0.682)(175)\}\text{K}] = \mathbf{423\ kJ}$$

Discussion The following routine in EES is used to get the solution above. Reading values from Fig. A-15 and A-29 together with properties in the book could yield different results.

```
"Given"
V=0.05 [m^3]
T1=175 [K]
P1=6000 [kPa]
T2=225 [K]

"Properties"
Fluid$='O2'
```

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```
R_u=8.314 [kJ/kmol-K]
T_cr=T_CRIT(Fluid$)
P_cr=P_CRIT(Fluid$)
MM=molarmass(Fluid$)
R=R_u/MM

"Analysis"
"(a)"
T_R1=T1/T_cr
P_R1=P1/P_cr
Z_h1=ENTHDEP(T_R1, P_R1) "the function that returns enthalpy departure factor at T_R1 and P_R1"
Z_1=COMPRESS(T_R1, P_R1) "the function that returns compressibility factor at T_R1 and P_R1"
T_R2=T2/T_cr
v_R2=(v2*P_cr)/(R*T_cr)
v1=Z_1*R*T1/P1
m=V/v1
v2=v1
Z_h2=ENTHDEP(T_R2, P_R2) "the function that returns enthalpy departure factor at T_R2 and P_R2"
Z_2=COMPRESS(T_R2, P_R2) "the function that returns compressibility factor at T_R2 and P_R2"
P2=Z_2*R*T2/v2
P2=P_R2*P_cr
"(b)"
h1_ideal=enthalpy(Fluid$, T=T1)
h2_ideal=enthalpy(Fluid$, T=T2)
DELTAh_ideal=(h2_ideal-h1_ideal)
DELTAh=R*T_cr*(Z_h1-Z_h2)+DELTAh_ideal
DELTAu=DELTAh-R*(Z_2*T2-Z_1*T1)
W_in=m*DELTAu
```

Solution

DELTAh=62.51 [kJ/kg]	T1=175 [K]
DELTAh_ideal=52.96 [kJ/kg]	T2=225 [K]
DELTAu=43.65 [kJ/kg]	T_cr=154.6 [K]
Fluid$='O2'	T_R1=1.132
h1_ideal=-121.8 [kJ/kg]	T_R2=1.456
h2_ideal=-68.8 [kJ/kg]	V=0.05 [m^3]
m=9.682 [kg]	v1=0.005164 [m^3/kg]
MM=32 [kg/kmol]	v2=0.005164 [m^3/kg]
P1=6000 [kPa]	v_R2=0.6485
P2=9652 [kPa]	**W_in=422.6 [kJ]**
P_cr=5043 [kPa]	Z_1=0.6815
P_R1=1.19	Z_2=0.8527
P_R2=1.914	Z_h1=1.331
R=0.2598 [kJ/kg-K]	Z_h2=1.094
R_u=8.314 [kJ/kmol-K]	

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12-83 The heat transfer and entropy changes of CO_2 during a process are to be determined assuming ideal gas behavior, using generalized charts, and real fluid (EES) data.

Analysis The temperature at the final state is

$$T_2 = T_1 \frac{P_2}{P_1} = (100 + 273\ \text{K}) \frac{8\ \text{MPa}}{1\ \text{MPa}} = 2984\ \text{K}$$

Using data from the ideal gas property table of CO_2 (Table A-20),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 161{,}293 - 12{,}{,}269 = 149{,}024\ \text{kJ/kmol}$$

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln \frac{P_2}{P_1} = 333.770 - 222.367 - 8.314 \times \ln \frac{8}{1} = 94.115\ \text{kJ/kmol} \cdot \text{K}$$

$$(h_2 - h_1)_{\text{ideal}} = \frac{(\bar{h}_2 - \bar{h}_1)_{\text{ideal}}}{M} = \frac{149{,}024\ \text{kJ/kmol}}{44\ \text{kg/kmol}} = 3386.9\ \text{kJ/kg}$$

The heat transfer is determined from an energy balance noting that there is no work interaction

$$q_{\text{ideal}} = (u_2 - u_1)_{\text{ideal}} = (h_2 - h_1)_{\text{ideal}} - R(T_2 - T_1) \quad ,,$$
$$= 3386.9\ \text{kJ/kg} - (0.1889\ \text{kJ/kg.K})(2984 - 373) = \mathbf{2893.7\ kJ/kg}$$

The entropy change is

$$\Delta s_{\text{ideal}} = (s_2 - s_1)_{\text{ideal}} = \frac{(\bar{s}_2 - \bar{s}_1)_{\text{ideal}}}{M} = \frac{94.115\ \text{kJ/kmol}}{44\ \text{kg/kmol}} = \mathbf{2.1390\ kJ/kg.K}$$

The compressibility factor and the enthalpy and entropy departures of CO_2 at the specified states are determined from the generalized charts to be (we used EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{373}{304.2} = 1.226 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{1}{7.39} = 0.135 \end{aligned}\right\} \longrightarrow Z_1 = 0.976,\ Z_{h1} = 0.1028 \text{ and } Z_{s1} = 0.05987$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{2985}{304.2} = 9.813 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{8}{7.39} = 1.083 \end{aligned}\right\} \longrightarrow Z_2 = 1.009,\ Z_{h2} = -0.1144 \text{ and } Z_{s2} = -0.002685$$

Thus,

$$q_{\text{chart}} = u_2 - u_1 = (h_2 - h_1)_{\text{ideal}} - RT_{\text{cr}}(Z_{h2} - Z_{h1}) - Z_1 R(T_2 - T_1)$$
$$= 3386.9 - (0.1889)(304.2)(-0.1144 - 0.1028) - (0.976)(0.1889)(2887 - 373) = \mathbf{2935.9\ kJ/kg}$$

$$\Delta s_{\text{chart}} = (s_2 - s_1)_{\text{chart}} = R(Z_{s1} - Z_{s2}) + (s_2 - s_1)_{\text{ideal}}$$
$$= (0.1889)(0.05987 - (-0.002685)) + 2.1390 = \mathbf{2.151 kJ/kg.K}$$

Note that the temperature at the final state in this case was determined from

$$T_2 = T_1 \frac{P_2}{P_1} \frac{Z_1}{Z_2} = (100 + 273\ \text{K}) \frac{8\ \text{MPa}}{1\ \text{MPa}} \frac{0.976}{1.009} = 2888\ \text{K}$$

The solution using EES built-in property data is as follows:

$$\left.\begin{aligned} T_1 &= 373\ \text{K} \\ P_1 &= 1\ \text{MPa} \end{aligned}\right\} \begin{aligned} \nu_1 &= 0.06885\ \text{m}^3/\text{kg} \\ u_1 &= -8.614\ \text{kJ/kg} \\ s_1 &= -0.2464\ \text{kJ/kg.K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 8\ \text{MPa} \\ \nu_2 &= \nu_1 = 0.06885\ \text{m}^3/\text{kg} \end{aligned}\right\} \begin{aligned} T_2 &= 2879\ \text{K} \\ u_2 &= 2754\ \text{kJ/kg} \\ s_2 &= 1.85\ \text{kJ/kg.K} \end{aligned}$$

Then

$$q_{\text{EES}} = u_2 - u_1 = 2754 - (-8.614) = \mathbf{2763\ kJ/kg}$$

$$\Delta s_{\text{EES}} = (s_2 - s_1)_{\text{EES}} = s_2 - s_1 = 1.85 - (-0.2464) = \mathbf{2.097\ kJ/kg.K}$$

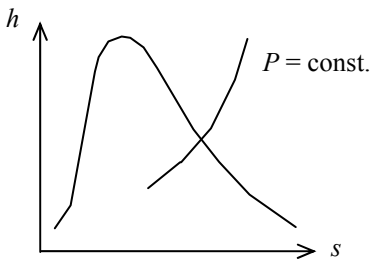
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Review Problems

12-84 It is to be shown that the slope of a constant-pressure line on an *h*-*s* diagram is constant in the saturation region and increases with temperature in the superheated region.

Analysis For P = constant, $dP = 0$ and the given relation reduces to $dh = Tds$, which can also be expressed as

$$\left(\frac{\partial h}{\partial s}\right)_P = T$$



Thus the slope of the P = constant lines on an *h*-*s* diagram is equal to the temperature.

(*a*) In the saturation region, T = constant for P = constant lines, and the slope remains constant.

(*b*) In the superheat region, the slope increases with increasing temperature since the slope is equal temperature.

12-85 Using the cyclic relation and the first Maxwell relation, the other three Maxwell relations are to be obtained.

Analysis (1) Using the properties P, s, v, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial s}\right)_v \left(\frac{\partial s}{\partial v}\right)_P \left(\frac{\partial v}{\partial P}\right)_s = -1$$

Substituting the first Maxwell relation, $\left(\frac{\partial T}{\partial v}\right)_s = -\left(\frac{\partial P}{\partial s}\right)_v$,

$$-\left(\frac{\partial T}{\partial v}\right)_s \left(\frac{\partial s}{\partial v}\right)_P \left(\frac{\partial v}{\partial P}\right)_s = -1 \longrightarrow \left(\frac{\partial T}{\partial P}\right)_s \left(\frac{\partial s}{\partial v}\right)_P = 1 \longrightarrow \left(\frac{\partial T}{\partial P}\right)_s = \left(\frac{\partial v}{\partial s}\right)_P$$

(2) Using the properties T, v, s, the cyclic relation can be expressed as

$$\left(\frac{\partial T}{\partial v}\right)_s \left(\frac{\partial v}{\partial s}\right)_T \left(\frac{\partial s}{\partial T}\right)_v = -1$$

Substituting the first Maxwell relation, $\left(\frac{\partial T}{\partial v}\right)_s = -\left(\frac{\partial P}{\partial s}\right)_v$,

$$-\left(\frac{\partial P}{\partial s}\right)_v \left(\frac{\partial v}{\partial s}\right)_T \left(\frac{\partial s}{\partial T}\right)_v = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial s}\right)_T = 1 \longrightarrow \left(\frac{\partial s}{\partial v}\right)_T = \left(\frac{\partial P}{\partial T}\right)_v$$

(3) Using the properties P, T, v, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial T}{\partial v}\right)_P \left(\frac{\partial v}{\partial P}\right)_T = -1$$

Substituting the third Maxwell relation, $\left(\frac{\partial s}{\partial v}\right)_T = \left(\frac{\partial P}{\partial T}\right)_v$,

$$\left(\frac{\partial s}{\partial v}\right)_T \left(\frac{\partial T}{\partial v}\right)_P \left(\frac{\partial v}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial s}{\partial P}\right)_T \left(\frac{\partial T}{\partial v}\right)_P = -1 \longrightarrow \left(\frac{\partial s}{\partial P}\right)_T = -\left(\frac{\partial v}{\partial T}\right)_P$$

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12-86 For $\beta \geq 0$, it is to be shown that at every point of a single-phase region of an *h*-*s* diagram, the slope of a constant-pressure line is greater than the slope of a constant-temperature line, but less than the slope of a constant-volume line.

Analysis It is given that $\beta > 0$.

Using the *Tds* relation: $dh = T\,ds + v\,dP \longrightarrow \dfrac{dh}{ds} = T + v\dfrac{dP}{ds}$

(1) P = constant: $\left(\dfrac{\partial h}{\partial s}\right)_P = T$

(2) T = constant: $\left(\dfrac{\partial h}{\partial s}\right)_T = T + v\left(\dfrac{\partial P}{\partial s}\right)_T$

But the 4th Maxwell relation: $\left(\dfrac{\partial P}{\partial s}\right)_T = -\left(\dfrac{\partial T}{\partial v}\right)_P$

Substituting: $\left(\dfrac{\partial h}{\partial s}\right)_T = T - v\left(\dfrac{\partial T}{\partial v}\right)_P = T - \dfrac{1}{\beta}$

Therefore, the slope of P = constant lines is **greater** than the slope of T = constant lines.

(3) v = constant: $\left(\dfrac{\partial h}{\partial s}\right)_v = T + v\left(\dfrac{\partial P}{\partial s}\right)_v \quad (a)$

From the ds relation: $ds = \dfrac{c_v}{T}\,dT + \left(\dfrac{\partial P}{\partial T}\right)_v dv$

Divide by dP holding v constant: $\left(\dfrac{\partial s}{\partial P}\right)_v = \dfrac{c_v}{T}\left(\dfrac{\partial T}{\partial P}\right)_v$ or $\left(\dfrac{\partial P}{\partial s}\right)_v = \dfrac{T}{c_v}\left(\dfrac{\partial P}{\partial T}\right)_v \quad (b)$

Using the properties P, T, v, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_v\left(\frac{\partial T}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v = -\left(\frac{\partial v}{\partial T}\right)_P\left(\frac{\partial P}{\partial v}\right)_T = (-\beta v)\left(\frac{1}{-\alpha v}\right) = \frac{\beta}{\alpha} \quad (c)$$

where we used the definitions of α and β. Substituting (*b*) and (*c*) into (*a*),

$$\left(\frac{\partial h}{\partial s}\right)_v = T + v\left(\frac{\partial P}{\partial s}\right)_v = T + \frac{T\beta v}{c_v \alpha} > T$$

Here α is positive for all phases of all substances. T is the absolute temperature that is also positive, so is c_v. Therefore, the second term on the right is always a positive quantity since β is given to be positive. Then we conclude that the slope of P = constant lines is **less** than the slope of v = constant lines.

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12-87 It is to be shown that

$$c_v = -T\left(\frac{\partial v}{\partial T}\right)_s \left(\frac{\partial P}{\partial T}\right)_v \quad \text{and} \quad c_p = T\left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial v}{\partial T}\right)_P$$

Analysis Using the definition of c_v,

$$c_v = T\left(\frac{\partial s}{\partial T}\right)_v = T\left(\frac{\partial s}{\partial P}\right)_v \left(\frac{\partial P}{\partial T}\right)_v$$

Substituting the first Maxwell relation $\left(\frac{\partial s}{\partial P}\right)_v = -\left(\frac{\partial v}{\partial T}\right)_s$,

$$c_v = -T\left(\frac{\partial v}{\partial T}\right)_s \left(\frac{\partial P}{\partial T}\right)_v$$

Using the definition of c_p,

$$c_p = T\left(\frac{\partial s}{\partial T}\right)_P = T\left(\frac{\partial s}{\partial v}\right)_P \left(\frac{\partial v}{\partial T}\right)_P$$

Substituting the second Maxwell relation $\left(\frac{\partial s}{\partial v}\right)_P = \left(\frac{\partial P}{\partial T}\right)_s$,

$$c_p = T\left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial v}{\partial T}\right)_P$$

12-88 It is to be proven that for a simple compressible substance $\left(\frac{\partial s}{\partial v}\right)_u = \frac{P}{T}$.

Analysis The proof is simply obtained as

$$\left(\frac{\partial s}{\partial v}\right)_u = \frac{-\left(\frac{\partial u}{\partial v}\right)_s}{\left(\frac{\partial u}{\partial s}\right)_v} = -\frac{-P}{T} = \frac{P}{T}$$

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12-89 It is to be proven by using the definitions of pressure and temperature, $T = \left(\frac{\partial u}{\partial s}\right)_{\nu}$ and $P = -\left(\frac{\partial u}{\partial \nu}\right)_{s}$ that for ideal gases, the development of the constant-pressure specific heat yields $\left(\frac{\partial h}{\partial P}\right)_{T} = 0$

Analysis The definition for enthalpy is

$$h = u + P\nu$$

Then,

$$\left(\frac{\partial h}{\partial P}\right)_{T} = \left(\frac{\partial u}{\partial P}\right)_{T} + P\left(\frac{\partial \nu}{\partial P}\right)_{T} + \nu\left(\frac{\partial P}{\partial P}\right)_{T}$$

Assume $u = u(s, \nu)$

Then,

$$du = \left(\frac{\partial u}{\partial s}\right)_{\nu} ds + \left(\frac{\partial u}{\partial \nu}\right)_{s} d\nu$$

$$\left(\frac{\partial u}{\partial P}\right)_{T} = \left(\frac{\partial u}{\partial s}\right)_{\nu}\left(\frac{\partial s}{\partial P}\right)_{T} + \left(\frac{\partial u}{\partial \nu}\right)_{s}\left(\frac{\partial \nu}{\partial P}\right)_{T}$$

$$\left(\frac{\partial u}{\partial P}\right)_{T} = T\left[-\left(\frac{\partial \nu}{\partial T}\right)_{P}\right] - P\left(\frac{\partial \nu}{\partial P}\right)_{T} = -(T + P)\left(\frac{\partial \nu}{\partial P}\right)_{T}$$

$$\left(\frac{\partial h}{\partial P}\right)_{T} = -(T + P)\left(\frac{\partial \nu}{\partial P}\right)_{T} + P\left(\frac{\partial \nu}{\partial P}\right)_{T} + \nu = -T\left(\frac{\partial \nu}{\partial P}\right)_{T} + \nu$$

For ideal gases

$$\nu = \frac{RT}{P} \quad \text{and} \quad \left(\frac{\partial \nu}{\partial P}\right)_{T} = \frac{R}{P}$$

Then,

$$\left(\frac{\partial h}{\partial P}\right)_{T} = -\frac{TR}{P} + \nu = -\nu + \nu = 0$$

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12-90 It is to be proven by using the definitions of pressure and temperature, $T = \left(\frac{\partial u}{\partial s}\right)_{\nu}$ and $P = -\left(\frac{\partial u}{\partial \nu}\right)_{s}$ that for ideal gases, the development of the constant-volume specific heat yields $\left(\frac{\partial u}{\partial \nu}\right)_{T} = 0$.

Analysis Assume $u = u(s, \nu)$

Then,

$$du = \left(\frac{\partial u}{\partial s}\right)_{\nu} ds + \left(\frac{\partial u}{\partial \nu}\right)_{s} d\nu$$

$$\left(\frac{\partial u}{\partial \nu}\right)_{T} = \left(\frac{\partial u}{\partial s}\right)_{\nu}\left(\frac{\partial s}{\partial \nu}\right)_{T} + \left(\frac{\partial u}{\partial \nu}\right)_{s}\left(\frac{\partial \nu}{\partial \nu}\right)_{T}$$

$$= T\left(\frac{\partial s}{\partial \nu}\right)_{T} + P$$

From Maxwell equation,

$$T\left(\frac{\partial s}{\partial \nu}\right)_{T} + P = T\left(\frac{\partial P}{\partial T}\right)_{\nu} - P$$

For ideal gases

$$P = \frac{RT}{\nu} \quad \text{and} \quad \left(\frac{\partial P}{\partial T}\right)_{\nu} = \frac{R}{\nu}$$

Then,

$$\left(\frac{\partial u}{\partial \nu}\right)_{T} = T\frac{R}{\nu} - P = P - P = 0$$

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12-91 Expressions for *h, u, s°, P_r,* and ν_r for an ideal gas whose c_p° is given by $c_p^{\circ} = \sum a_i \left[\ln\left(\frac{T}{T_r}\right) \right]^{i-n}$ are to be developed.

Analysis By making the change in variable, $x = \ln(T/T_r)$, the enthalpy of this substance relative to a reference state is given by

$$h = \int_{T_{\text{ref}}}^{T} c_p dT = \sum a_i \int_{x_{\text{ref}}}^{x} x^{i-n} e^x dx$$

$$= \sum a_i e^x \begin{bmatrix} x^{i-n} - (i-n)x^{i-n-1} + (i-n)(i-n-1)x^{i-n-2} - \ldots + (-1)^{i-n}(i-n)! \\ - x_{ref}^{i-n} + (i-n)x_{ref}^{i-n-1} - (i-n)(i-n-1)x_{ref}^{i-n-2} + \ldots - (-1)^{i-n}(i-n)! \end{bmatrix}$$

Similarly, s^0 is given by

$$s^o = \int_{T_{\text{ref}}}^{T} \left(\frac{c_p}{T}\right) dT = \sum a_i T_r \int_{x_{\text{ref}}}^{x} x^{i-n} e^{2x} dx$$

$$= \sum \frac{a_i e^{2x}}{2^{i-n}} \begin{bmatrix} (2x)^{i-n} - (i-n)(2x)^{i-n-1} + (i-n)(i-n-1)(2x)^{i-n-2} - \ldots + (-1)^{i-n}(i-n)! \\ - (2x_{ref})^{i-n} + (i-n)(2x_{ref})^{i-n-1} - (i-n)(i-n-1)(2x_{ref})^{i-n-2} + \ldots - (-1)^{i-n}(i-n)! \end{bmatrix}$$

With these two results,

$$u = h - P\nu$$

$$P_r = e^{s^{\circ}/R}$$

According to the *du* form of Gibbs equations,

$$\frac{du}{T} = -R\frac{d\nu}{\nu}$$

Noting that for ideal gases, $c_\nu = c_p - R$ and $du = c_\nu dT$, this expression reduces to

$$(c_p - R)\frac{dT}{T} = -R\frac{d\nu}{\nu}$$

When this is integrated between the reference and actual states, the result is

$$\int c_p \frac{dT}{T} - R\ln\frac{T}{T_{\text{ref}}} = -R\ln\frac{\nu}{\nu_{\text{ref}}}$$

Solving this for the specific volume ratio gives

$$\frac{\nu}{\nu_{\text{ref}}} = \exp\left[-\left(s^o - R(T - T_{ref})\right)R\right]$$

The ratio of the specific volumes at two states which have the same entropy is then

$$\frac{\nu}{\nu_{\text{ref}}} = \exp\left[-\left(s_2^{\circ} - s_1^{\circ} - R(T_2 - T_1)\right)R\right]$$

Inspection of this result gives

$$\nu_r = \exp\left[-(s^o - RT)/R\right]$$

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12-92 It is to be shown that the position of the Joule-Thompson coefficient inversion curve on the *T*-*P* plane is given by $(\partial Z/\partial T)_P = 0$.

Analysis The inversion curve is the locus of the points at which the Joule-Thompson coefficient μ is zero,

$$\mu = \frac{1}{c_p}\left(T\left(\frac{\partial \nu}{\partial T}\right)_P - \nu\right) = 0$$

which can also be written as

$$T\left(\frac{\partial \nu}{\partial T}\right)_P - \frac{ZRT}{P} = 0 \qquad (a)$$

since it is given that

$$\nu = \frac{ZRT}{P} \qquad (b)$$

Taking the derivative of (*b*) with respect to *T* holding *P* constant gives

$$\left(\frac{\partial \nu}{\partial T}\right)_P = \left(\frac{\partial (ZRT/P)}{\partial T}\right)_P = \frac{R}{P}\left(T\left(\frac{\partial Z}{\partial T}\right)_P + Z\right)$$

Substituting in (*a*),

$$\frac{TR}{P}\left(T\left(\frac{\partial Z}{\partial T}\right)_P + Z\right) - \frac{ZRT}{P} = 0$$

$$T\left(\frac{\partial Z}{\partial T}\right)_P + Z - Z = 0$$

$$\left(\frac{\partial Z}{\partial T}\right)_P = 0$$

which is the desired relation.

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12-93 It is to be shown that for an isentropic expansion or compression process $P\nu^k$ = constant. It is also to be shown that the isentropic expansion exponent k reduces to the specific heat ratio c_p/c_ν for an ideal gas.

Analysis We note that $ds = 0$ for an isentropic process. Taking $s = s(P, \nu)$, the total differential ds can be expressed as

$$ds = \left(\frac{\partial s}{\partial P}\right)_\nu dP + \left(\frac{\partial s}{\partial \nu}\right)_P d\nu = 0 \qquad (a)$$

We now substitute the Maxwell relations below into (*a*)

$$\left(\frac{\partial s}{\partial P}\right)_\nu = -\left(\frac{\partial \nu}{\partial T}\right)_s \text{ and } \left(\frac{\partial s}{\partial \nu}\right)_P = \left(\frac{\partial P}{\partial T}\right)_s$$

to get

$$-\left(\frac{\partial \nu}{\partial T}\right)_s dP + \left(\frac{\partial P}{\partial T}\right)_s d\nu = 0$$

Rearranging,

$$dP - \left(\frac{\partial T}{\partial \nu}\right)_s \left(\frac{\partial P}{\partial T}\right)_s d\nu = 0 \longrightarrow dP - \left(\frac{\partial P}{\partial \nu}\right)_s d\nu = 0$$

Dividing by P, $$\frac{dP}{P} - \frac{1}{P}\left(\frac{\partial P}{\partial \nu}\right)_s d\nu = 0 \qquad (b)$$

We now define isentropic expansion exponent k as

$$k = -\frac{\nu}{P}\left(\frac{\partial P}{\partial \nu}\right)_s$$

Substituting in (*b*),

$$\frac{dP}{P} + k\frac{d\nu}{\nu} = 0$$

Taking k to be a constant and integrating,

$$\ln P + k \ln \nu = \text{constant} \longrightarrow \ln P\nu^k = \text{constant}$$

Thus,

$$P\nu^k = \text{constant}$$

To show that $k = c_p/c_\nu$ for an ideal gas, we write the cyclic relations for the following two groups of variables:

$$(s, T, \nu) \longrightarrow \left(\frac{\partial s}{\partial T}\right)_\nu \left(\frac{\partial \nu}{\partial s}\right)_T \left(\frac{\partial T}{\partial \nu}\right)_s = -1 \longrightarrow \frac{c_\nu}{T}\left(\frac{\partial \nu}{\partial s}\right)_T \left(\frac{\partial T}{\partial \nu}\right)_s = -1 \quad (c)$$

$$(s, T, P) \longrightarrow \left(\frac{\partial s}{\partial T}\right)_P \left(\frac{\partial P}{\partial s}\right)_T \left(\frac{\partial T}{\partial P}\right)_s = -1 \longrightarrow \frac{c_p}{T}\left(\frac{\partial P}{\partial s}\right)_T \left(\frac{\partial T}{\partial P}\right)_s = -1 \quad (d)$$

where we used the relations

$$c_\nu = T\left(\frac{\partial s}{\partial T}\right)_\nu \text{ and } c_p = T\left(\frac{\partial s}{\partial T}\right)_P$$

Setting Eqs. (*c*) and (*d*) equal to each other,

$$\frac{c_p}{T}\left(\frac{\partial P}{\partial s}\right)_T \left(\frac{\partial T}{\partial P}\right)_s = \frac{c_\nu}{T}\left(\frac{\partial \nu}{\partial s}\right)_T \left(\frac{\partial T}{\partial \nu}\right)_s$$

or,

$$\frac{c_p}{c_\nu} = \left(\frac{\partial s}{\partial P}\right)_T \left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial \nu}{\partial s}\right)_T \left(\frac{\partial T}{\partial \nu}\right)_s = \left(\frac{\partial s}{\partial P}\frac{\partial \nu}{\partial s}\right)_T \left(\frac{\partial P}{\partial T}\frac{\partial T}{\partial \nu}\right)_s = \left(\frac{\partial \nu}{\partial P}\right)_T \left(\frac{\partial P}{\partial \nu}\right)_s$$

but

$$\left(\frac{\partial \nu}{\partial P}\right)_T = \left(\frac{\partial (RT/P)}{\partial P}\right)_T = -\frac{\nu}{P}$$

Substituting,

$$\frac{c_p}{c_\nu} = -\frac{\nu}{P}\left(\frac{\partial P}{\partial \nu}\right)_s = k$$

which is the desired relation.

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12-94 The c_p of nitrogen at 300 kPa and 400 K is to be estimated using the relation given and its definition, and the results are to be compared to the value listed in Table A-2*b*.

Analysis (*a*) We treat nitrogen as an ideal gas with $R = 0.297$ kJ/kg·K and $k = 1.397$. Note that $PT^{k/(k-1)} = C$ = constant for the isentropic processes of ideal gases. The c_p relation is given as

$$c_p = T\left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial \nu}{\partial T}\right)_P$$

$$\nu = \frac{RT}{P} \longrightarrow \left(\frac{\partial \nu}{\partial T}\right)_P = \frac{R}{P}$$

$$P = CT^{k/(k-1)} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_s = \frac{k}{k-1} CT^{k/(k-1)-1} = \frac{k}{k-1}\left(PT^{-k/(k-1)}\right)T^{k/(k-1)-1} = \frac{kP}{T(k-1)}$$

Substituting,

$$c_p = T\left(\frac{kP}{T(k-1)}\right)\left(\frac{R}{P}\right) = \frac{kR}{k-1} = \frac{1.397(0.297 \text{ kJ/kg}\cdot\text{K})}{1.397-1} = 1.045 \text{ kJ/kg}\cdot\text{K}$$

(*b*) The c_p is defined as $c_p = \left(\frac{\partial h}{\partial T}\right)_P$. Replacing the differentials by differences,

$$c_p \cong \left(\frac{\Delta h}{\Delta T}\right)_{P=300\text{kPa}} = \frac{h(410\ K) - h(390\ \text{K})}{(410-390)\text{K}} = \frac{(11{,}932 - 11{,}347)/28.0 \text{ kJ/kg}}{(410-390)\text{K}} = \mathbf{1.045\ kJ/kg\cdot K}$$

(Compare: Table A-2*b* at 400 K $\rightarrow$ $c_p = 1.044$ kJ/kg·K)

12-95 The temperature change of steam and the average Joule-Thompson coefficient during a throttling process are to be estimated.

Analysis The enthalpy of steam at 4.5 MPa and $T = 300$°C is $h = 2944.2$ kJ/kg. Now consider a throttling process from this state to 2.5 MPa. The temperature of the steam at the end of this throttling process will be

$$\left.\begin{array}{l} P = 2.5 \text{ MPa} \\ h = 2944.2 \text{ kJ/kg} \end{array}\right\} T_2 = 273.72°\text{C}$$

Thus the temperature drop during this throttling process is

$$\Delta T = T_2 - T_1 = 273.72 - 300 = \mathbf{-26.28°C}$$

The average Joule-Thomson coefficient for this process is determined from

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=3204.7\text{kJ/kg}} = \frac{(273.72-300)°\text{C}}{(2.5-4.5)\text{MPa}} = \mathbf{13.14°C/MPa}$$

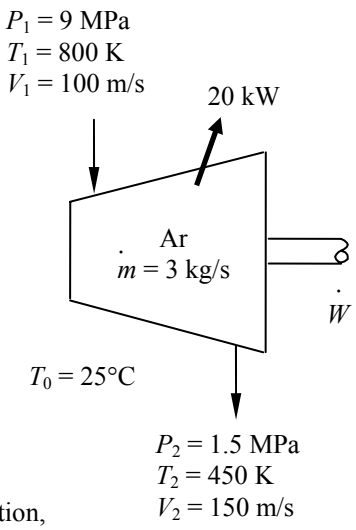
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12-96 Argon enters a turbine at a specified state and leaves at another specified state. Power output of the turbine and exergy destruction during this process are to be determined using the generalized charts.

Properties The gas constant and critical properties of Argon are $R = 0.2081$ kJ/kg.K, $T_{cr} = 151$ K, and $P_{cr} = 4.86$ MPa (Table A-1).

Analysis (*a*) The enthalpy and entropy departures of argon at the specified states are determined from the generalized charts to be

$$\left.\begin{aligned} T_{R_1} &= \frac{T_1}{T_{cr}} = \frac{800}{151} = 5.30 \\ P_{R_1} &= \frac{P_1}{P_{cr}} = \frac{9}{4.86} = 1.85 \end{aligned}\right\} Z_{h_1} \cong 0 \text{ and } Z_{s_1} \cong 0$$



Thus argon behaves as an ideal gas at turbine inlet. Also,

$$\left.\begin{aligned} T_{R_2} &= \frac{T_2}{T_{cr}} = \frac{450}{151} = 2.98 \\ P_{R_2} &= \frac{P_2}{P_{cr}} = \frac{1.5}{4.86} = 0.309 \end{aligned}\right\} Z_{h_2} = 0.024 \text{ and } Z_{s_2} = 0.013$$

Thus,

$$\begin{aligned} h_2 - h_1 &= RT_{cr}\left(Z_{h_1} - Z_{h_2}\right) + \left(h_2 - h_1\right)_{\text{ideal}} \\ &= (0.2081)(151)(0 - 0.024) + 0.5203(450 - 800) = -182.9 \text{ kJ/kg} \end{aligned}$$

The power output of the turbine is to be determined from the energy balance equation,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} = 0 \text{ (steady)} \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) + \dot{Q}_{\text{out}} + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = -\dot{m}\left[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}\right] - \dot{Q}_{\text{out}}$$

Substituting,

$$\dot{W}_{\text{out}} = -(3 \text{ kg/s})\left(-182.9 + \frac{(150 \text{ m/s})^2 - (100 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right) - 20 \text{ kJ/s} = \mathbf{510\ kW}$$

(*b*) Under steady conditions, the rate form of the entropy balance for the turbine simplifies to

$$\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = \Delta \dot{S}_{\text{system}}^{\nearrow 0} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{b,\text{out}}} + \dot{S}_{\text{gen}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_2) + \frac{\dot{Q}_{\text{out}}}{T_0}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0\left(\dot{m}(s_2 - s_2) + \frac{\dot{Q}_{\text{out}}}{T_0}\right)$$

where $s_2 - s_1 = R\left(Z_{s_1} - Z_{s_2}\right) + \left(s_2 - s_1\right)_{\text{ideal}}$

and $\left(s_2 - s_1\right)_{\text{ideal}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = 0.5203 \ln\frac{450}{800} - 0.2081 \ln\frac{1.5}{9} = 0.0735 \text{ kJ/kg} \cdot \text{K}$

Thus, $s_2 - s_1 = R\left(Z_{s_1} - Z_{s_2}\right) + \left(s_2 - s_1\right)_{\text{ideal}} = (0.2081)[0 - (0.013)] + 0.0735 = 0.0708 \text{ kJ/kg} \cdot \text{K}$

Substituting,

$$\dot{X}_{\text{destroyed}} = (298 \text{ K})\left((3 \text{ kg/s})(0.0708 \text{ kJ/kg} \cdot \text{K}) + \frac{20 \text{ kW}}{298 \text{ K}}\right) = \mathbf{83.3\ kW}$$

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12-97 Problem 12-96 is reconsidered. The problem is to be solved assuming steam is the working fluid by using the generalized chart method and EES data for steam. The power output and the exergy destruction rate for these two calculation methods against the turbine exit pressure are to be plotted.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
" Input Data "
T[1]=800 [K]
P[1]=9000 [kPa]
Vel[1]=100 [m/s]
T[2]=450 [K]
P[2]=1500 [kPa]
Vel[2]=150 [m/s]
Q_dot_out=20 [kW]
T_o=25+273 "[K]"
m_dot=3 [kg/s]
Name$='Steam_iapws'
 T_critical=647.3 [K]
P_critical=22090 [kPa]
Fluid$='H2O'

R_u=8.314
M=molarmass(Fluid$)
R=R_u/M

"****** IDEAL GAS SOLUTION ******"
"State 1"
h_ideal[1]=enthalpy(Fluid$,T=T[1])    "Enthalpy of ideal gas"
s_ideal[1]=entropy(Fluid$, T=T[1], P=P[1]) "Entropy of ideal gas"
"State 2"
h_ideal[2]=enthalpy(Fluid$,T=T[2])    "Enthalpy of ideal gas"
s_ideal[2]=entropy(Fluid$, T=T[2], P=P[2]) "Entropy of ideal gas"

"Conservation of Energy, Steady-flow: "
"E_dot_in=E_dot_out"

m_dot*(h_ideal[1]+Vel[1]^2/2*convert(m^2/s^2,kJ/kg))=m_dot*(h_ideal[2]+Vel[2]^2/2*convert(m^2/s^2,kJ/kg))+Q
_dot_out+W_dot_out_ideal

"Second Law analysis:"
"S_dot_in-S_dot_out+S_dot_gen = 0"
m_dot*s_ideal[1] - m_dot*s_ideal[2] - Q_dot_out/T_o + S_dot_gen_ideal = 0

"Exergy Destroyed:"
X_dot_destroyed_ideal = T_o*S_dot_gen_ideal

"***** COMPRESSABILITY CHART  SOLUTION ******"
"State 1"
Tr[1]=T[1]/T_critical
Pr[1]=P[1]/P_critical
Z[1]=COMPRESS(Tr[1], Pr[1])
DELTAh[1]=ENTHDEP(Tr[1], Pr[1])*R*T_critical "Enthalpy departure"
h_chart[1]=h_ideal[1]-DELTAh[1]      "Enthalpy of real gas using charts"
DELTAs[1]=ENTRDEP(Tr[1], Pr[1])*R "Entropy departure"
s_chart[1]=s_ideal[1]-DELTAs[1]      "Entropy of real gas using charts"
"State 2"
Tr[2]=T[2]/T_critical
Pr[2]=P[2]/P_critical
Z[2]=COMPRESS(Tr[2], Pr[2])
```

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```
DELTAh[2]=ENTHDEP(Tr[2], Pr[2])*R*T_critical "Enthalpy departure"
DELTAs[2]=ENTRDEP(Tr[2], Pr[2])*R "Entropy departure"
h_chart[2]=h_ideal[2]-DELTAh[2]      "Enthalpy of real gas using charts"
s_chart[2]=s_ideal[2]-DELTAs[2]      "Entropy of real gas using charts"

"Conservation of Energy, Steady-flow: "
"E_dot_in=E_dot_out"

m_dot*(h_chart[1]+Vel[1]^2/2*convert(m^2/s^2,kJ/kg))=m_dot*(h_chart[2]+Vel[2]^2/2*convert(m^2/s^2,kJ/kg))+Q
_dot_out+W_dot_out_chart

"Second Law analysis:"
"S_dot_in-S_dot_out+S_dot_gen = 0"
m_dot*s_chart[1] - m_dot*s_chart[2] - Q_dot_out/T_o + S_dot_gen_chart = 0

"Exergy Destroyed:"

X_dot_destroyed_chart = T_o*S_dot_gen_chart"[kW]"

"***** SOLUTION USING EES BUILT-IN PROPERTY DATA *****"
"At state 1"
h_ees[1]=enthalpy(Name$,T=T[1],P=P[1])
s_ees[1]=entropy(Name$,T=T[1],P=P[1])
"At state 2"
h_ees[2]=enthalpy(Name$,T=T[2],P=P[2])
s_ees[2]=entropy(Name$,T=T[2],P=P[2])

"Conservation of Energy, Steady-flow: "
"E_dot_in=E_dot_out"

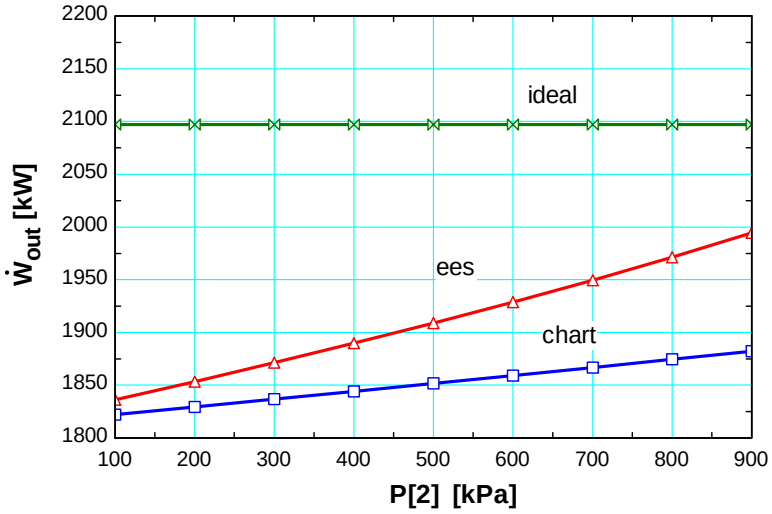
m_dot*(h_ees[1]+Vel[1]^2/2*convert(m^2/s^2,kJ/kg))=m_dot*(h_ees[2]+Vel[2]^2/2*convert(m^2/s^2,kJ/kg))+Q_d
ot_out+W_dot_out_ees

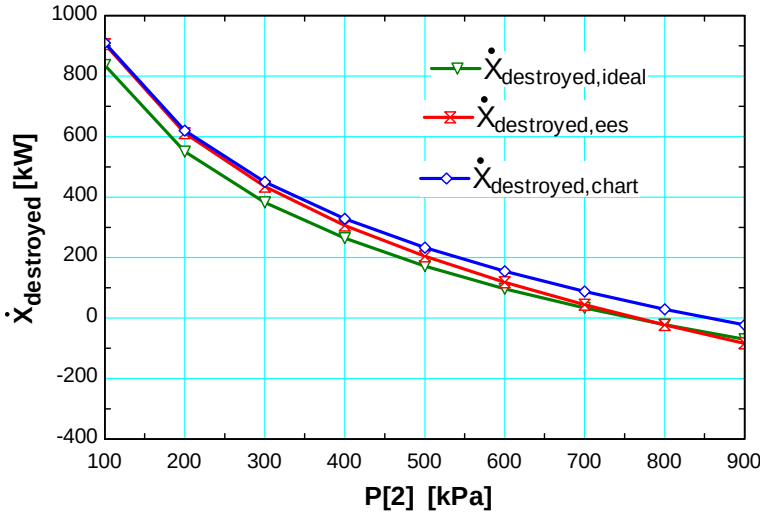
"Second Law analysis:"
"S_dot_in-S_dot_out+S_dot_gen = 0"
m_dot*s_ees[1] - m_dot*s_ees[2] - Q_dot_out/T_o + S_dot_gen_ees= 0

"Exergy Destroyed:"
X_dot_destroyed_ees = T_o*S_dot_gen_ees
```

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P_2 [kPa]	T_2 [K]	$W_{outchart}$ [kW]	W_{outees} [kW]	$W_{outideal}$ [kW]	$X_{destroyedchart}$ [kW]	$X_{destroyedees}$ [kW]	$X_{destroyedideal}$ [kW]
1100	450	1822	1836	2097	909.1	905.7	836
200	450	1829	1853	2097	620	610.9	550.1
300	450	1837	1871	2097	449.7	434.4	382.8
400	450	1844	1890	2097	327.9	306.2	264.1
500	450	1852	1909	2097	232.7	204.2	172
600	450	1859	1929	2097	154.3	118.6	96.79
700	450	1867	1950	2097	87.51	44.06	33.19
800	450	1874	1971	2097	29.18	-22.56	-21.9
900	450	1882	1994	2097	-22.7	-83.47	-70.5





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12-98 An adiabatic storage tank that is initially evacuated is connected to a supply line that carries nitrogen. A valve is opened, and nitrogen flows into the tank. The final temperature in the tank is to be determined by treating nitrogen as an ideal gas and using the generalized charts, and the results are to be compared to the given actual value.

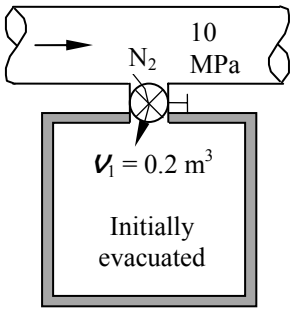
Assumptions **1** Uniform flow conditions exist. **2** Kinetic and potential energies are negligible.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$

Energy balance: $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \quad \rightarrow \quad 0 + m_i h_i = m_2 u_2$

Combining the two balances: $u_2 = h_{\text{i}}$



(*a*) From the ideal gas property table of nitrogen, at 225 K we read

$$\overline{u}_2 = \overline{h}_i = \overline{h}_{@\,225\text{ K}} = 6{,}537 \text{ kJ/kmol}$$

The temperature that corresponds to this $\overline{u}_2$ value is

$$T_2 = \mathbf{314.8\ K} \qquad (7.4\% \text{ error})$$

(*b*) Using the generalized enthalpy departure chart, h_{i} is determined to be

$$\left.\begin{aligned} T_{R,i} &= \frac{T_i}{T_{\text{cr}}} = \frac{225}{126.2} = 1.78 \\ P_{R,i} &= \frac{P_i}{P_{\text{cr}}} = \frac{10}{3.39} = 2.95 \end{aligned}\right\} Z_{h,i} = \frac{\overline{h}_{i,\text{ideal}} - \overline{h}_i}{R_u T_{cr}} = 0.9 \qquad \text{(Fig. A-29)}$$

Thus,

$$\overline{h}_i = \overline{h}_{i,\text{ideal}} - 0.9 R_u T_{\text{cr}} = 6{,}537 - (0.9)(8.314)(126.2) = 5{,}593 \text{ kJ/kmol}$$

and

$$\overline{u}_2 = \overline{h}_i = 5{,}593 \text{ kJ/kmol}$$

Try $T_2 = 280$ K. Then at $P_{R2} = 2.95$ and $T_{R2} = 2.22$ we read $Z_2 = 0.98$ and $(\overline{h}_{2,ideal} - \overline{h}_2) / R_u T_{\text{cr}} = 0.55$

Thus,

$$\overline{h}_2 = \overline{h}_{2,\text{ideal}} - 0.55 R_u T_{\text{cr}} = 8{,}141 - (0.55)(8.314)(126.2) = 7{,}564 \text{ kJ/kmol}$$
$$\overline{u}_2 = \overline{h}_2 - Z R_u T_2 = 7{,}564 - (0.98)(8.314)(280) = 5{,}283 \text{ kJ/kmol}$$

Try $T_2 = 300$ K. Then at $P_{R2} = 2.95$ and $T_{R2} = 2.38$ we read $Z_2 = 1.0$ and $(\overline{h}_{2,ideal} - \overline{h}_2) / R_u T_{\text{cr}} = 0.50$

Thus,

$$\overline{h}_2 = \overline{h}_{2,\text{ideal}} - 0.50 R_u T_{\text{cr}} = 8{,}723 - (0.50)(8.314)(126.2) = 8{,}198 \text{ kJ/kmol}$$
$$\overline{u}_2 = \overline{h}_2 - Z R_u T_2 = 8{,}198 - (1.0)(8.314)(300) = 5{,}704 \text{ kJ/kmol}$$

By linear interpolation,

$$T_2 = \mathbf{294.7\ K} \qquad (0.6\% \text{ error})$$

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12-99E Methane is to be adiabatically and reversibly compressed in a steady-flow device. The specific work required for this compression is to be determined using the departure charts and treating the methane as an ideal gas with temperature variable specific heats.

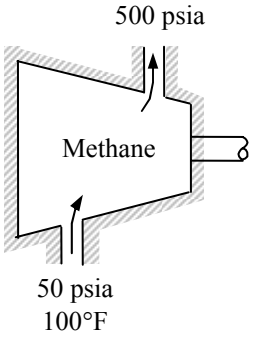
Properties The properties of methane are (Table A-1E)

$$M = 16.043 \text{ lbm/lbmol},\ R = 0.1238 \text{ Btu/lbm} \cdot \text{R},\ T_{\text{cr}} = 343.9 \text{ R},\ P_{\text{cr}} = 673 \text{ psia}$$

Analysis The temperature at the exit state may be determined by the fact that the process is isentropic and the molar entropy change between the inlet and exit is zero. When the expression of Table A-2Ec is substituted for c_p and the integration performed, we obtain

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = \int_1^2 \frac{c_p}{T} dT - R_u \ln \frac{P_2}{P_1} = \int_1^2 \left(\frac{a}{T} + b + cT + dT^2 \right) dT - R_u \ln \frac{P_2}{P_1}$$

$$= a \ln \frac{T_2}{T_1} + b(T_2 - T_1) + \frac{c}{2}(T_2^2 - T_1^2) + \frac{d}{3}(T_2^3 - T_1^3) - R_u \ln \frac{P_2}{P_1}$$



Substituting,

$$0 = 4.75 \ln \frac{T_2}{560} + 0.006666(T_2 - 560) + \frac{0.09352 \times 10^{-5}}{2}(T_2^2 - 560^2)$$

$$- \frac{0.4510 \times 10^{-9}}{3}(T_2^3 - 560^3) - (1.9858) \ln \frac{500}{50}$$

Solving this equation by EES or an iterative solution gives

$$T_2 = 892 \text{ R}$$

When en energy balance is applied to the compressor, it becomes

$$\overline{w}_{\text{in}} = (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \int_1^2 c_p dT = \int_1^2 (a + bT + cT^2 + dT^3) dT$$

$$= a(T_2 - T_1) + \frac{b}{2}(T_2^2 - T_1^2) + \frac{c}{3}(T_2^3 - T_1^3) + \frac{d}{4}(T_2^4 - T_1^4)$$

$$= 4.75(892 - 560) + \frac{0.006666}{2}(892^2 - 560^2) + \frac{0.09352 \times 10^{-5}}{3}(892^3 - 560^3)$$

$$- \frac{0.4510 \times 10^{-9}}{4}(892^4 - 560^4)$$

$$= 3290 \text{ Btu/lbmol}$$

The work input per unit mass basis is

$$w_{\text{in}} = \frac{\overline{w}_{\text{in}}}{M} = \frac{3290 \text{ Btu/lbmol}}{16.043 \text{ lbm/lbmol}} = \mathbf{205.1 \text{ Btu/lbm}}$$

The enthalpy departures of propane at the specified states are determined from the generalized charts to be (Fig. A-29 or from EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{560}{343.9} = 1.628 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{50}{673} = 0.0743 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.0332$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{892}{343.9} = 2.594 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{500}{673} = 0.743 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.0990$$

The work input is determined to be

$$w_{\text{in}} = h_2 - h_1 = (h_2 - h_1)_{\text{ideal}} - RT_{\text{cr}}(Z_{h2} - Z_{h1})$$

$$= 205.1 \text{ Btu/lbm} - (0.1238 \text{ Btu/lbm} \cdot \text{R})(343.9 \text{ R})(0.0990 - 0.0332)$$

$$= \mathbf{202.3 \text{ Btu/lbm}}$$

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12-100 The volume expansivity of water is given. The change in volume of water when it is heated at constant pressure is to be determined.

Properties The volume expansivity of water is given to be 0.207×10^{-6} K^{-1} at 20°C.

Analysis We take $v = v(P, T)$. Its total differential is

$$dv = \left(\frac{\partial v}{\partial T}\right)_P dT + \left(\frac{\partial v}{\partial P}\right)_T dP$$

which, for a constant pressure process, reduces to

$$dv = \left(\frac{\partial v}{\partial T}\right)_P dT$$

Dividing by v and using the definition of β,

$$\frac{dv}{v} = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_P dT = \beta dT$$

Taking β to be a constant, integration from 1 to 2 yields

$$\ln\frac{v_2}{v_1} = \beta(T_2 - T_1)$$

or

$$\frac{v_2}{v_1} = \exp[\beta(T_2 - T_1)]$$

Substituting the given values and noting that for a fixed mass $V_2/V_1 = v_2/v_1$,

$$V_2 = V_1 \exp[\beta(T_2 - T_1)] = (0.5\ \text{m}^3)\exp[(0.207\times10^{-6}\ \text{K}^{-1})(50-10)°\text{C}]$$
$$= 0.50000414\ \text{m}^3$$

Therefore,

$$\Delta V = V_2 - V_1 = 0.50000414 - 0.5 = 0.00000414\ \text{m}^3 = \mathbf{4.14\ cm^3}$$

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12-101 The work done by the refrigerant 134a as it undergoes an isothermal process in a closed system is to be determined using the tabular (EES) data and the generalized charts.

Analysis The solution using EES built-in property data is as follows:

$$\left.\begin{matrix} T_1 = 40°\text{C} \\ P_1 = 2\ \text{MPa} \end{matrix}\right\} \begin{matrix} u_1 = 106.37\ \text{kJ/kg} \\ s_1 = 0.3916\ \text{kJ/kg.K} \end{matrix}$$

$$\left.\begin{matrix} T_2 = 40°\text{C} \\ P_2 = 0.1\ \text{MPa} \end{matrix}\right\} \begin{matrix} u_2 = 264.25\ \text{kJ/kg} \\ s_2 = 1.1484\ \text{kJ/kg.K} \end{matrix}$$

$$\Delta s_{\text{EES}} = s_2 - s_1 = 1.1484 - 0.3916 = 0.7568\ \text{kJ/kg.K}$$

$$q_{\text{EES}} = T_1 \Delta s_{\text{EES}} = (40 + 273.15\ \text{K})(0.7568\ \text{kJ/kg.K}) = 237.00\ \text{kJ/kg}$$

$$w_{\text{EES}} = q_{\text{EES}} - (u_2 - u_1) = 237.00 - (264.25 - 106.37) = \mathbf{79.1\ kJ/kg}$$

For the generalized chart solution we first determine the following factors using EES as

$$\left.\begin{matrix} T_{R1} = \dfrac{T_1}{T_{cr}} = \dfrac{313.2}{374.2} = 0.8369 \\ P_{R1} = \dfrac{P_1}{P_{cr}} = \dfrac{2}{4.059} = 0.4927 \end{matrix}\right\} \longrightarrow Z_1 = 0.08357,\ Z_{h1} = 4.82 \ \text{ and } \ Z_{s1} = 5.147$$

$$\left.\begin{matrix} T_{R2} = \dfrac{T_2}{T_{cr}} = \dfrac{313.2}{374.2} = 0.8369 \\ P_{R2} = \dfrac{P_2}{P_{cr}} = \dfrac{0.1}{4.059} = 0.02464 \end{matrix}\right\} \longrightarrow Z_2 = 0.9857,\ Z_{h2} = 0.03396 \ \text{ and } \ Z_{s2} = 0.02635$$

Then,

$$\Delta h_1 = Z_{h1} R T_{\text{cr}} = (4.82)(0.08148\ \text{kJ/kg.K})(374.2\ \text{K}) = 146.97\ \text{kJ/kg}$$
$$\Delta s_1 = Z_{s1} R = (5.147)(0.08148\ \text{kJ/kg.K}) = 0.4194\ \text{kJ/kg.K}$$

$$\Delta h_2 = Z_{h2} R T_{\text{cr}} = (0.03396)(0.08148\ \text{kJ/kg.K})(374.2\ \text{K}) = 1.04\ \text{kJ/kg}$$
$$\Delta s_2 = Z_{s2} R = (0.02635)(0.08148\ \text{kJ/kg.K}) = 0.002147\ \text{kJ/kg.K}$$

$$\Delta s_{\text{ideal}} = R \ln \frac{P_2}{P_1} = (0.08148\ \text{kJ/kg.K}) \ln\left(\frac{0.1}{2}\right) = 0.2441\ \text{kJ/kg} \cdot \text{K}$$

$$\Delta s_{\text{chart}} = \Delta s_{\text{ideal}} - (\Delta s_2 - \Delta s_1) = 0.2441 - (0.002147 - 0.4194) = 0.6613\ \text{kJ/kg} \cdot \text{K}$$

$$q_{\text{chart}} = T_1 \Delta s_{\text{chart}} = (40 + 273.15\ \text{K})(0.6613\ \text{kJ/kg.K}) = 207.09\ \text{kJ/kg}$$

$$\Delta u_{\text{chart}} = \Delta h_{\text{ideal}} - (\Delta h_2 - \Delta h_1) - (Z_2 R T_2 - Z_1 R T_1)$$
$$= 0 - (1.04 - 146.97) - \left[(0.9857)(0.08148)(313.2) - (0.08357)(0.08148)(313.2)\right] = 122.92\ \text{kJ/kg}$$

$$w_{\text{chart}} = q_{\text{chart}} - \Delta u_{\text{chart}} = 207.09 - 122.92 = \mathbf{84.2\ kJ/kg}$$

The copy of the EES solution of this problem is given next.

```
"Input data"
T_critical=T_CRIT(R134a) "[K]"
P_critical=P_CRIT(R134a) "[kpa]"
T[1]=40+273.15"[K]"
T[2]=T[1]"[K]"
P[1]=2000"[kPa]"
```

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```
P[2]=100"[kPa]"
R_u=8.314"[kJ/kmol-K]"
M=molarmass(R134a)
R=R_u/M"[kJ/kg-K]"

"***** SOLUTION USING EES BUILT-IN PROPERTY DATA *****"

"For the isothermal process, the heat transfer is T*(s[2] - s[1]):"
DELTAs_EES=(entropy(R134a,T=T[2],P=P[2])-entropy(R134a,T=T[1],P=P[1]))
q_EES=T[1]*DELTAs_EES

s_2=entropy(R134a,T=T[2],P=P[2])
s_1=entropy(R134a,T=T[1],P=P[1])

"Conservation of energy for the closed system:"
DELTAu_EES=intEnergy(R134a,T=T[2],p=P[2])-intEnergy(R134a,T=T[1],P=P[1])
q_EES-w_EES=DELTAu_EES
u_1=intEnergy(R134a,T=T[1],P=P[1])
u_2=intEnergy(R134a,T=T[2],p=P[2])

"***** COMPRESSABILITY CHART  SOLUTION ******"
"State 1"
Tr[1]=T[1]/T_critical
pr[1]=p[1]/p_critical
Z[1]=COMPRESS(Tr[1], Pr[1])
DELTAh[1]=ENTHDEP(Tr[1], Pr[1])*R*T_critical"Enthalpy departure"
Z_h1=ENTHDEP(Tr[1], Pr[1])
DELTAs[1]=ENTRDEP(Tr[1], Pr[1])*R "Entropy departure"
Z_s1=ENTRDEP(Tr[1], Pr[1])

"State 2"
Tr[2]=T[2]/T_critical
Pr[2]=P[2]/P_critical
Z[2]=COMPRESS(Tr[2], Pr[2])
DELTAh[2]=ENTHDEP(Tr[2], Pr[2])*R*T_critical"Enthalpy departure"
Z_h2=ENTHDEP(Tr[2], Pr[2])
DELTAs[2]=ENTRDEP(Tr[2], Pr[2])*R "Entropy departure"
Z_s2=ENTRDEP(Tr[2], Pr[2])

"Entropy Change"
DELTAs_ideal= -R*ln(P[2]/P[1])
DELTAs_chart=DELTAs_ideal-(DELTAs[2]-DELTAs[1])

"For the isothermal process, the heat transfer is T*(s[2] - s[1]):"
q_chart=T[1]*DELTAs_chart

"Conservation of energy for the closed system:"
DELTAh_ideal=0
DELTAu_chart=DELTAh_ideal-(DELTAh[2]-DELTAh[1])-(Z[2]*R*T[2]-Z[1]*R*T[1])
q_chart-w_chart=DELTAu_chart
```

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SOLUTION

DELTAh[1]=146.97 [kJ/kg]
DELTAh[2]=1.04 [kJ/kg]
DELTAh_ideal=0 [kJ/kg]
DELTAs[1]=0.4194 [kJ/kg-K]
DELTAs[2]=0.002147 [kJ/kg-K]
DELTAs_chart=0.6613 [kJ/kg-K]
DELTAs_EES=0.7568 [kJ/kg-K]
DELTAs_ideal=0.2441 [kJ/kg-K]
DELTAu_chart=122.92 [kJ/kg]
DELTAu_EES=157.9 [kJ/kg]
M=102
P[1]=2000 [kPa]
P[2]=100 [kPa]
pr[1]=0.4927
Pr[2]=0.02464
P_critical=4059 [kpa]
q_chart=207.09 [kJ/kg]
q_EES=237.00 [kJ/kg]
R=0.08148 [kJ/kg-K]
R_u=8.314 [kJ/kmol-K]
s_1=0.3916
s_2=1.1484 [kJ/kg-K]
T[1]=313.2 [K]
T[2]=313.2 [K]
Tr[1]=0.8369
Tr[2]=0.8369
T_critical=374.2 [K]
u_1=106.37
u_2=264.25 [kJ/kg]
w_chart=84.18 [kJ/kg]
w_EES=79.12 [kJ/kg]
Z[1]=0.08357
Z[2]=0.9857
Z_h1=4.82
Z_h2=0.03396
Z_s1=5.147
Z_s2=0.02635

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12-102 The heat transfer, work, and entropy changes of methane during a process in a piston-cylinder device are to be determined assuming ideal gas behavior, using generalized charts, and real fluid (EES) data.

Analysis The ideal gas solution: (Properties are obtained from EES)

State 1:

$$T_1 = 100°\text{C} \longrightarrow h_1 = -4492\ \text{kJ/kg}$$
$$T_1 = 100°\text{C},\ P_1 = 4\ \text{MPa} \longrightarrow s_1 = 10.22\ \text{kJ/kg.K}$$
$$u_1 = h_1 - RT_1 = (-4492) - (0.5182)(100 + 273.15) = -4685\ \text{kJ/kg}$$
$$\nu_1 = R\frac{T_1}{P_1} = (0.5182\ \text{kJ/kg.K})\left(\frac{100 + 273.15\ \text{K}}{4000\ \text{kPa}}\right) = 0.04834\ \text{m}^3/\text{kg}$$

State 2:

$$T_2 = 350°\text{C} \longrightarrow h_2 = -3770\ \text{kJ/kg}$$
$$T_2 = 350°\text{C},\ P_2 = 4\ \text{MPa} \longrightarrow s_2 = 11.68\ \text{kJ/kg.K}$$
$$u_2 = h_2 - RT_2 = (-3770) - (0.5182)(350 + 273.15) = -4093\ \text{kJ/kg}$$
$$\nu_2 = R\frac{T_2}{P_2} = (0.5182\ \text{kJ/kg.K})\left(\frac{350 + 273.15\ \text{K}}{4000\ \text{kPa}}\right) = 0.08073\ \text{m}^3/\text{kg}$$

$$w_{\text{ideal}} = P(\nu_2 - \nu_1) = (4000\ \text{kPa})(0.08073 - 0.04834)\text{m}^3/\text{kg} = \mathbf{129.56\ kJ/kg}$$

$$q_{\text{ideal}} = w_{\text{ideal}} + (u_2 - u_1) = 129.56 + \left[(-4093) - (-4685)\right] = \mathbf{721.70\ kJ/kg}$$

$$\Delta s_{\text{ideal}} = s_2 - s_1 = 11.68 - 10.22 = \mathbf{1.46\ kJ/kg}$$

For the generalized chart solution we first determine the following factors using EES as

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{373}{304.2} = 1.227 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{4}{7.39} = 0.5413 \end{aligned}\right\} \longrightarrow Z_1 = 0.9023,\ Z_{h1} = 0.4318 \ \text{ and } \ Z_{s1} = 0.2555$$

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{623}{304.2} = 2.048 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{4}{7.39} = 0.5413 \end{aligned}\right\} \longrightarrow Z_2 = 0.995,\ Z_{h2} = 0.1435 \ \text{ and } \ Z_{s2} = 0.06446$$

State 1:

$$\Delta h_1 = Z_{h1}RT_{\text{cr}} = (0.4318)(0.5182\ \text{kJ/kg.K})(304.2\ \text{K}) = 68.07\ \text{kJ/kg}$$
$$h_1 = h_{1,\text{ideal}} - \Delta h_1 = (-4492) - 68.07 = -4560\ \text{kJ/kg}$$
$$u_1 = h_1 - Z_1RT_1 = (-4560) - (0.9023)(0.5182)(373.15) = -4734\ \text{kJ/kg}$$

$$\nu_1 = Z_1R\frac{T_1}{P_1} = (0.9023)(0.5182)\frac{373.15}{4000} = 0.04362\ \text{m}^3/\text{kg}$$

$$\Delta s_1 = Z_{s1}R = (0.2555)(0.5182\ \text{kJ/kg.K}) = 0.1324\ \text{kJ/kg.K}$$
$$s_1 = s_{1,\text{ideal}} - \Delta s_1 = 10.22 - 0.1324 = 10.09\ \text{kJ/kg.K}$$

State 2:

$$\Delta h_2 = Z_{h2}RT_{\text{cr}} = (0.1435)(0.5182\ \text{kJ/kg.K})(304.2\ \text{K}) = 22.62\ \text{kJ/kg}$$
$$h_2 = h_{2,\text{ideal}} - \Delta h_2 = (-3770) - 22.62 = -3793\ \text{kJ/kg}$$
$$u_2 = h_2 - Z_2RT_2 = (-3793) - (0.995)(0.5182)(623.15) = -4114\ \text{kJ/kg}$$

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$$\nu_2 = Z_2 R \frac{T_2}{P_2} = (0.995)(0.5182)\frac{623.15}{4000} = 0.08033\ \text{m}^3/\text{kg}$$

$$\Delta s_2 = Z_{s2} R = (0.06446)(0.5182\ \text{kJ/kg.K}) = 0.03341\ \text{kJ/kg.K}$$

$$s_2 = s_{2,\text{ideal}} - \Delta s_2 = 11.68 - 0.03341 = 11.65\ \text{kJ/kg.K}$$

Then,

$$w_{\text{chart}} = P(\nu_2 - \nu_1) = (4000\ \text{kPa})(0.08033 - 0.04362)\text{m}^3/\text{kg} = \mathbf{146.84\ kJ/kg}$$

$$q_{\text{chart}} = w_{\text{chart}} + (u_2 - u_1) = 146.84 + \left[(-4114) - (-4734)\right] = \mathbf{766.84\ kJ/kg}$$

$$\Delta s_{\text{chart}} = s_2 - s_1 = 11.65 - 10.09 = \mathbf{1.56\ kJ/kg}$$

The solution using EES built-in property data is as follows:

$$\left.\begin{array}{l} T_1 = 100°\text{C} \\ P_1 = 4\ \text{MPa} \end{array}\right\} \begin{array}{l} \nu_1 = 0.04717\ \text{m}^3/\text{kg} \\ u_1 = -39.82\ \text{kJ/kg} \\ s_1 = -1.439\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 350°\text{C} \\ P_2 = 4\ \text{MPa} \end{array}\right\} \begin{array}{l} \nu_2 = 0.08141\ \text{m}^3/\text{kg} \\ u_2 = 564.52\ \text{kJ/kg} \\ s_2 = 0.06329\ \text{kJ/kg.K} \end{array}$$

$$w_{\text{EES}} = P(\nu_2 - \nu_1) = (4000\ \text{kPa})(0.08141 - 0.04717)\text{m}^3/\text{kg} = \mathbf{136.96\ kJ/kg}$$

$$q_{\text{EES}} = w_{\text{EES}} + (u_2 - u_1) = 136.97 + \left[564.52 - (-39.82)\right] = \mathbf{741.31\ kJ/kg}$$

$$\Delta s_{\text{EES}} = s_2 - s_1 = 0.06329 - (-1.439) = \mathbf{1.50\ kJ/kg}$$

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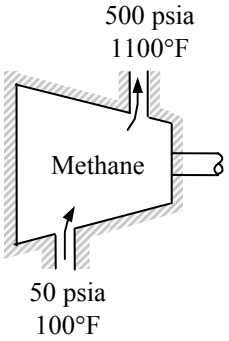
12-103E Methane is compressed steadily. The entropy change and the specific work required are to be determined using the departure charts and the property tables.

Properties The properties of methane are (Table A-1E)

$M = 16.043$ lbm/lbmol, $T_{cr} = 343.9$ R, $P_{cr} = 673$ psia, $c_p = 0.532$ Btu/lbm · R, $R = 0.1238$ Btu/lbm · R

Analysis (*a*) Using empirical correlation for the c_p of methane as given in Table A-2Ec gives

$$\begin{aligned}\bar{h}_2 - \bar{h}_1 &= \int c_p dT = \int (a + bT + cT^2 + dT^3)dT \\ &= a(T_2 - T_1) + \frac{b}{2}(T_2^2 - T_1^2) + \frac{c}{3}(T_2^3 - T_1^3) + \frac{d}{4}(T_2^4 - T_1^4) \\ &= 4.750(1000) + \frac{0.6666\times10^{-2}}{2}(1560^2 - 560^2) + \frac{0.09352\times10^{-5}}{3}(1560^3 - 560^3) \\ &\quad + \frac{-0.4510\times10^{-9}}{4}(1560^4 - 560^4) \\ &= 12{,}288 \text{ Btu/lbmol} \cdot \text{R}\end{aligned}$$



The work input is equal to the enthalpy change. The enthalpy change per unit mass is

$$w_{in} = h_2 - h_1 = \frac{\bar{h}_2 - \bar{h}_1}{M} = \frac{12{,}288 \text{ Btu/lbmol}}{16.043 \text{ lbm/lbmol}} = \mathbf{765.9\ Btu/lbm}$$

Similarly, the entropy change is given by

$$\begin{aligned}(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} &= \int_1^2 \frac{c_p}{T} dT - R_u \ln\frac{P_2}{P_1} = \int_1^2 \left(\frac{a}{T} + b + cT + dT^2\right)dT - R_u \ln\frac{P_2}{P_1} \\ &= a\ln\frac{T_2}{T_1} + b(T_2 - T_1) + \frac{c}{2}(T_2^2 - T_1^2) + \frac{d}{3}(T_2^3 - T_1^3) - R_u \ln\frac{P_2}{P_1} \\ &= 4.750\ln\frac{1560}{560} + 0.6666\times10^{-2}(1560 - 560) + \frac{0.09352\times10^{-5}}{2}(1560^2 - 560^2) \\ &\quad + \frac{-0.4510\times10^{-9}}{3}(1560^3 - 560^3) - (1.9858)\ln\frac{500}{50} \\ &= 7.407 \text{ Btu/lbmol} \cdot \text{R}\end{aligned}$$

The entropy change per unit mass is

$$(s_2 - s_1)_{\text{ideal}} = \frac{(\bar{s}_2 - \bar{s}_1)_{\text{ideal}}}{M} = \frac{7.407 \text{ Btu/lbmol} \cdot \text{R}}{16.043 \text{ lbm/lbmol}} = \mathbf{0.4617\ Btu/lbm \cdot R}$$

(b) The enthalpy and entropy departures of water vapor at the specified states are determined from the generalized charts to be (Figs. A-29, A-30 or from EES. We used EES.)

$$\left.\begin{aligned}T_{R1} &= \frac{T_1}{T_{cr}} = \frac{560}{343.9} = 1.63 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{50}{673} = 0.0743\end{aligned}\right\} \longrightarrow Z_{h1} = 0.03313 \text{ and } Z_{s1} = 0.01617$$

and

$$\left.\begin{aligned}T_{R2} &= \frac{T_2}{T_{cr}} = \frac{1560}{343.9} = 4.54 \\ P_{R2} &= \frac{P_2}{P_{cr}} = \frac{500}{673} = 0.743\end{aligned}\right\} \longrightarrow Z_{h2} = 0 \text{ and } Z_{s2} = 0.00695$$

The work input and entropy changes are

$$\begin{aligned}w_{in} &= h_2 - h_1 = (h_2 - h_1)_{\text{ideal}} - RT_{cr}(Z_{h2} - Z_{h1}) \\ &= 765.9 - (0.1238)(343.9)(0 - 0.03313) = \mathbf{767.4\ Btu/lbm}\end{aligned}$$

$$\begin{aligned}s_2 - s_1 &= (s_2 - s_1)_{\text{ideal}} - R(Z_{s2} - Z_{s1}) \\ &= 0.4617 - (0.1238)(0.00695 - 0.01617) = \mathbf{0.4628\ Btu/lbm \cdot R}\end{aligned}$$

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12-104E Methane is compressed in a steady-flow device. The second-law efficiency of the compression process is to be determined.

Analysis The reversible work input to the compressor is determined from

$$w_{\text{rev}} = h_2 - h_1 - T_0(s_2 - s_1) = 767.4\ \text{Btu/lbm} - (537\ \text{R})(0.4628\ \text{Btu/lbm}\cdot\text{R}) = 518.8\ \text{Btu/lbm}$$

The second-law efficiency of the compressor is

$$\eta_{\text{II}} = \frac{w_{\text{rev}}}{w_{\text{actual}}} = \frac{518.8}{767.4} = 0.676 = \mathbf{67.6\%}$$

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Fundamentals of Engineering (FE) Exam Problems

12-105 A substance whose Joule-Thomson coefficient is negative is throttled to a lower pressure. During this process, (select the correct statement)

(a) the temperature of the substance will increase.
(b) the temperature of the substance will decrease.
(c) the entropy of the substance will remain constant.
(d) the entropy of the substance will decrease.
(e) the enthalpy of the substance will decrease.

Answer (a) the temperature of the substance will increase.

12-106 Consider the liquid-vapor saturation curve of a pure substance on the *P-T* diagram. The magnitude of the slope of the tangent line to this curve at a temperature *T* (in Kelvin) is

(a) proportional to the enthalpy of vaporization h_{fg} at that temperature,
(b) proportional to the temperature *T*,
(c) proportional to the square of the temperature *T,*
(d) proportional to the volume change v_{fg} at that temperature,
(e) inversely proportional to the entropy change s_{fg} at that temperature,

Answer (a) proportional to the enthalpy of vaporization h_{fg} at that temperature,

12-107 Based on the generalized charts, the error involved in the enthalpy of CO_2 at 300 K and 5 MPa if it is assumed to be an ideal gas is

(a) 0 (b) 9% (c) 16% (d) 22% (e) 27%

Answer (e) 27%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=300 "K"
P=5000 "kPa"
Pcr=P_CRIT(CarbonDioxide)
Tcr=T_CRIT(CarbonDioxide)
Tr=T/Tcr
Pr=P/Pcr
hR=ENTHDEP(Tr, Pr)
h_ideal=11351/Molarmass(CO2) "Table A-20 of the text"
h_chart=h_ideal-R*Tcr*hR
R=0.1889
Error=(h_chart-h_ideal)/h_chart*Convert(, %)
```

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12-108 Based on data from the refrigerant-134a tables, the Joule-Thompson coefficient of refrigerant-134a at 0.8 MPa and 100°C is approximately

(a) 0 (b) -5°C/MPa (c) 11°C/MPa (d) 8°C/MPa (e) 26°C/MPa

Answer (c) 11°C/MPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=100 "C"
P1=800 "kPa"
h1=ENTHALPY(R134a,T=T1,P=P1)
Tlow=TEMPERATURE(R134a,h=h1,P=P1+100)
Thigh=TEMPERATURE(R134a,h=h1,P=P1-100)
JT=(Tlow-Thigh)/200
```

12-109 For a gas whose equation of state is $P(v - b) = RT$, the specific heat difference $c_p - c_v$ is equal to

(a) R (b) $R - b$ (c) $R + b$ (d) 0 (e) $R(1 + v/b)$

Answer (a) R

Solution The general relation for the specific heat difference $c_p - c_v$ is

$$c_p - c_v = -T\left(\frac{\partial v}{\partial T}\right)_P^2 \left(\frac{\partial P}{\partial v}\right)_T$$

For the given gas, $P(v - b) = RT$. Then,

$$v = \frac{RT}{P} + b \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

$$P = \frac{RT}{v - b} \longrightarrow \left(\frac{\partial P}{\partial v}\right)_T = -\frac{RT}{(v-b)^2} = -\frac{P}{v-b}$$

Substituting,

$$c_p - c_v = -T\left(\frac{R}{P}\right)^2 \left(-\frac{P}{v-b}\right) = \frac{TR^2}{P(v-b)} = R$$

12-110 ··· 12-112 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 13
GAS MIXTURES

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Composition of Gas Mixtures

13-1C The ratio of the mass of a component to the mass of the mixture is called the mass fraction (*mf*), and the ratio of the mole number of a component to the mole number of the mixture is called the mole fraction (*y*).

13-2C The mass fractions will be identical, but the mole fractions will not.

13-3C Yes.

13-4C Yes, because both CO_2 and N_2O has the same molar mass, M = 44 kg/kmol.

13-5C No. We can do this only when each gas has the same mole fraction.

13-6C It is the average or the equivalent molar mass of the gas mixture. No.

13-7 From the definition of mass fraction,

$$\text{mf}_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \left(\frac{M_i}{M_m} \right)$$

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13-8 A mixture consists of two gases. Relations for mole fractions when mass fractions are known are to be obtained .

Analysis The mass fractions of *A* and *B* are expressed as

$$\text{mf}_A = \frac{m_A}{m_m} = \frac{N_A M_A}{N_m M_m} = y_A \frac{M_A}{M_m} \qquad \text{and} \qquad \text{mf}_B = y_B \frac{M_B}{M_m}$$

Where *m* is mass, *M* is the molar mass, *N* is the number of moles, and *y* is the mole fraction. The apparent molar mass of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{N_A M_A + N_B M_B}{N_m} = y_A M_A + y_B M_B$$

Combining the two equation above and noting that $y_A + y_B = 1$ gives the following convenient relations for converting mass fractions to mole fractions,

$$y_A = \frac{M_B}{M_A(1/\text{mf}_A - 1) + M_B} \qquad \text{and} \qquad y_B = 1 - y_A$$

which are the desired relations.

13-9 The definitions for the mass fraction, weight, and the weight fractions are

$$(\text{mf})_i = \frac{m_i}{m_{\text{total}}}$$
$$W = mg$$
$$(\text{wf})_i = \frac{W_i}{W_{\text{total}}}$$

Since the total system consists of one mass unit, the mass of the *i*th component in this mixture is m_i. The weight of this one component is then

$$W_i = g(\text{mf})_i$$

Hence, the weight fraction for this one component is

$$(\text{wf})_i = \frac{g(\text{mf})_i}{\sum g(\text{mf})_i} = (\text{mf})_i$$

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13-10E The moles of components of a gas mixture are given. The mole fractions and the apparent molecular weight are to be determined.

Properties The molar masses of He, O_2, N_2, and H_2O are 4.0, 32.0, 28.0 and 18.0 lbm/lbmol, respectively (Table A-1).

Analysis The total mole number of the mixture is

$$N_m = N_{\text{He}} + N_{\text{O2}} + N_{\text{H2O}} + N_{\text{N2}} = 3 + 1.5 + 0.3 + 2.5 = 7.3\,\text{lbmol}$$

and the mole fractions are

$$y_{\text{He}} = \frac{N_{\text{He}}}{N_m} = \frac{3\,\text{lbmol}}{7.3\,\text{lbmol}} = \mathbf{0.411}$$

$$y_{\text{O2}} = \frac{N_{\text{O2}}}{N_m} = \frac{1.5\,\text{lbmol}}{7.3\,\text{lbmol}} = \mathbf{0.206}$$

$$y_{\text{H2O}} = \frac{N_{\text{H2O}}}{N_m} = \frac{0.3\,\text{lbmol}}{7.3\,\text{lbmol}} = \mathbf{0.0411}$$

$$y_{\text{N2}} = \frac{N_{\text{N2}}}{N_m} = \frac{2.5\,\text{lbmol}}{7.3\,\text{lbmol}} = \mathbf{0.343}$$

3 lbmol He
1.5 lbmol O_2
0.3 lbmol H_2O
2.5 lbmol N_2

The total mass of the mixture is

$$\begin{aligned} m_m &= m_{\text{He}} + m_{\text{O2}} + m_{\text{H2O}} + + m_{\text{N2}} \\ &= N_{\text{He}} M_{\text{He}} + N_{\text{O2}} M_{\text{O2}} + N_{\text{H2O}} M_{\text{H2O}} + N_{\text{N2}} M_{\text{N2}} \\ &= (3\,\text{lbm})(4\,\text{lbm/lbmol}) + (1.5\,\text{lbm})(32\,\text{lbm/lbmol}) + (0.3\,\text{lbm})(18\,\text{lbm/lbmol}) + (2.5\,\text{lbm})(28\,\text{lbm/lbmol}) \\ &= 135.4\,\text{lbm} \end{aligned}$$

Then the apparent molecular weight of the mixture becomes

$$M_m = \frac{m_m}{N_m} = \frac{135.4\,\text{lbm}}{7.3\,\text{lbmol}} = \mathbf{18.6\ lbm/lbmol}$$

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13-11 The masses of the constituents of a gas mixture are given. The mass fractions, the mole fractions, the average molar mass, and gas constant are to be determined.

Properties The molar masses of O_2, N_2, and CO_2 are 32.0, 28.0 and 44.0 kg/kmol, respectively (Table A-1)

Analysis (*a*) The total mass of the mixture is

$$m_m = m_{O_2} + m_{N_2} + m_{CO_2} = 5\,\text{kg} + 8\,\text{kg} + 10\,\text{kg} = 23\,\text{kg}$$

Then the mass fraction of each component becomes

5 kg O_2
8 kg N_2
10 kg CO_2

$$\text{mf}_{O_2} = \frac{m_{O_2}}{m_m} = \frac{5\,\text{kg}}{23\,\text{kg}} = \mathbf{0.217}$$

$$\text{mf}_{N_2} = \frac{m_{N_2}}{m_m} = \frac{8\,\text{kg}}{23\,\text{kg}} = \mathbf{0.348}$$

$$\text{mf}_{CO_2} = \frac{m_{CO_2}}{m_m} = \frac{10\,\text{kg}}{23\,\text{kg}} = \mathbf{0.435}$$

(*b*) To find the mole fractions, we need to determine the mole numbers of each component first,

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{5\,\text{kg}}{32\,\text{kg/kmol}} = 0.156\,\text{kmol}$$

$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{8\,\text{kg}}{28\,\text{kg/kmol}} = 0.286\,\text{kmol}$$

$$N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{10\,\text{kg}}{44\,\text{kg/kmol}} = 0.227\,\text{kmol}$$

Thus,

$$N_m = N_{O_2} + N_{N_2} + N_{CO_2} = 0.156\,\text{kmol} + 0.286\,\text{kmol} + 0.227\,\text{kmol} = 0.669\,\text{kmol}$$

and

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{0.156\,\text{kmol}}{0.699\,\text{kmol}} = \mathbf{0.233}$$

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{0.286\,\text{kmol}}{0.669\,\text{kmol}} = \mathbf{0.428}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{0.227\,\text{kmol}}{0.669\,\text{kmol}} = \mathbf{0.339}$$

(*c*) The average molar mass and gas constant of the mixture are determined from their definitions:

$$M_m = \frac{m_m}{N_m} = \frac{23\,\text{kg}}{0.669\,\text{kmol}} = \mathbf{34.4\,kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314\,\text{kJ/kmol}\cdot\text{K}}{34.4\,\text{kg/kmol}} = \mathbf{0.242\,kJ/kg\cdot K}$$

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13-12 The mass fractions of the constituents of a gas mixture are given. The mole fractions of the gas and gas constant are to be determined.

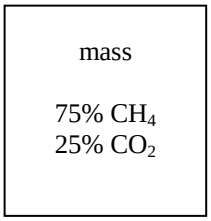
Properties The molar masses of CH_4, and CO_2 are 16.0 and 44.0 kg/kmol, respectively (Table A-1)

Analysis For convenience, consider 100 kg of the mixture. Then the number of moles of each component and the total number of moles are

$$m_{CH_4} = 75 \text{ kg} \longrightarrow N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{75 \text{ kg}}{16 \text{ kg/kmol}} = 4.688 \text{ kmol}$$

$$m_{CO_2} = 25 \text{ kg} \longrightarrow N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{25 \text{ kg}}{44 \text{ kg/kmol}} = 0.568 \text{ kmol}$$

$$N_m = N_{CH_4} + N_{CO_2} = 4.688 \text{ kmol} + 0.568 \text{ kmol} = 5.256 \text{ kmol}$$



Then the mole fraction of each component becomes

$$y_{CH_4} = \frac{N_{CH_4}}{N_m} = \frac{4.688 \text{ kmol}}{5.256 \text{ kmol}} = 0.892 \text{ or } \mathbf{89.2\%}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{0.568 \text{ kmol}}{5.256 \text{ kmol}} = 0.108 \text{ or } \mathbf{10.8\%}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{100 \text{ kg}}{5.256 \text{ kmol}} = 19.03 \text{ kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{19.03 \text{ kg/kmol}} = \mathbf{0.437 \text{ kJ/kg} \cdot \text{K}}$$

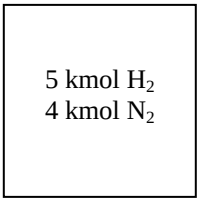
13-13 The mole numbers of the constituents of a gas mixture are given. The mass of each gas and the apparent gas constant are to be determined.

Properties The molar masses of H_2, and N_2 are 2.0 and 28.0 kg/kmol, respectively (Table A-1)

Analysis The mass of each component is determined from

$$N_{H_2} = 5 \text{ kmol} \longrightarrow m_{H_2} = N_{H_2} M_{H_2} = (5 \text{ kmol})(2.0 \text{ kg/kmol}) = \mathbf{10 \text{ kg}}$$

$$N_{N_2} = 4 \text{ kmol} \longrightarrow m_{N_2} = N_{N_2} M_{N_2} = (4 \text{ kmol})(28 \text{ kg/kmol}) = \mathbf{112 \text{ kg}}$$



The total mass and the total number of moles are

$$m_m = m_{H_2} + m_{N_2} = 10 \text{ kg} + 112 \text{ kg} = 122 \text{ kg}$$

$$N_m = N_{H_2} + N_{N_2} = 5 \text{ kmol} + 4 \text{ kmol} = 9 \text{ kmol}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{122 \text{ kg}}{9 \text{ kmol}} = 13.56 \text{ kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{13.56 \text{ kg/kmol}} = \mathbf{0.613 \text{ kJ/kg} \cdot \text{K}}$$

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13-14 The mass fractions of the constituents of a gas mixture are given. The volumetric analysis of the mixture and the apparent gas constant are to be determined.

Properties The molar masses of O_2, N_2 and CO_2 are 32.0, 28, and 44.0 kg/kmol, respectively (Table A-1)

Analysis For convenience, consider 100 kg of the mixture. Then the number of moles of each component and the total number of moles are

$$m_{O_2} = 20 \text{ kg} \longrightarrow N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{20 \text{ kg}}{32 \text{ kg/kmol}} = 0.625 \text{ kmol}$$

$$m_{N_2} = 20 \text{ kg} \longrightarrow N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{30 \text{ kg}}{28 \text{ kg/kmol}} = 1.071 \text{ kmol}$$

$$m_{CO_2} = 50 \text{ kg} \longrightarrow N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{50 \text{ kg}}{44 \text{ kg/kmol}} = 1.136 \text{ kmol}$$

mass

20% O_2
30% N_2
50% CO_2

$$N_m = N_{O_2} + N_{N_2} + N_{CO_2} = 0.625 + 1.071 + 1.136 = 2.832 \text{ kmol}$$

Noting that the volume fractions are same as the mole fractions, the volume fraction of each component becomes

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{0.625 \text{ kmol}}{2.832 \text{ kmol}} = 0.221 \text{ or } \mathbf{22.1\%}$$

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{1.071 \text{ kmol}}{2.832 \text{ kmol}} = 0.378 \text{ or } \mathbf{37.8\%}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{1.136 \text{ kmol}}{2.832 \text{ kmol}} = 0.401 \text{ or } \mathbf{40.1\%}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{100 \text{ kg}}{2.832 \text{ kmol}} = 35.31 \text{ kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{35.31 \text{ kg/kmol}} = \mathbf{0.235 \text{ kJ/kg} \cdot \text{K}}$$

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P-v-T Behavior of Gas Mixtures

13-15C Normally yes. Air, for example, behaves as an ideal gas in the range of temperatures and pressures at which oxygen and nitrogen behave as ideal gases.

13-16C The pressure of a gas mixture is equal to the sum of the pressures each gas would exert if existed alone at the mixture temperature and volume. This law holds exactly for ideal gas mixtures, but only approximately for real gas mixtures.

13-17C The volume of a gas mixture is equal to the sum of the volumes each gas would occupy if existed alone at the mixture temperature and pressure. This law holds exactly for ideal gas mixtures, but only approximately for real gas mixtures.

13-18C The *P-v-T* behavior of a component in an ideal gas mixture is expressed by the ideal gas equation of state using the properties of the individual component instead of the mixture, $P_i v_i = R_i T_i$. The *P-v-T* behavior of a component in a real gas mixture is expressed by more complex equations of state, or by $P_i v_i = Z_i R_i T_i$, where Z_i is the compressibility factor.

13-19C Component pressure is the pressure a component would exert if existed alone at the mixture temperature and volume. Partial pressure is the quantity $y_i P_m$, where y_i is the mole fraction of component *i*. These two are identical for ideal gases.

13-20C Component volume is the volume a component would occupy if existed alone at the mixture temperature and pressure. Partial volume is the quantity $y_i V_m$, where y_i is the mole fraction of component *i*. These two are identical for ideal gases.

13-21C The one with the highest mole number.

13-22C The partial pressures will decrease but the pressure fractions will remain the same.

13-23C The partial pressures will increase but the pressure fractions will remain the same.

13-24C No. The correct expression is "the volume of a gas mixture is equal to the sum of the volumes each gas would occupy if existed alone at the mixture temperature and pressure."

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13-25C No. The correct expression is "the temperature of a gas mixture is equal to the temperature of the individual gas components."

13-26C Yes, it is correct.

13-27C With Kay's rule, a real-gas mixture is treated as a pure substance whose critical pressure and temperature are defined in terms of the critical pressures and temperatures of the mixture components as

$$P'_{\text{cr},m} = \sum y_i P_{\text{cr},i} \quad \text{and} \quad T'_{\text{cr},m} = \sum y_i T_{\text{cr},i}$$

The compressibility factor of the mixture (Z_m) is then easily determined using these pseudo-critical point values.

13-28 The partial pressure of R-134a in atmospheric air to form a 100-ppm contaminant is to be determined.

Analysis Noting that volume fractions and mole fractions are equal, the molar fraction of R-134a in air is

$$y_{\text{R134a}} = \frac{100}{10^6} = 0.0001$$

The partial pressure of R-134a in air is then

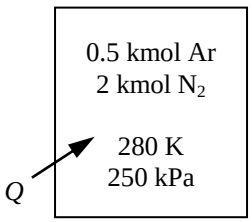
$$P_{\text{R134a}} = y_{\text{R134a}} P_m = (0.0001)(100\ \text{kPa}) = \mathbf{0.01\ kPa}$$

13-29 A tank contains a mixture of two gases of known masses at a specified pressure and temperature. The mixture is now heated to a specified temperature. The volume of the tank and the final pressure of the mixture are to be determined.

Assumptions Under specified conditions both Ar and N_2 can be treated as ideal gases, and the mixture as an ideal gas mixture.

Analysis The total number of moles is

$$N_m = N_{\text{Ar}} + N_{\text{N}_2} = 0.5\ \text{kmol} + 2\ \text{kmol} = 2.5\ \text{kmol}$$



and

$$V_m = \frac{N_m R_u T_m}{P_m} = \frac{(2.5\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(280\ \text{K})}{250\ \text{kPa}} = \mathbf{23.3\ m^3}$$

Also,

$$\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{400\ \text{K}}{280\ \text{K}}(250\ \text{kPa}) = \mathbf{357.1\ kPa}$$

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13-30 The volume fractions of components of a gas mixture are given. The mass fractions and apparent molecular weight of the mixture are to be determined.

Properties The molar masses of H_2, He, and N_2 are 2.0, 4.0, and 28.0 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$m_{H2} = N_{H2} M_{H2} = (30\ \text{kmol})(2\ \text{kg/kmol}) = 60\ \text{kg}$$
$$m_{He} = N_{He} M_{He} = (40\ \text{kmol})(4\ \text{kg/kmol}) = 160\ \text{kg}$$
$$m_{N2} = N_{N2} M_{N2} = (30\ \text{kmol})(28\ \text{kg/kmol}) = 840\ \text{kg}$$

30% H_2
40% He
30% N_2
(by volume)

The total mass is

$$m_m = m_{H2} + m_{He} + N_{N2} = 60 + 160 + 840 = 1060\ \text{kg}$$

Then the mass fractions are

$$\text{mf}_{H2} = \frac{m_{H2}}{m_m} = \frac{60\ \text{kg}}{1060\ \text{kg}} = \mathbf{0.05660}$$

$$\text{mf}_{He} = \frac{m_{He}}{m_m} = \frac{160\ \text{kg}}{1060\ \text{kg}} = \mathbf{0.1509}$$

$$\text{mf}_{N2} = \frac{m_{N2}}{m_m} = \frac{840\ \text{kg}}{1060\ \text{kg}} = \mathbf{0.7925}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{1060\ \text{kg}}{100\ \text{kmol}} = \mathbf{10.60\ kg/kmol}$$

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13-31 The partial pressures of a gas mixture are given. The mole fractions, the mass fractions, the mixture molar mass, the apparent gas constant, the constant-volume specific heat, and the specific heat ratio are to be determined.

Properties The molar masses of CO_2, O_2 and N_2 are 44.0, 32.0, and 28.0 kg/kmol, respectively (Table A-1). The constant-volume specific heats of these gases at 300 K are 0.657, 0.658, and 0.743 kJ/kg·K, respectively (Table A-2a).

Analysis The total pressure is

$$P_{\text{total}} = P_{\text{CO2}} + P_{\text{O2}} + P_{\text{N2}} = 12.5 + 37.5 + 50 = 100 \text{ kPa}$$

Partial pressures
CO_2, 12.5 kPa
O_2, 37.5 kPa
N_2, 50 kPa

The volume fractions are equal to the pressure fractions. Then,

$$y_{\text{CO2}} = \frac{P_{\text{CO2}}}{P_{\text{total}}} = \frac{12.5}{100} = \mathbf{0.125}$$

$$y_{\text{O2}} = \frac{P_{\text{O2}}}{P_{\text{total}}} = \frac{37.5}{100} = \mathbf{0.375}$$

$$y_{\text{N2}} = \frac{P_{\text{N2}}}{P_{\text{total}}} = \frac{50}{100} = \mathbf{0.50}$$

We consider 100 kmol of this mixture. Then the mass of each component are

$$m_{\text{CO2}} = N_{\text{CO2}} M_{\text{CO2}} = (12.5 \text{ kmol})(44 \text{ kg/kmol}) = 550 \text{ kg}$$
$$m_{\text{O2}} = N_{\text{O2}} M_{\text{O2}} = (37.5 \text{ kmol})(32 \text{ kg/kmol}) = 1200 \text{ kg}$$
$$m_{\text{N2}} = N_{\text{N2}} M_{\text{N2}} = (50 \text{ kmol})(28 \text{ kg/kmol}) = 1400 \text{ kg}$$

The total mass is

$$m_m = m_{\text{N2}} + m_{\text{O2}} + m_{\text{Ar}} = 550 + 1200 + 1400 = 3150 \text{ kg}$$

Then the mass fractions are

$$\text{mf}_{\text{CO2}} = \frac{m_{\text{CO2}}}{m_m} = \frac{550 \text{ kg}}{3150 \text{ kg}} = \mathbf{0.1746}$$

$$\text{mf}_{\text{O2}} = \frac{m_{\text{O2}}}{m_m} = \frac{1200 \text{ kg}}{3150 \text{ kg}} = \mathbf{0.3810}$$

$$\text{mf}_{\text{N2}} = \frac{m_{\text{N2}}}{m_m} = \frac{1400 \text{ kg}}{3150 \text{ kg}} = \mathbf{0.4444}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{3150 \text{ kg}}{100 \text{ kmol}} = \mathbf{31.50 \text{ kg/kmol}}$$

The constant-volume specific heat of the mixture is determined from

$$\begin{aligned} c_v &= \text{mf}_{\text{Co2}} c_{v,\text{CO2}} + \text{mf}_{\text{O2}} c_{v,\text{O2}} + \text{mf}_{\text{N2}} c_{v,\text{N2}} \\ &= 0.1746 \times 0.657 + 0.3810 \times 0.658 + 0.4444 \times 0.743 \\ &= \mathbf{0.6956 \text{ kJ/kg} \cdot \text{K}} \end{aligned}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{31.50 \text{ kg/kmol}} = \mathbf{0.2639 \text{ kJ/kg} \cdot \text{K}}$$

The constant-pressure specific heat of the mixture and the specific heat ratio are

$$c_p = c_v + R = 0.6956 + 0.2639 = \mathbf{0.9595 \text{ kJ/kg} \cdot \text{K}}$$

$$k = \frac{c_p}{c_v} = \frac{0.9595 \text{ kJ/kg} \cdot \text{K}}{0.6956 \text{ kJ/kg} \cdot \text{K}} = \mathbf{1.379}$$

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13-32 The mole numbers of combustion gases are given. The partial pressure of water vapor is to be determined.

Analysis The total mole of the mixture and the mole fraction of water vapor are

$$N_{\text{total}} = 0.75 + 1.66 + 5.65 = 8.06 \text{ kmol}$$

$$y_{\text{H2O}} = \frac{N_{\text{H2O}}}{N_{\text{total}}} = \frac{1.66}{8.06} = 0.2060$$

Noting that molar fraction is equal to pressure fraction, the partial pressure of water vapor is

$$P_{\text{H2O}} = y_{\text{H2O}} P_{\text{total}} (0.2060)(101.3 \text{ kPa}) = \mathbf{20.86 \text{ kPa}}$$

13-33 An additional 5% of oxygen is mixed with standard atmospheric air. The molecular weight of this mixture is to be determined.

Properties The molar masses of N_2 and O_2 are 28.0 and 32.0 kg/kmol, respectively (Table A-1).

Analysis Standard air is taken as 79% nitrogen and 21% oxygen by mole. That is,

$$y_{\text{O2}} = 0.21$$
$$y_{\text{N2}} = 0.79$$

Adding another 0.05 moles of O_2 to 1 kmol of standard air gives

$$y_{\text{O2}} = \frac{0.26}{1.05} = 0.2476$$
$$y_{\text{N2}} = \frac{0.79}{1.05} = 0.7524$$

Then,

$$M_m = y_{\text{O2}} M_{\text{O2}} + y_{\text{N2}} M_{\text{N2}} = 0.2476 \times 32 + 0.7524 \times 28 = \mathbf{28.99 \text{ kg/kmol}}$$

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13-34 The masses, temperatures, and pressures of two gases contained in two tanks connected to each other are given. The valve connecting the tanks is opened and the final temperature is measured. The volume of each tank and the final pressure are to be determined.

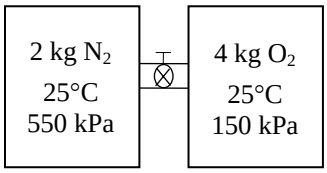
Assumptions Under specified conditions both N_2 and O_2 can be treated as ideal gases, and the mixture as an ideal gas mixture

Properties The molar masses of N_2 and O_2 are 28.0 and 32.0 kg/kmol, respectively. The gas constants of N_2 and O_2 are 0.2968 and 0.2598 kPa·m^3/kg·K, respectively (Table A-1).

Analysis The volumes of the tanks are

$$V_{N_2} = \left(\frac{mRT}{P}\right)_{N_2} = \frac{(2\text{ kg})(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}{550\text{ kPa}} = \mathbf{0.322\ m^3}$$

$$V_{O_2} = \left(\frac{mRT}{P}\right)_{O_2} = \frac{(4\text{ kg})(0.2598\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}{150\text{ kPa}} = \mathbf{2.065\ m^3}$$



$$V_{total} = V_{N_2} + V_{O_2} = 0.322\text{ m}^3 + 2.065\text{ m}^3 = 2.386\text{ m}^3$$

Also,

$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{2\text{ kg}}{28\text{ kg/kmol}} = 0.07143\text{ kmol}$$

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{4\text{ kg}}{32\text{ kg/kmol}} = 0.125\text{ kmol}$$

$$N_m = N_{N_2} + N_{O_2} = 0.07143\text{ kmol} + 0.125\text{ kmol} = 0.1964\text{ kmol}$$

Thus,

$$P_m = \left(\frac{NR_uT}{V}\right)_m = \frac{(0.1964\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(298\text{ K})}{2.386\text{ m}^3} = \mathbf{204\ kPa}$$

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13-35 The masses of components of a gas mixture are given. The apparent molecular weight of this mixture, the volume it occupies, the partial volume of the oxygen, and the partial pressure of the helium are to be determined.

Properties The molar masses of O_2, CO_2, and He are 32.0, 44.0, and 4.0 kg/kmol, respectively (Table A-1).

Analysis The total mass of the mixture is

$$m_m = m_{O2} + m_{CO2} + m_{He} = 0.1 + 1 + 0.5 = 1.6 \text{ kg}$$

The mole numbers of each component are

0.1 kg O_2
1 kg CO_2
0.5 kg He

$$N_{O2} = \frac{m_{O2}}{M_{O2}} = \frac{0.1 \text{ kg}}{32 \text{ kg/kmol}} = 0.003125 \text{ kmol}$$

$$N_{CO2} = \frac{m_{CO2}}{M_{CO2}} = \frac{1 \text{ kg}}{44 \text{ kg/kmol}} = 0.02273 \text{ kmol}$$

$$N_{He} = \frac{m_{He}}{M_{He}} = \frac{0.5 \text{ kg}}{4 \text{ kg/kmol}} = 0.125 \text{ kmol}$$

The mole number of the mixture is

$$N_m = N_{O2} + N_{CO2} + N_{He} = 0.003125 + 0.02273 + 0.125 = 0.15086 \text{ kmol}$$

Then the apparent molecular weight of the mixture becomes

$$M_m = \frac{m_m}{N_m} = \frac{1.6 \text{ kg}}{0.15086 \text{ kmol}} = \mathbf{10.61 \text{ kg/kmol}}$$

The volume of this ideal gas mixture is

$$V_m = \frac{N_m R_u T}{P} = \frac{(0.1509 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(300 \text{ K})}{100 \text{ kPa}} = \mathbf{3.764 \text{ m}^3}$$

The partial volume of oxygen in the mixture is

$$V_{O2} = y_{O2} V_m = \frac{N_{O2}}{N_m} V_m = \frac{0.003125 \text{ kmol}}{0.1509 \text{ kmol}} (3.764 \text{ m}^3) = \mathbf{0.07795 \text{ m}^3}$$

The partial pressure of helium in the mixture is

$$P_{He} = y_{He} P_m = \frac{N_{He}}{N_m} P_m = \frac{0.125 \text{ kmol}}{0.1509 \text{ kmol}} (100 \text{ kPa}) = \mathbf{82.84 \text{ kPa}}$$

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13-36 The mass fractions of components of a gas mixture are given. The volume occupied by 100 kg of this mixture is to be determined.

Properties The molar masses of CH_4, C_3H_8, and C_4H_{10} are 16.0, 44.0, and 58.0 kg/kmol, respectively (Table A-1).

Analysis The mole numbers of each component are

$$N_{CH4} = \frac{m_{CH4}}{M_{CH4}} = \frac{60\ \text{kg}}{16\ \text{kg/kmol}} = 3.75\ \text{kmol}$$

$$N_{C3H8} = \frac{m_{C3H8}}{M_{C3H8}} = \frac{25\ \text{kg}}{44\ \text{kg/kmol}} = 0.5682\ \text{kmol}$$

$$N_{C4H10} = \frac{m_{C4H10}}{M_{C4H10}} = \frac{15\ \text{kg}}{58\ \text{kg/kmol}} = 0.2586\ \text{kmol}$$

60% CH_4
25% C_3H_8
15% C_4H_{10}
(by mass)

The mole number of the mixture is

$$N_m = N_{CH4} + N_{C3H8} + N_{C4H10} = 3.75 + 0.5682 + 0.2586 = 4.5768\ \text{kmol}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{100\ \text{kg}}{4.5768\ \text{kmol}} = 21.85\ \text{kg/kmol}$$

Then the volume of this ideal gas mixture is

$$V_m = \frac{N_m R_u T}{P} = \frac{(4.5768\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(310\ \text{K})}{3000\ \text{kPa}} = \mathbf{3.93\ m^3}$$

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13-37E The mass fractions of components of a gas mixture are given. The mass of 7 ft^3 of this mixture and the partial volumes of the components are to be determined.

Properties The molar masses of N_2, O_2, and He are 28.0, 32.0, and 4.0 lbm/lbmol, respectively (Table A-1E).

Analysis We consider 100 lbm of this mixture for calculating the molar mass of the mixture. The mole numbers of each component are

$$N_{N2} = \frac{m_{N2}}{M_{N2}} = \frac{45\,\text{lbm}}{28\,\text{lbm/lbmol}} = 1.607\,\text{lbmol}$$
$$N_{O2} = \frac{m_{O2}}{M_{O2}} = \frac{35\,\text{lbm}}{32\,\text{lbm/lbmol}} = 1.094\,\text{lbmol}$$
$$N_{He} = \frac{m_{He}}{M_{He}} = \frac{20\,\text{lbm}}{4\,\text{lbm/lbmol}} = 5\,\text{lbmol}$$

7 ft^3
45% N_2
35% O_2
20% He
(by mass)

The mole number of the mixture is

$$N_m = N_{N2} + N_{O2} + N_{He} = 1.607 + 1.094 + 5 = 7.701\,\text{lbmol}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{100\,\text{lbm}}{7.701\,\text{lbmol}} = 12.99\,\text{lbm/lbmol}$$

Then the mass of this ideal gas mixture is

$$m = \frac{PVM_m}{R_u T} = \frac{(300\,\text{psia})(7\,\text{ft}^3)(12.99\,\text{lbm/lbmol})}{(10.73\,\text{psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(520\,\text{R})} = \mathbf{4.887\ lbm}$$

The mole fractions are

$$y_{N2} = \frac{N_{N2}}{N_m} = \frac{1.607\,\text{lbmol}}{7.701\,\text{lbmol}} = 0.2087$$
$$y_{O2} = \frac{N_{O2}}{N_m} = \frac{1.094\,\text{lbmol}}{7.701\,\text{lbmol}} = 0.142$$
$$y_{He} = \frac{N_{He}}{N_m} = \frac{5\,\text{lbmol}}{7.701\,\text{lbmol}} = 0.6493$$

Noting that volume fractions are equal to mole fractions, the partial volumes are determined from

$$V_{N2} = y_{N2}V_m = (0.2087)(7\,\text{ft}^3) = \mathbf{1.461\,ft^3}$$
$$V_{O2} = y_{O2}V_m = (0.142)(7\,\text{ft}^3) = \mathbf{0.994\,ft^3}$$
$$V_{He} = y_{He}V_m = (0.6493)(7\,\text{ft}^3) = \mathbf{4.545\,ft^3}$$

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13-38 The mass fractions of components of a gas mixture are given. The partial pressure of ethane is to be determined.

Properties The molar masses of CH_4 and C_2H_6 are 16.0 and 30.0 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kg of this mixture. The mole numbers of each component are

$$N_{CH4} = \frac{m_{CH4}}{M_{CH4}} = \frac{70\text{ kg}}{16\text{ kg/kmol}} = 4.375\text{ kmol}$$

$$N_{C2H6} = \frac{m_{C2H6}}{M_{C2H6}} = \frac{30\text{ kg}}{30\text{ kg/kmol}} = 1.0\text{ kmol}$$

70% CH_4
30% C_2H_6
(by mass)
100 m^3
130 kPa, 25°C

The mole number of the mixture is

$$N_m = N_{CH4} + N_{C2H6} = 4.375 + 1.0 = 5.375\text{ kmol}$$

The mole fractions are

$$y_{CH4} = \frac{N_{CH4}}{N_m} = \frac{4.375\text{ kmol}}{5.375\text{ kmol}} = 0.8139$$

$$y_{C2H6} = \frac{N_{C2H6}}{N_m} = \frac{1.0\text{ kmol}}{5.375\text{ kmol}} = 0.1861$$

The final pressure of ethane in the final mixture is

$$P_{C2H6} = y_{C2H6} P_m = (0.1861)(130\text{ kPa}) = \mathbf{24.19\text{ kPa}}$$

13-39 A container contains a mixture of two fluids. The volume of the container and the total weight of its contents are to be determined.

Assumptions The volume of the mixture is the sum of the volumes of the two constituents.

Properties The specific volumes of the two fluids are given to be 0.001 m^3/kg and 0.008 m^3/kg.

Analysis The volumes of the two fluids are given by

$$V_A = m_A v_A = (1\text{ kg})(0.001\text{ m}^3/\text{kg}) = 0.001\text{ m}^3$$

$$V_B = m_B v_B = (2\text{ kg})(0.008\text{ m}^3/\text{kg}) = 0.016\text{ m}^3$$

1 kg fluid A
2 kg fluid B

The volume of the container is then

$$V = V_A + V_B = 0.001 + 0.016 = \mathbf{0.017\text{ m}^3}$$

The total mass is

$$m = m_A + m_B = 1 + 2 = 3\text{ kg}$$

and the weight of this mass will be

$$W = mg = (3\text{ kg})(9.6\text{ m/s}^2) = 28.8\text{ kg}\cdot\text{m/s}^2 = \mathbf{28.8\text{ N}}$$

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13-40E A mixture consists of liquid water and another fluid. The specific weight of this mixture is to be determined.

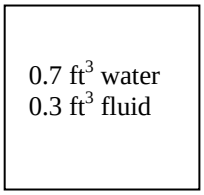
Properties The densities of water and the fluid are given to be 62.4 lbm/ft^3 and 50.0 lbm/ft^3, respectively.

Analysis We consider 1 ft^3 of this mixture. The volume of the water in the mixture is 0.7 ft^3 which has a mass of

$$m_w = \rho_w V_w = (62.4\,\text{lbm/ft}^3)(0.7\,\text{ft}^3) = 43.68\,\text{lbm}$$

The weight of this water is

$$W_w = m_w g = (43.68\,\text{lbm})(31.9\,\text{ft/s}^2)\left(\frac{1\,\text{lbf}}{32.174\,\text{lbm}\cdot\text{ft/s}^2}\right) = 43.31\,\text{lbf}$$



Similarly, the volume of the second fluid is 0.3 ft^3, and the mass of this fluid is

$$m_f = \rho_f V_f = (50\,\text{lbm/ft}^3)(0.3\,\text{ft}^3) = 15\,\text{lbm}$$

The weight of the fluid is

$$W_f = m_f g = (15\,\text{lbm})(31.9\,\text{ft/s}^2)\left(\frac{1\,\text{lbf}}{32.174\,\text{lbm}\cdot\text{ft/s}^2}\right) = 14.87\,\text{lbf}$$

The specific weight of this mixture is then

$$\gamma = \frac{W_w + W_f}{V_w + V_f} = \frac{(43.31+14.87)\,\text{lbf}}{(0.7+0.3)\,\text{ft}^3} = \mathbf{58.2\ lbf/ft^3}$$

13-41 The mole fractions of components of a gas mixture are given. The mass flow rate of the mixture is to be determined.

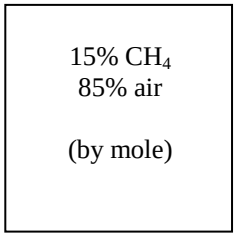
Properties The molar masses of air and CH_4 are 28.97 and 16.0 kg/kmol, respectively (Table A-1).

Analysis The molar fraction of air is

$$y_{\text{air}} = 1 - y_{\text{CH4}} = 1 - 0.15 = 0.85$$

The molar mass of the mixture is determined from

$$\begin{aligned} M_m &= y_{\text{CH4}} M_{\text{CH4}} + y_{\text{air}} M_{\text{air}} \\ &= 0.15\times 16 + 0.85\times 28.97 \\ &= 27.02\ \text{kg/kmol} \end{aligned}$$



Given the engine displacement and speed and assuming that this is a 4-stroke engine (2 revolutions per cycle), the volume flow rate is determined from

$$\dot{V} = \frac{\dot{n} V_d}{2} = \frac{(3000\,\text{rev/min})(0.005\,\text{m}^3)}{2\,\text{rev/cycle}} = 7.5\,\text{m}^3/\text{min}$$

The specific volume of the mixture is

$$v = \frac{R_u T}{M_m P} = \frac{(8.314\,\text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293\,\text{K})}{(27.02\,\text{kg/kmol})(80\,\text{kPa})} = 1.127\,\text{m}^3/\text{kg}$$

Hence the mass flow rate is

$$\dot{m} = \frac{\dot{V}}{v} = \frac{7.5\,\text{m}^3/\text{min}}{1.127\,\text{m}^3/\text{kg}} = \mathbf{6.65\ kg/min}$$

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13-42E The volumetric fractions of components of a natural gas mixture are given. The mass and volume flow rates of the mixture are to be determined.

Properties The molar masses of CH_4 and C_2H_6 are 16.0 and 30.0 lbm/lbmol, respectively (Table A-1E).

Analysis The molar mass of the mixture is determined from

$$M_m = y_{CH4} M_{CH4} + y_{C2H6} M_{C2H6} = 0.95 \times 16 + 0.05 \times 30 = 16.70 \text{ lbm/lbmol}$$

The specific volume of the mixture is

$$v = \frac{R_u T}{M_m P} = \frac{(10.73 \text{ psia} \cdot \text{ft}^3/\text{lbmol} \cdot \text{R})(520 \text{ R})}{(16.70 \text{ lbm/lbmol})(100 \text{ psia})} = 3.341 \text{ ft}^3/\text{lbm}$$

→ 95% CH_4, 5% C_2H_6 (by volume)

The volume flow rate is

$$\dot{V} = AV = \frac{\pi D^2}{4} V = \frac{\pi (36/12 \text{ ft})^2}{4} (10 \text{ ft/s}) = \mathbf{70.69 \text{ ft}^3/\text{s}}$$

and the mass flow rate is

$$\dot{m} = \frac{\dot{V}}{v} = \frac{70.69 \text{ ft}^3/\text{s}}{3.341 \text{ ft}^3/\text{lbm}} = \mathbf{21.16 \text{ lbm/s}}$$

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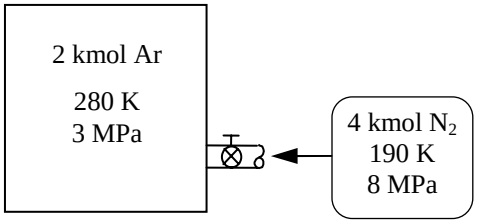
13-43 The mole numbers, temperatures, and pressures of two gases forming a mixture are given. The final temperature is also given. The pressure of the mixture is to be determined using two methods.

Analysis (*a*) Under specified conditions both Ar and N_2 will considerably deviate from the ideal gas behavior. Treating the mixture as an ideal gas,

$$\left.\begin{array}{l}\text{Initial state:}\quad P_1V_1 = N_1R_uT_1 \\ \text{Final state:}\quad P_2V_2 = N_2R_uT_2\end{array}\right\} P_2 = \frac{N_2T_2}{N_1T_1}P_1 = \frac{(6)(230\text{ K})}{(2)(280\text{ K})}(3\text{ MPa}) = \mathbf{7.39\ MPa}$$

(*b*) Initially,

$$\left.\begin{array}{l} T_R = \dfrac{T_1}{T_{\text{cr,Ar}}} = \dfrac{280\text{ K}}{151.0\text{ K}} = 1.854 \\ P_R = \dfrac{P_1}{P_{\text{cr,Ar}}} = \dfrac{3\text{ MPa}}{4.86\text{ MPa}} = 0.6173 \end{array}\right\} Z_{\text{Ar}} = 0.985 \text{ (Fig. A-15 or EES)}$$



Then the volume of the tank is

$$V = \frac{ZN_{\text{Ar}}R_uT}{P} = \frac{(0.985)(2\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(280\text{ K})}{3000\text{ kPa}} = 1.529\text{ m}^3$$

After mixing,

Ar:
$$\left.\begin{array}{l} T_{R,\text{Ar}} = \dfrac{T_m}{T_{\text{cr,Ar}}} = \dfrac{230\text{ K}}{151.0\text{ K}} = 1.523 \\ v_{R,\text{Ar}} = \dfrac{v_{\text{Ar}}}{R_uT_{\text{cr,Ar}}/P_{\text{cr,Ar}}} = \dfrac{V_m/N_{\text{Ar}}}{R_uT_{\text{cr,Ar}}/P_{\text{cr,Ar}}} \\ \quad = \dfrac{(1.529\text{ m}^3)/(2\text{ kmol})}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(151.0\text{ K})/(4860\text{ kPa})} = 2.96 \end{array}\right\} P_R = 0.496 \quad \text{(Fig. A-15 or EES)}$$

N_2:
$$\left.\begin{array}{l} T_{R,\text{N}_2} = \dfrac{T_m}{T_{\text{cr,N}_2}} = \dfrac{230\text{ K}}{126.2\text{ K}} = 1.823 \\ V_{R,\text{N}_2} = \dfrac{v_{\text{N}_2}}{R_uT_{\text{cr,N}_2}/P_{\text{cr,N}_2}} = \dfrac{V_m/N_{\text{N}_2}}{R_uT_{\text{cr,N}_2}/P_{\text{cr,N}_2}} \\ \quad = \dfrac{(1.529\text{ m}^3)/(4\text{ kmol})}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(126.2\text{ K})/(3390\text{ kPa})} = 1.235 \end{array}\right\} P_R = 1.43 \quad \text{(Fig. A-15 or EES)}$$

Thus,

$$P_{\text{Ar}} = (P_RP_{\text{cr}})_{\text{Ar}} = (0.496)(4.86\text{ MPa}) = 2.41\text{ MPa}$$
$$P_{\text{N}_2} = (P_RP_{\text{cr}})_{\text{N}_2} = (1.43)(3.39\text{ MPa}) = 4.85\text{ MPa}$$

and

$$P_m = P_{\text{Ar}} + P_{\text{N}_2} = 2.41\text{ MPa} + 4.85\text{ MPa} = \mathbf{7.26\ MPa}$$

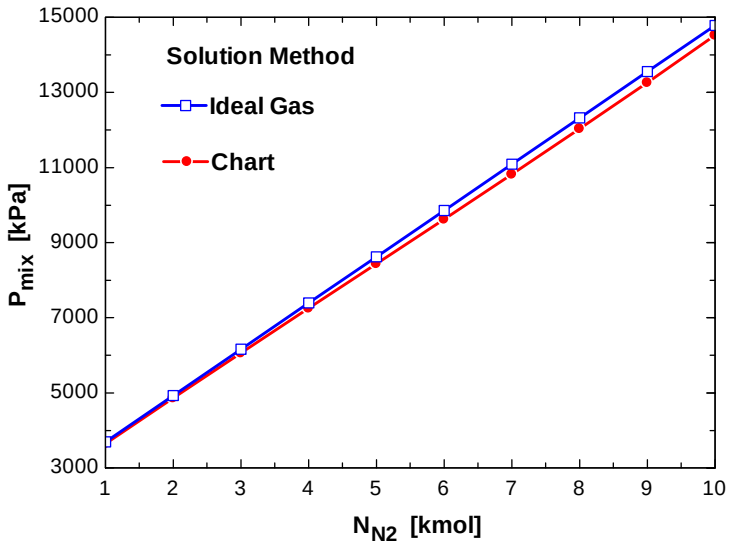
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13-44 Problem 13-43 is reconsidered. The effect of the moles of nitrogen supplied to the tank on the final pressure of the mixture is to be studied using the ideal-gas equation of state and the compressibility chart with Dalton's law.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
R_u = 8.314 [kJ/kmol-K] "universal Gas Constant"
T_Ar = 280 [K]
P_Ar = 3000 [kPa]  "Pressure for only Argon in the tank initially."
N_Ar = 2 [kmol]
{N_N2 = 4 [kmol]}
T_mix = 230 [K]
T_cr_Ar=151.0 [K]  "Critical Constants are found in Table A.1 of the text"
P_cr_Ar=4860 [kPa]
T_cr_N2=126.2 [K]
P_cr_N2=3390 [kPa]
"Ideal-gas Solution:"
P_Ar*V_Tank_IG = N_Ar*R_u*T_Ar  "Apply the ideal gas law the gas in the tank."
P_mix_IG*V_Tank_IG = N_mix*R_u*T_mix  "Ideal-gas mixture pressure"
N_mix=N_Ar + N_N2  "Moles of mixture"
"Real Gas Solution:"
P_Ar*V_Tank_RG = Z_Ar_1*N_Ar*R_u*T_Ar "Real gas volume of tank"
T_R=T_Ar/T_cr_Ar "Initial reduced Temp. of Ar"
P_R=P_Ar/P_cr_Ar "Initial reduced Press. of Ar"
Z_Ar_1=COMPRESS(T_R, P_R ) "Initial compressibility factor for Ar"
P_Ar_mix*V_Tank_RG = Z_Ar_mix*N_Ar*R_u*T_mix  "Real gas Ar Pressure in mixture"
T_R_Ar_mix=T_mix/T_cr_Ar          "Reduced Temp. of Ar in mixture"
P_R_Ar_mix=P_Ar_mix/P_cr_Ar       "Reduced Press. of Ar in mixture"
Z_Ar_mix=COMPRESS(T_R_Ar_mix, P_R_Ar_mix ) "Compressibility factor for Ar in mixture"
P_N2_mix*V_Tank_RG = Z_N2_mix*N_N2*R_u*T_mix "Real gas N2 Pressure in mixture"
T_R_N2_mix=T_mix/T_cr_N2 "Reduced Temp. of N2 in mixture"
P_R_N2_mix=P_N2_mix/P_cr_N2 "Reduced Press. of N2 in mixture"
Z_N2_mix=COMPRESS(T_R_N2_mix, P_R_N2_mix ) "Compressibility factor for N2 in mixture"
P_mix=P_R_Ar_mix*P_cr_Ar +P_R_N2_mix*P_cr_N2 "Mixture pressure by Dalton's law. 23800"
```

N_{N2} [kmol]	P_{mix} [kPa]	$P_{mix,IG}$ [kPa]
1	3647	3696
2	4863	4929
3	6063	6161
4	7253	7393
5	8438	8625
6	9626	9857
7	10822	11089
8	12032	12321
9	13263	13554
10	14521	14786



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13-45E The mass fractions of gases forming a mixture at a specified pressure and temperature are given. The mass of the gas mixture is to be determined using four methods.

Properties The molar masses of CH_4 and C_2H_6 are 16.0 and 30.0 lbm/lbmol, respectively (Table A-1E).

Analysis (*a*) We consider 100 lbm of this mixture. Then the mole numbers of each component are

$$N_{CH4} = \frac{m_{CH4}}{M_{CH4}} = \frac{75\,\text{lbm}}{16\,\text{lbm/lbmol}} = 4.6875\,\text{lbmol}$$

$$N_{C2H6} = \frac{m_{C2H6}}{M_{C2H6}} = \frac{25\,\text{lbm}}{30\,\text{lbm/lbmol}} = 0.8333\,\text{lbmol}$$

75% CH_4
25% C_2H_6
(by mass)
2000 psia
300°F

The mole number of the mixture and the mole fractions are

$$N_m = 4.6875 + 0.8333 = 5.5208\,\text{lbmol}$$

$$y_{CH4} = \frac{N_{CH4}}{N_m} = \frac{4.6875\,\text{lbmol}}{5.5208\,\text{lbmol}} = 0.8491$$

$$y_{C2H6} = \frac{N_{C2H6}}{N_m} = \frac{0.8333\,\text{lbmol}}{5.5208\,\text{lbmol}} = 0.1509$$

Then the apparent molecular weight of the mixture becomes

$$M_m = \frac{m_m}{N_m} = \frac{100\,\text{lbm}}{5.5208\,\text{lbmol}} = 18.11\,\text{lbm/lbmol}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{10.73\,\text{psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R}}{18.11\,\text{lbm/lbmol}} = 0.5925\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$$

The mass of this mixture in a 1 million ft^3 tank is

$$m = \frac{P\mathcal{V}}{RT} = \frac{(2000\,\text{psia})(1\times10^6\,\text{ft}^3)}{(0.5925\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(760\,\text{R})} = \mathbf{4.441\times10^6\ lbm}$$

(*b*) To use the Amagat's law for this real gas mixture, we first need the compressibility factor of each component at the mixture temperature and pressure. The compressibility factors are obtained using Fig. A-15 to be

$$\left.\begin{aligned} T_{R,CH4} &= \frac{T_m}{T_{cr,CH4}} = \frac{760\,\text{R}}{343.9\,\text{R}} = 2.210 \\ P_{R,CH4} &= \frac{P_m}{P_{cr,CH4}} = \frac{2000\,\text{psia}}{673\,\text{psia}} = 2.972 \end{aligned}\right\} Z_{CH4} = 0.98$$

$$\left.\begin{aligned} T_{R,C2H6} &= \frac{760\,\text{R}}{549.8\,\text{R}} = 1.382 \\ P_{R,C2H6} &= \frac{1500\,\text{psia}}{708\,\text{psia}} = 2.119 \end{aligned}\right\} Z_{C2H6} = 0.77$$

Then,

$$Z_m = \sum y_i Z_i = y_{CH4}Z_{CH4} + y_{C2H6}Z_{C2H6} = (0.8491)(0.98) + (0.1509)(0.77) = 0.9483$$

$$m = \frac{P\mathcal{V}}{Z_m RT} = \frac{(2000\,\text{psia})(1\times10^6\,\text{ft}^3)}{(0.9483)(0.5925\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(760\,\text{R})} = \mathbf{4.684\times10^6\ lbm}$$

(*c*) To use Dalton's law with compressibility factors: (Fig. A-15)

$$\left.\begin{aligned} T_{R,CH4} &= 2.210 \\ \upsilon_{R,CH4} &= \frac{\mathcal{V}_m/m_{CH4}}{R_{CH4}T_{cr,CH4}/P_{cr,CH4}} = \frac{(1\times10^6\,\text{ft}^3)/(4.441\times10^6\times0.75\,\text{lbm})}{(0.6688\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(343.9\,\text{R})/(673\,\text{psia})} = 0.8782 \end{aligned}\right\} Z_{CH4} = 0.98$$

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$$\left.\begin{aligned} T_{R,\mathrm{C2H6}} &= 1.382 \\ \nu_{R,\mathrm{C2H6}} &= \frac{\nu_m / m_{\mathrm{C2H6}}}{RT_{\mathrm{cr,C2H6}} / P_{\mathrm{cr,C2H6}}} = \frac{(1\times10^6\ \mathrm{ft}^3)/(4.441\times10^6\times0.25\,\mathrm{lbm})}{(0.3574\,\mathrm{psia}\cdot\mathrm{ft}^3/\mathrm{lbm}\cdot\mathrm{R})(549.8\,\mathrm{R})/(708\,\mathrm{psia})} = 3.244 \end{aligned}\right\} Z_{\mathrm{CH4}} = 0.92$$

Note that we used $m = 4.441\times10^6\times0.25\,\mathrm{lbm}$ in above calculations, the value obtained by ideal gas behavior. The solution normally requires iteration until the assumed and calculated mass values match. The mass of the component gas is obtained by multiplying the mass of the mixture by its mass fraction. Then,

$$Z_m = \sum y_i Z_i = y_{\mathrm{CH4}} Z_{\mathrm{CH4}} + y_{\mathrm{C2H6}} Z_{\mathrm{C2H6}} = (0.8491)(0.98) + (0.1509)(0.92) = 0.9709$$

$$m = \frac{P\nu}{Z_m RT} = \frac{(2000\,\mathrm{psia})(1\times10^6\ \mathrm{ft}^3)}{(0.9709)(0.5925\,\mathrm{psia}\cdot\mathrm{ft}^3/\mathrm{lbm}\cdot\mathrm{R})(760\,\mathrm{R})} = \mathbf{4.575\times10^6\ lbm}$$

This mass is sufficiently close to the assumed mass value of $4.441\times10^6\times0.25\,\mathrm{lbm}$. Therefore, there is no need to repeat the calculations at this calculated mass.

(*d*) To use Kay's rule, we need to determine the pseudo-critical temperature and pseudo-critical pressure of the mixture using the critical point properties of gases.

$$\begin{aligned} T'_{\mathrm{cr},m} &= \sum y_i T_{\mathrm{cr},i} = y_{\mathrm{CH4}} T_{\mathrm{cr,Ch4}} + y_{\mathrm{C2H6}} T_{\mathrm{cr,C2H6}} \\ &= (0.8491)(343.9\,\mathrm{R}) + (0.1509)(549.8\,\mathrm{R}) = 375.0\,\mathrm{R} \\ P'_{\mathrm{cr},m} &= \sum y_i P_{\mathrm{cr},i} = y_{\mathrm{Ch4}} P_{\mathrm{cr,Ch4}} + y_{\mathrm{C2H6}} P_{\mathrm{cr,C2H6}} \\ &= (0.8491)(673\,\mathrm{psia}) + (0.1509)(708\,\mathrm{psia}) = 678.3\,\mathrm{psia} \end{aligned}$$

Then,

$$\left.\begin{aligned} T_R &= \frac{T_m}{T'_{\mathrm{cr},m}} = \frac{760\,\mathrm{R}}{375.0\,\mathrm{R}} = 2.027 \\ P_R &= \frac{P_m}{P'_{\mathrm{cr},m}} = \frac{2000\,\mathrm{psia}}{678.3\,\mathrm{psia}} = 2.949 \end{aligned}\right\} Z_m = 0.97 \qquad \text{(Fig. A-15)}$$

$$m = \frac{P\nu}{Z_m RT} = \frac{(2000\,\mathrm{psia})(1\times10^6\ \mathrm{ft}^3)}{(0.97)(0.5925\,\mathrm{psia}\cdot\mathrm{ft}^3/\mathrm{lbm}\cdot\mathrm{R})(760\,\mathrm{R})} = \mathbf{4.579\times10^6\ lbm}$$

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13-46 The volumetric analysis of a mixture of gases is given. The volumetric and mass flow rates are to be determined using three methods.

Properties The molar masses of O_2, N_2, CO_2, and CH_4 are 32.0, 28.0, 44.0, and 16.0 kg/kmol, respectively (Table A-1).

Analysis (*a*) We consider 100 kmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$m_{O2} = N_{O2}M_{O2} = (30\ \text{kmol})(32\ \text{kg/kmol}) = 960\ \text{kg}$$
$$m_{N2} = N_{N2}M_{N2} = (40\ \text{kmol})(28\ \text{kg/kmol}) = 1120\ \text{kg}$$
$$m_{CO2} = N_{CO2}M_{CO2} = (10\ \text{kmol})(44\ \text{kg/kmol}) = 440\ \text{kg}$$
$$m_{CH4} = N_{CH4}M_{CH4} = (20\ \text{kmol})(16\ \text{kg/kmol}) = 320\ \text{kg}$$

30% O_2
40% N_2
10% CO_2
20% CH_4
(by volume)

The total mass is

$$m_m = m_{O2} + m_{N2} + m_{CO2} + m_{CH4}$$
$$= 960 + 1120 + 440 + 320 = 2840\ \text{kg}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{2840\ \text{kg}}{100\ \text{kmol}} = 28.40\ \text{kg/kmol}$$

Mixture
8 MPa, 15°C

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol}\cdot\text{K}}{28.40\ \text{kg/kmol}} = 0.2927\ \text{kJ/kg}\cdot\text{K}$$

The specific volume of the mixture is

$$v = \frac{RT}{P} = \frac{(0.2927\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{8000\ \text{kPa}} = 0.01054\ \text{m}^3/\text{kg}$$

The volume flow rate is

$$\dot{V} = AV = \frac{\pi D^2}{4}V = \frac{\pi(0.02\ \text{m})^2}{4}(3\ \text{m/s}) = \mathbf{0.0009425\ m^3/s}$$

and the mass flow rate is

$$\dot{m} = \frac{\dot{V}}{v} = \frac{0.0009425\ \text{m}^3/\text{s}}{0.01054\ \text{m}^3/\text{kg}} = \mathbf{0.08942\ kg/s}$$

(*b*) To use the Amagat's law for this real gas mixture, we first need the mole fractions and the *Z* of each component at the mixture temperature and pressure. The compressibility factors are obtained using Fig. A-15 to be

$$\left.\begin{aligned} T_{R,O2} &= \frac{T_m}{T_{cr,O2}} = \frac{288\ \text{K}}{154.8\ \text{K}} = 1.860 \\ P_{R,O2} &= \frac{P_m}{P_{cr,O2}} = \frac{8\ \text{MPa}}{5.08\ \text{MPa}} = 1.575 \end{aligned}\right\} Z_{O2} = 0.95$$

$$\left.\begin{aligned} T_{R,N2} &= \frac{288\ \text{K}}{126.2\ \text{K}} = 2.282 \\ P_{R,N2} &= \frac{8\ \text{MPa}}{3.39\ \text{MPa}} = 2.360 \end{aligned}\right\} Z_{N2} = 0.99$$

$$\left.\begin{aligned} T_{R,CO2} &= \frac{288\ \text{K}}{304.2\ \text{K}} = 0.947 \\ P_{R,CO2} &= \frac{8\ \text{MPa}}{7.39\ \text{MPa}} = 1.083 \end{aligned}\right\} Z_{CO2} = 0.199$$

$$\left.\begin{aligned} T_{R,CH4} &= \frac{288\ \text{K}}{191.1\ \text{K}} = 1.507 \\ P_{R,CH4} &= \frac{8\ \text{MPa}}{4.64\ \text{MPa}} = 1.724 \end{aligned}\right\} Z_{CH4} = 0.85$$

and

$$Z_m = \sum y_i Z_i = y_{O2}Z_{O2} + y_{O2}Z_{O2} + y_{CO2}Z_{CO2} + y_{CH4}Z_{CH4}$$
$$= (0.30)(0.95) + (0.40)(0.99) + (0.10)(0.199) + (0.20)(0.85) = 0.8709$$

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Then,

$$v = \frac{Z_m RT}{P} = \frac{(0.8709)(0.2927\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{8000\ \text{kPa}} = 0.009178\ \text{m}^3/\text{kg}$$

$$\dot{V} = \mathbf{0.0009425\ m^3/s}$$

$$\dot{m} = \frac{\dot{V}}{v} = \frac{0.0009425\ \text{m}^3/\text{s}}{0.009178\ \text{m}^3/\text{kg}} = \mathbf{0.10269\ kg/s}$$

(*c*) To use Kay's rule, we need to determine the pseudo-critical temperature and pseudo-critical pressure of the mixture using the critical point properties of mixture gases.

$$\begin{aligned} T'_{\text{cr},m} &= \sum y_i T_{\text{cr},i} = y_{\text{O2}} T_{\text{cr,O2}} + y_{\text{N2}} T_{\text{cr,N2}} + y_{\text{CO2}} T_{\text{cr,CO2}} + y_{\text{CH4}} T_{\text{cr,CH4}} \\ &= (0.30)(154.8\ \text{K}) + (0.40)(126.2\ \text{K}) + (0.10)(304.2\ \text{K}) + (0.20)(191.1\ \text{K}) = 165.6\ \text{K} \\ P'_{\text{cr},m} &= \sum y_i P_{\text{cr},i} = y_{\text{O2}} P_{\text{cr,O2}} + + y_{\text{N2}} P_{\text{cr,N2}} + y_{\text{CO2}} P_{\text{cr,CO2}} + y_{\text{CH4}} P_{\text{cr,CH4}} \\ &= (0.30)(5.08\ \text{MPa}) + (0.40)(3.39\ \text{MPa}) + (0.10)(7.39\ \text{MPa}) + (0.20)(4.64\ \text{MPa}) = 4.547\ \text{MPa} \end{aligned}$$

and

$$\left.\begin{aligned} T_R &= \frac{T_m}{T'_{\text{cr},m}} = \frac{288\ \text{K}}{165.6\ \text{K}} = 1.739 \\ P_R &= \frac{P_m}{P'_{\text{cr},m}} = \frac{8\ \text{MPa}}{4.547\ \text{MPa}} = 1.759 \end{aligned}\right\} Z_m = 0.92 \qquad \text{(Fig. A-15)}$$

Then,

$$v = \frac{Z_m RT}{P} = \frac{(0.92)(0.2927\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{8000\ \text{kPa}} = 0.009694\ \text{m}^3/\text{kg}$$

$$\dot{V} = \mathbf{0.0009425\ m^3/s}$$

$$\dot{m} = \frac{\dot{V}}{v} = \frac{0.0009425\ \text{m}^3/\text{s}}{0.09694\ \text{m}^3/\text{kg}} = \mathbf{0.009723\ kg/s}$$

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Properties of Gas Mixtures

13-47C Yes. Yes (extensive property).

13-48C No (intensive property).

13-49C The answers are the same for entropy.

13-50C Yes. Yes (conservation of energy).

13-51C We have to use the partial pressure.

13-52C No, this is an approximate approach. It assumes a component behaves as if it existed alone at the mixture temperature and pressure (i.e., it disregards the influence of dissimilar molecules on each other.)

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13-53 The volume fractions of components of a gas mixture are given. This mixture is heated while flowing through a tube at constant pressure. The heat transfer to the mixture per unit mass of the mixture is to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

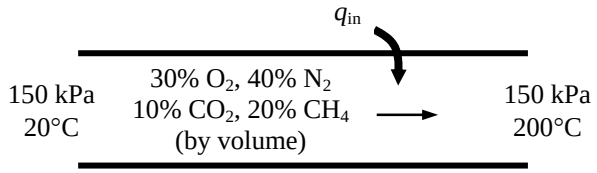
Properties The molar masses of O_2, N_2, CO_2, and CH_4 are 32.0, 28.0, 44.0, and 16.0 kg/kmol, respectively (Table A-1). The constant-pressure specific heats of these gases at room temperature are 0.918, 1.039, 0.846, and 2.2537 kJ/kg·K, respectively (Table A-2a).

Analysis We consider 100 kmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$m_{O2} = N_{O2} M_{O2} = (30\ \text{kmol})(32\ \text{kg/kmol}) = 960\ \text{kg}$$
$$m_{N2} = N_{N2} M_{N2} = (40\ \text{kmol})(28\ \text{kg/kmol}) = 1120\ \text{kg}$$
$$m_{CO2} = N_{CO2} M_{CO2} = (10\ \text{kmol})(44\ \text{kg/kmol}) = 440\ \text{kg}$$
$$m_{CH4} = N_{CH4} M_{CH4} = (20\ \text{kmol})(16\ \text{kg/kmol}) = 320\ \text{kg}$$

The total mass is

$$m_m = m_{O2} + m_{N2} + m_{CO2} + m_{CH4}$$
$$= 960 + 1120 + 440 + 320$$
$$= 2840\ \text{kg}$$



Then the mass fractions are

$$\text{mf}_{O2} = \frac{m_{O2}}{m_m} = \frac{960\ \text{kg}}{2840\ \text{kg}} = 0.3380$$
$$\text{mf}_{N2} = \frac{m_{N2}}{m_m} = \frac{1120\ \text{kg}}{2840\ \text{kg}} = 0.3944$$
$$\text{mf}_{CO2} = \frac{m_{CO2}}{m_m} = \frac{440\ \text{kg}}{2840\ \text{kg}} = 0.1549$$
$$\text{mf}_{CH4} = \frac{m_{CH4}}{m_m} = \frac{320\ \text{kg}}{2840\ \text{kg}} = 0.1127$$

The constant-pressure specific heat of the mixture is determined from

$$c_p = \text{mf}_{O2} c_{p,O2} + \text{mf}_{N2} c_{p,N2} + \text{mf}_{CO2} c_{p,CO2} + \text{mf}_{CH4} c_{p,CH4}$$
$$= 0.3380 \times 0.918 + 0.3944 \times 1.039 + 0.1549 \times 0.846 + 0.1127 \times 2.2537$$
$$= 1.1051\ \text{kJ/kg} \cdot \text{K}$$

An energy balance on the tube gives

$$q_{in} = c_p (T_2 - T_1) = (1.1051\ \text{kJ/kg} \cdot \text{K})(200 - 20)\ \text{K} = \mathbf{199\ kJ/kg}$$

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13-54E A mixture of helium and nitrogen is heated at constant pressure in a closed system. The work produced is to be determined.

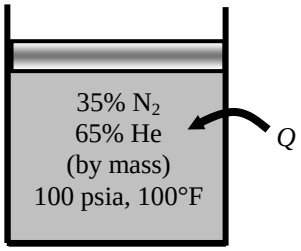
Assumptions **1** Helium and nitrogen are ideal gases. **2** The process is reversible.

Properties The mole numbers of helium and nitrogen are 4.0 and 28.0 lbm/lbmol, respectively (Table A-1E).

Analysis One lbm of this mixture consists of 0.35 lbm of nitrogen and 0.65 lbm of helium or 0.35 lbm/(28.0 lbm/lbmol) = 0.0125 lbmol of nitrogen and 0.65 lbm/(4.0 lbm/lbmol) = 0.1625 lbmol of helium. The total mole is 0.0125+0.1625=0.175 lbmol. The constituent mole fraction are then

$$y_{N2} = \frac{N_{N2}}{N_{total}} = \frac{0.0125\,\text{lbmol}}{0.175\,\text{lbmol}} = 0.07143$$

$$y_{He} = \frac{N_{He}}{N_{total}} = \frac{0.1625\,\text{lbmol}}{0.175\,\text{lbmol}} = 0.9286$$



The effective molecular weight of this mixture is

$$\begin{aligned} M &= y_{N2} M_{N2} + y_{He} M_{He} \\ &= (0.07143)(28) + (0.9286)(4) \\ &= 5.714\,\text{lbm/lbmol} \end{aligned}$$

The work done is determined from

$$w = \int_1^2 P\,dv = P_2 v_2 - P_1 v_1 = R(T_2 - T_1)$$

$$= \frac{R_u}{M}(T_2 - T_1) = \frac{1.9858\,\text{Btu/lbmol}\cdot\text{R}}{5.714\,\text{lbm/lbmol}}(500 - 100)\text{R} = \mathbf{139.0\ Btu/lbm}$$

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13-55 The volume fractions of components of a gas mixture are given. This mixture is expanded isentropically to a specified pressure. The work produced per unit mass of the mixture is to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

Properties The molar masses of H_2, He, and N_2 are 2.0, 4.0, and 28.0 kg/kmol, respectively (Table A-1). The constant-pressure specific heats of these gases at room temperature are 14.307, 5.1926, and 1.039 kJ/kg·K, respectively (Table A-2a).

Analysis We consider 100 kmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$m_{H2} = N_{H2} M_{H2} = (30\ \text{kmol})(2\ \text{kg/kmol}) = 60\ \text{kg}$$
$$m_{He} = N_{He} M_{He} = (40\ \text{kmol})(4\ \text{kg/kmol}) = 160\ \text{kg}$$
$$m_{N2} = N_{N2} M_{N2} = (30\ \text{kmol})(28\ \text{kg/kmol}) = 840\ \text{kg}$$

30% H_2
40% He
30% N_2
(by volume)
5 MPa, 600°C

The total mass is

$$m_m = m_{H2} + m_{He} + m_{N2} = 60 + 160 + 840 = 1060\ \text{kg}$$

Then the mass fractions are

$$\text{mf}_{H2} = \frac{m_{H2}}{m_m} = \frac{60\ \text{kg}}{1060\ \text{kg}} = 0.05660$$
$$\text{mf}_{He} = \frac{m_{He}}{m_m} = \frac{160\ \text{kg}}{1060\ \text{kg}} = 0.1509$$
$$\text{mf}_{N2} = \frac{m_{N2}}{m_m} = \frac{840\ \text{kg}}{1060\ \text{kg}} = 0.7925$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{1060\ \text{kg}}{100\ \text{kmol}} = 10.60\ \text{kg/kmol}$$

The constant-pressure specific heat of the mixture is determined from

$$\begin{aligned} c_p &= \text{mf}_{H2} c_{p,H2} + \text{mf}_{He} c_{p,He} + \text{mf}_{N2} c_{p,N2} \\ &= 0.05660 \times 14.307 + 0.1509 \times 5.1926 + 0.7925 \times 1.039 \\ &= 2.417\ \text{kJ/kg} \cdot \text{K} \end{aligned}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol} \cdot \text{K}}{10.60\ \text{kg/kmol}} = 0.7843\ \text{kJ/kg} \cdot \text{K}$$

Then the constant-volume specific heat is

$$c_v = c_p - R = 2.417 - 0.7843 = 1.633\ \text{kJ/kg} \cdot \text{K}$$

The specific heat ratio is

$$k = \frac{c_p}{c_v} = \frac{2.417}{1.633} = 1.480$$

The temperature at the end of the expansion is

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (873\ \text{K}) \left(\frac{200\ \text{kPa}}{5000\ \text{kPa}} \right)^{0.48/1.48} = 307\ \text{K}$$

An energy balance on the adiabatic expansion process gives

$$w_{out} = c_p (T_1 - T_2) = (2.417\ \text{kJ/kg} \cdot \text{K})(873 - 307)\ \text{K} = \mathbf{1368\ kJ/kg}$$

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13-56 The mass fractions of components of a gas mixture are given. This mixture is enclosed in a rigid, well-insulated vessel, and a paddle wheel in the vessel is turned until specified amount of work have been done on the mixture. The mixture's final pressure and temperature are to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

Properties The molar masses of N_2, He, CH_4, and C_2H_6 are 28.0, 4.0, 16.0, and 30.0 kg/kmol, respectively (Table A-1). The constant-pressure specific heats of these gases at room temperature are 1.039, 5.1926, 2.2537, and 1.7662 kJ/kg·K, respectively (Table A-2a).

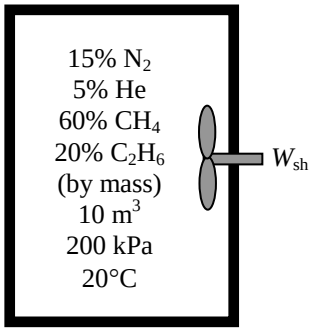
Analysis We consider 100 kg of this mixture. The mole numbers of each component are

$$N_{\text{N2}} = \frac{m_{\text{N2}}}{M_{\text{N2}}} = \frac{15\text{ kg}}{28\text{ kg/kmol}} = 0.5357\text{ kmol}$$

$$N_{\text{He}} = \frac{m_{\text{He}}}{M_{\text{He}}} = \frac{5\text{ kg}}{4\text{ kg/kmol}} = 1.25\text{ kmol}$$

$$N_{\text{CH4}} = \frac{m_{\text{CH4}}}{M_{\text{CH4}}} = \frac{60\text{ kg}}{16\text{ kg/kmol}} = 3.75\text{ kmol}$$

$$N_{\text{C2H6}} = \frac{m_{\text{C2H6}}}{M_{\text{C2H6}}} = \frac{20\text{ kg}}{30\text{ kg/kmol}} = 0.6667\text{ kmol}$$



The mole number of the mixture is

$$N_m = N_{\text{N2}} + N_{\text{He}} + N_{\text{CH4}} + N_{\text{C2H6}} = 0.5357 + 1.25 + 3.75 + 0.6667 = 6.2024\text{ kmol}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{100\text{ kg}}{6.2024\text{ kmol}} = 16.12\text{ kg/kmol}$$

The constant-pressure specific heat of the mixture is determined from

$$\begin{aligned} c_p &= \text{mf}_{\text{N2}} c_{p,\text{N2}} + \text{mf}_{\text{He}} c_{p,\text{He}} + \text{mf}_{\text{CH4}} c_{p,\text{CH4}} + \text{mf}_{\text{C2H6}} c_{p,\text{C2H6}} \\ &= 0.15 \times 1.039 + 0.05 \times 5.1926 + 0.60 \times 2.2537 + 0.20 \times 1.7662 \\ &= 2.121\text{ kJ/kg} \cdot \text{K} \end{aligned}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.134\text{ kJ/kmol} \cdot \text{K}}{16.12\text{ kg/kmol}} = 0.5158\text{ kJ/kg} \cdot \text{K}$$

Then the constant-volume specific heat is

$$c_v = c_p - R = 2.121 - 0.5158 = 1.605\text{ kJ/kg} \cdot \text{K}$$

The mass in the container is

$$m_m = \frac{P_1 V_m}{RT_1} = \frac{(200\text{ kPa})(10\text{ m}^3)}{(0.5158\text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293\text{ K})} = 13.23\text{ kg}$$

An energy balance on the system gives

$$W_{\text{sh,in}} = m_m c_v (T_2 - T_1) \longrightarrow T_2 = T_1 + \frac{W_{\text{sh,in}}}{m_m c_v} = (293\text{ K}) + \frac{100\text{ kJ}}{(13.23\text{ kg})(1.605\text{ kJ/kg} \cdot \text{K})} = \mathbf{297.7\text{ K}}$$

Since the volume remains constant and this is an ideal gas,

$$P_2 = P_1 \frac{T_2}{T_1} = (200\text{ kPa}) \frac{297.7\text{ K}}{293\text{ K}} = \mathbf{203.2\text{ kPa}}$$

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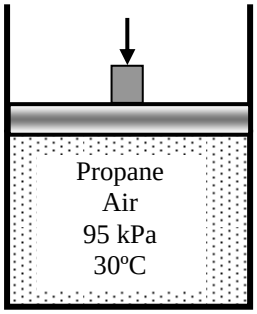
13-57 Propane and air mixture is compressed isentropically in an internal combustion engine. The work input is to be determined.

Assumptions Under specified conditions propane and air can be treated as ideal gases, and the mixture as an ideal gas mixture.

Properties The molar masses of C_3H_8 and air are 44.0 and 28.97 kg/kmol, respectively (TableA-1).

Analysis Given the air-fuel ratio, the mass fractions are determined to be

$$\text{mf}_{\text{air}} = \frac{\text{AF}}{\text{AF}+1} = \frac{16}{17} = 0.9412$$
$$\text{mf}_{C_3H_8} = \frac{1}{\text{AF}+1} = \frac{1}{17} = 0.05882$$



The molar mass of the mixture is determined to be

$$M_m = \frac{1}{\dfrac{\text{mf}_{\text{air}}}{M_{\text{air}}} + \dfrac{\text{mf}_{C_3H_8}}{M_{C_3H_8}}} = \frac{1}{\dfrac{0.9412}{28.97\ \text{kg/kmol}} + \dfrac{0.05882}{44.0\ \text{kg/kmol}}} = 29.56\ \text{kg/kmol}$$

The mole fractions are

$$y_{\text{air}} = \text{mf}_{\text{air}} \frac{M_m}{M_{\text{air}}} = (0.9412)\frac{29.56\ \text{kg/kmol}}{28.97\ \text{kg/kmol}} = 0.9606$$
$$y_{C_3H_8} = \text{mf}_{C_3H_8} \frac{M_m}{M_{C_3H_8}} = (0.05882)\frac{29.56\ \text{kg/kmol}}{44.0\ \text{kg/kmol}} = 0.03944$$

The final pressure is expressed from ideal gas relation to be

$$P_2 = P_1 r \frac{T_2}{T_1} = (95\ \text{kPa})(9.5)\frac{T_2}{(30+273.15)\ \text{K}} = 2.977T_2 \qquad (1)$$

since the final temperature is not known. Using Dalton's law to find partial pressures, the entropies at the initial state are determined from EES to be:

$$T = 30°\text{C},\ P = (0.9606\times 95) = 91.26\ \text{kPa} \longrightarrow s_{\text{air},1} = 5.7417\ \text{kJ/kg.K}$$
$$T = 30°\text{C},\ P = (0.03944\times 95) = 3.75\ \text{kPa} \longrightarrow s_{C_3H_8,1} = 6.7697\ \text{kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final pressure and temperature are not known. However, for an isentropic process, the entropy change is zero and the final temperature and the final pressure may be determined from

$$\Delta s_{\text{total}} = mf_{\text{air}}\Delta s_{\text{air}} + mf_{C_3H_8}\Delta s_{C_3H_8} = 0$$

and using Eq. (1). The solution may be obtained using EES to be

$$T_2 = 654.9\ \text{K},\ P_2 = 1951\ \text{kPa}$$

The initial and final internal energies are (from EES)

$$T_1 = 30°\text{C} \longrightarrow \begin{matrix} u_{\text{air},1} = 216.5\ \text{kJ/kg} \\ u_{C_3H_8,1} = -2404\ \text{kJ/kg} \end{matrix} \qquad T_2 = 654.9\ \text{K} \longrightarrow \begin{matrix} u_{\text{air},2} = 477.1\ \text{kJ/kg} \\ u_{C_3H_8,2} = -1607\ \text{kJ/kg} \end{matrix}$$

Noting that the heat transfer is zero, an energy balance on the system gives

$$q_{\text{in}} + w_{\text{in}} = \Delta u_m \longrightarrow w_{\text{in}} = \Delta u_m$$

where $\Delta u_m = mf_{\text{air}}(u_{\text{air},2} - u_{\text{air},1}) + mf_{C_3H_8}(u_{C_3H_8,2} - u_{C_3H_8,1})$

Substituting, $w_{\text{in}} = \Delta u_m = (0.9412)(477.1 - 216.5) + (0.05882)\big[(-1607) - (-2404)\big] = \mathbf{292.2\ kJ/kg}$

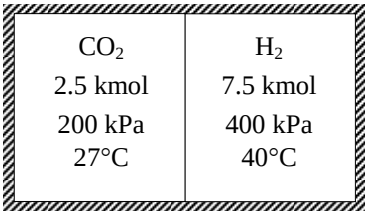
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13-58 The moles, temperatures, and pressures of two gases forming a mixture are given. The mixture temperature and pressure are to be determined.

Assumptions **1** Under specified conditions both CO_2 and H_2 can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The tank is insulated and thus there is no heat transfer. **3** There are no other forms of work involved.

Properties The molar masses and specific heats of CO_2 and H_2 are 44.0 kg/kmol, 2.0 kg/kmol, 0.657 kJ/kg.°C, and 10.183 kJ/kg.°C, respectively. (Tables A-1 and A-2b).

Analysis (*a*) We take both gases as our system. No heat, work, or mass crosses the system boundary, therefore this is a closed system with $Q = 0$ and $W = 0$. Then the energy balance for this closed system reduces to



$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$
$$0 = \Delta U = \Delta U_{CO_2} + \Delta U_{H_2}$$
$$0 = [mc_v(T_m - T_1)]_{CO_2} + [mc_v(T_m - T_1)]_{H_2}$$

Using c_v values at room temperature and noting that $m = NM$, the final temperature of the mixture is determined to be

$$(2.5 \times 44 \text{ kg})(0.657 \text{ kJ/kg} \cdot °\text{C})(T_m - 27°\text{C}) + (7.5 \times 2 \text{ kg})(10.183 \text{ kJ/kg} \cdot °\text{C})(T_m - 40°\text{C}) = 0$$
$$T_m = \mathbf{35.8°C} \quad (308.8 \text{ K})$$

(*b*) The volume of each tank is determined from

$$V_{CO_2} = \left(\frac{NR_uT_1}{P_1}\right)_{CO_2} = \frac{(2.5 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(300 \text{ K})}{200 \text{ kPa}} = 31.18 \text{ m}^3$$
$$V_{H_2} = \left(\frac{NR_uT_1}{P_1}\right)_{H_2} = \frac{(7.5 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(313 \text{ K})}{400 \text{ kPa}} = 48.79 \text{ m}^3$$

Thus,

$$V_m = V_{CO_2} + V_{H_2} = 31.18 \text{ m}^3 + 48.79 \text{ m}^3 = 79.97 \text{ m}^3$$
$$N_m = N_{CO_2} + N_{H_2} = 2.5 \text{ kmol} + 7.5 \text{ kmol} = 10.0 \text{ kmol}$$

and

$$P_m = \frac{N_m R_u T_m}{V_m} = \frac{(10.0 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(308.8 \text{ K})}{79.97 \text{ m}^3} = \mathbf{321 \text{ kPa}}$$

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13-59 The mass fractions of components of a gas mixture are given. This mixture is compressed in a reversible, isothermal, steady-flow compressor. The work and heat transfer for this compression per unit mass of the mixture are to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

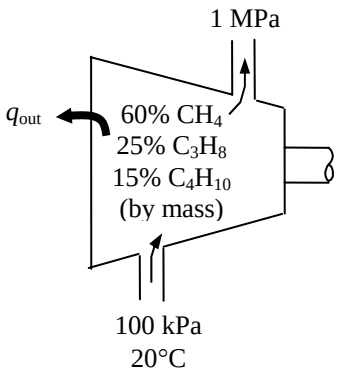
Properties The molar masses of CH_4, C_3H_8, and C_4H_{10} are 16.0, 44.0, and 58.0 kg/kmol, respectively (Table A-1).

Analysis The mole numbers of each component are

$$N_{CH4} = \frac{m_{CH4}}{M_{CH4}} = \frac{60\text{ kg}}{16\text{ kg/kmol}} = 3.75\text{ kmol}$$

$$N_{C3H8} = \frac{m_{C3H8}}{M_{C3H8}} = \frac{25\text{ kg}}{44\text{ kg/kmol}} = 0.5682\text{ kmol}$$

$$N_{C4H10} = \frac{m_{C4H10}}{M_{C4H10}} = \frac{15\text{ kg}}{58\text{ kg/kmol}} = 0.2586\text{ kmol}$$



The mole number of the mixture is

$$N_m = N_{CH4} + N_{C3H8} + N_{C4H10}$$
$$= 3.75 + 0.5682 + 0.2586 = 4.5768\text{ kmol}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{100\text{ kg}}{4.5768\text{ kmol}} = 21.85\text{ kg/kmol}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\text{ kJ/kmol}\cdot\text{K}}{21.85\text{ kg/kmol}} = 0.3805\text{ kJ/kg}\cdot\text{K}$$

For a reversible, isothermal process, the work input is

$$w_{in} = RT\ln\left(\frac{P_2}{P_1}\right) = (0.3805\text{ kJ/kg}\cdot\text{K})(293\text{ K})\ln\left(\frac{1000\text{ kPa}}{100\text{ kPa}}\right) = \mathbf{257\text{ kJ/kg}}$$

An energy balance on the control volume gives

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}}^{\nearrow 0\ \text{(steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}h_1 + \dot{W}_{in} = \dot{m}h_2 + \dot{Q}_{out}$$
$$\dot{W}_{in} - \dot{Q}_{out} = \dot{m}(h_2 - h_1)$$
$$w_{in} - q_{out} = c_p(T_2 - T_1) = 0 \quad \text{since} \quad T_2 = T_1$$
$$w_{in} = q_{out}$$

That is,

$$q_{out} = w_{in} = \mathbf{257\text{ kJ/kg}}$$

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13-60 The volume fractions of components of a gas mixture during the expansion process of the ideal Otto cycle are given. The thermal efficiency of this cycle is to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

Properties The molar masses of N_2, O_2, H_2O, and CO_2 are 28.0, 32.0, 18.0, and 44.0 kg/kmol, respectively (Table A-1). The constant-pressure specific heats of these gases at room temperature are 1.039, 0.918, 1.8723, and 0.846 kJ/kg·K, respectively. The air properties at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, k = 1.4 (Table A-2a).

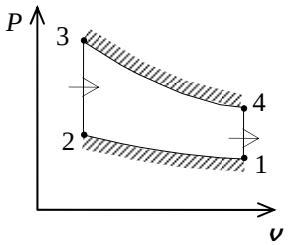
Analysis We consider 100 kmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$m_{N2} = N_{N2}M_{N2} = (30\ \text{kmol})(28\ \text{kg/kmol}) = 840\ \text{kg}$$
$$m_{O2} = N_{O2}M_{O2} = (10\ \text{kmol})(32\ \text{kg/kmol}) = 320\ \text{kg}$$
$$m_{H2O} = N_{H2O}M_{H2O} = (35\ \text{kmol})(18\ \text{kg/kmol}) = 630\ \text{kg}$$
$$m_{CO2} = N_{CO2}M_{CO2} = (25\ \text{kmol})(44\ \text{kg/kmol}) = 1100\ \text{kg}$$

30% N_2
10% O_2
35% H_2O
25% CO_2
(by volume)

The total mass is

$$m_m = m_{N2} + m_{O2} + m_{H2O} + m_{CO2}$$
$$= 840 + 320 + 630 + 1100$$
$$= 2890\ \text{kg}$$



Then the mass fractions are

$$\text{mf}_{N2} = \frac{m_{N2}}{m_m} = \frac{840\ \text{kg}}{2890\ \text{kg}} = 0.2907$$
$$\text{mf}_{O2} = \frac{m_{O2}}{m_m} = \frac{320\ \text{kg}}{2890\ \text{kg}} = 0.1107$$
$$\text{mf}_{H2O} = \frac{m_{H2O}}{m_m} = \frac{630\ \text{kg}}{2890\ \text{kg}} = 0.2180$$
$$\text{mf}_{CO2} = \frac{m_{CO2}}{m_m} = \frac{1100\ \text{kg}}{2890\ \text{kg}} = 0.3806$$

The constant-pressure specific heat of the mixture is determined from

$$c_p = \text{mf}_{N2}c_{p,N2} + \text{mf}_{O2}c_{p,O2} + \text{mf}_{H2O}c_{p,H2O} + \text{mf}_{CO2}c_{p,CO2}$$
$$= 0.2907 \times 1.039 + 0.1107 \times 0.918 + 0.2180 \times 1.8723 + 0.3806 \times 0.846$$
$$= 1.134\ \text{kJ/kg} \cdot \text{K}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{2890\ \text{kg}}{100\ \text{kmol}} = 28.90\ \text{kg/kmol}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol} \cdot \text{K}}{28.90\ \text{kg/kmol}} = 0.2877\ \text{kJ/kg} \cdot \text{K}$$

Then the constant-volume specific heat is

$$c_v = c_p - R = 1.134 - 0.2877 = 0.846\ \text{kJ/kg} \cdot \text{K}$$

The specific heat ratio is

$$k = \frac{c_p}{c_v} = \frac{1.134}{0.846} = 1.340$$

The average of the air properties at room temperature and combustion gas properties are

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$$c_{p,\text{avg}} = 0.5(1.134 + 1.005) = 1.070 \text{ kJ/kg} \cdot \text{K}$$
$$c_{v,\text{avg}} = 0.5(0.846 + 0.718) = 0.782 \text{ kJ/kg} \cdot \text{K}$$
$$k_{\text{avg}} = 0.5(1.34 + 1.4) = 1.37$$

These average properties will be used for heat addition and rejection processes. For compression, the air properties at room temperature and during expansion, the mixture properties will be used. During the compression process,

$$T_2 = T_1 r^{k-1} = (288 \text{ K})(8)^{0.4} = 662 \text{ K}$$

During the heat addition process,

$$q_{\text{in}} = c_{v,\text{avg}}(T_3 - T_2) = (0.782 \text{ kJ/kg} \cdot \text{K})(1373 - 662) \text{ K} = 556 \text{ kJ/kg}$$

During the expansion process,

$$T_4 = T_3\left(\frac{1}{r}\right)^{k-1} = (1373 \text{ K})\left(\frac{1}{8}\right)^{0.37} = 636.1 \text{ K}$$

During the heat rejection process,

$$q_{\text{out}} = c_{v,\text{avg}}(T_4 - T_1) = (0.782 \text{ kJ/kg} \cdot \text{K})(636.1 - 288) \text{ K} = 272.2 \text{ kJ/kg}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{272.2 \text{ kJ/kg}}{556 \text{ kJ/kg}} = \mathbf{0.511}$$

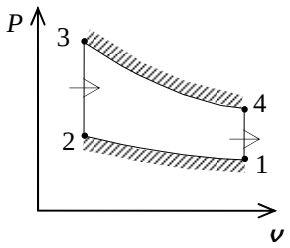
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13-61 The thermal efficiency of the cycle in the previous problem is to be compared to that predicted by air standard analysis.

Assumptions Air-standard assumptions are applicable.

Properties The air properties at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, k = 1.4 (Table A-2a).

Analysis In the previous problem, the thermal efficiency of the cycle was determined to be **0.511** (51.1%). The thermal efficiency with air-standard model is determined from



$$\eta_{th} = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{8^{0.4}} = \mathbf{0.565}$$

which is greater than that calculated with gas mixture analysis in the previous problem.

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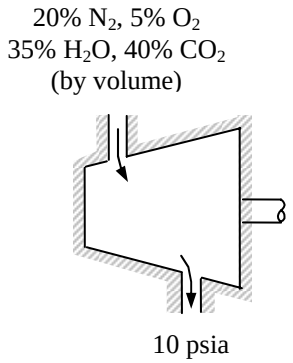
13-62E The volume fractions of components of a gas mixture passing through the turbine of a simple ideal Brayton cycle are given. The thermal efficiency of this cycle is to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

Properties The molar masses of N_2, O_2, H_2O, and CO_2 are 28.0, 32.0, 18.0, and 44.0 lbm/lbmol, respectively (Table A-1E). The constant-pressure specific heats of these gases at room temperature are 0.248, 0.219, 0.445, and 0.203 Btu/lbm·R, respectively. The air properties at room temperature are c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, k = 1.4 (Table A-2Ea).

Analysis We consider 100 lbmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$m_{N2} = N_{N2}M_{N2} = (20\,\text{lbmol})(28\,\text{lbm/lbmol}) = 560\,\text{lbm}$$
$$m_{O2} = N_{O2}M_{O2} = (5\,\text{lbmol})(32\,\text{lbm/lbmol}) = 160\,\text{lbm}$$
$$m_{H2O} = N_{H2O}M_{H2O} = (35\,\text{lbmol})(18\,\text{lbm/lbmol}) = 630\,\text{lbm}$$
$$m_{CO2} = N_{CO2}M_{CO2} = (40\,\text{lbmol})(44\,\text{lbm/lbmol}) = 1760\,\text{lbm}$$

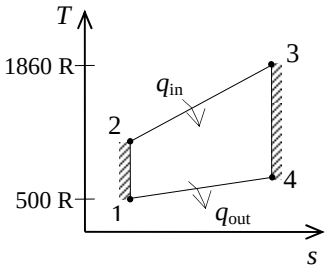


The total mass is

$$m_m = m_{N2} + m_{O2} + m_{H2O} + m_{CO2}$$
$$= 560 + 160 + 630 + 1760$$
$$= 3110\,\text{lbm}$$

Then the mass fractions are

$$\text{mf}_{N2} = \frac{m_{N2}}{m_m} = \frac{560\,\text{lbm}}{3110\,\text{lbm}} = 0.1801$$
$$\text{mf}_{O2} = \frac{m_{O2}}{m_m} = \frac{160\,\text{lbm}}{3110\,\text{lbm}} = 0.05145$$
$$\text{mf}_{H2O} = \frac{m_{H2O}}{m_m} = \frac{630\,\text{lbm}}{3110\,\text{lbm}} = 0.2026$$
$$\text{mf}_{CO2} = \frac{m_{CO2}}{m_m} = \frac{1760\,\text{lbm}}{3110\,\text{lbm}} = 0.5659$$



The constant-pressure specific heat of the mixture is determined from

$$c_p = \text{mf}_{N2}c_{p,N2} + \text{mf}_{O2}c_{p,O2} + \text{mf}_{H2O}c_{p,H2O} + \text{mf}_{CO2}c_{p,CO2}$$
$$= 0.1801 \times 0.248 + 0.05145 \times 0.219 + 0.2026 \times 0.445 + 0.5659 \times 0.203$$
$$= 0.2610\,\text{Btu/lbm} \cdot \text{R}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{3110\,\text{lbm}}{100\,\text{lbmol}} = 31.10\,\text{lbm/lbmol}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{1.9858\,\text{Btu/lbmol} \cdot \text{R}}{31.10\,\text{lbm/lbmol}} = 0.06385\,\text{Btu/lbm} \cdot \text{R}$$

Then the constant-volume specific heat is

$$c_v = c_p - R = 0.2610 - 0.06385 = 0.1971\,\text{Btu/lbm} \cdot \text{R}$$

The specific heat ratio is

$$k = \frac{c_p}{c_v} = \frac{0.2610}{0.1971} = 1.324$$

The average of the air properties at room temperature and combustion gas properties are

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$$c_{p,\text{avg}} = 0.5(0.2610 + 0.240) = 0.2505 \text{ Btu/lbm} \cdot \text{R}$$
$$c_{v,\text{avg}} = 0.5(0.1971 + 0.171) = 0.1841 \text{ Btu/lbm} \cdot \text{R}$$
$$k_{\text{avg}} = 0.5(1.324 + 1.4) = 1.362$$

These average properties will be used for heat addition and rejection processes. For compression, the air properties at room temperature and during expansion, the mixture properties will be used. During the compression process,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (500 \text{ R})(6)^{0.4/1.4} = 834.3 \text{ R}$$

During the heat addition process,

$$q_{\text{in}} = c_{p,\text{avg}}(T_3 - T_2) = (0.2505 \text{ Btu/lbm} \cdot \text{R})(1860 - 834.3) \text{ R} = 256.9 \text{ Btu/lbm}$$

During the expansion process,

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1860 \text{ R})\left(\frac{1}{6}\right)^{0.362/1.362} = 1155.3 \text{ R}$$

During the heat rejection process,

$$q_{\text{out}} = c_{p,\text{avg}}(T_4 - T_1) = (0.2505 \text{ Btu/lbm} \cdot \text{R})(1155.3 - 500) \text{ R} = 164.2 \text{ Btu/lbm}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{164.2 \text{ Btu/lbm}}{256.9 \text{ Btu/lbm}} = 0.361 = \mathbf{36.1\%}$$

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13-63E The thermal efficiency of the cycle in the previous problem is to be compared to that predicted by air standard analysis?

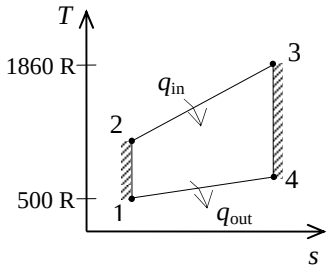
Assumptions Air-standard assumptions are applicable.

Properties The air properties at room temperature are c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, k = 1.4 (Table A-2Ea).

Analysis In the previous problem, the thermal efficiency of the cycle was determined to be **0.361** (36.1%). The thermal efficiency with air-standard model is determined from

$$\eta_{\text{th}} = 1 - \frac{1}{r_p^{(k-1)/k}} = 1 - \frac{1}{6^{0.4/1.4}} = 0.401 = \mathbf{40.1\%}$$

which is greater than that calculated with gas mixture analysis in the previous problem.



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13-64E The mass fractions of a natural gas mixture at a specified pressure and temperature trapped in a geological location are given. This natural gas is pumped to the surface. The work required is to be determined using Kay's rule and the enthalpy-departure method.

Properties The molar masses of CH_4 and C_2H_6 are 16.0 and 30.0 lbm/lbmol, respectively. The critical properties are 343.9 R, 673 psia for CH_4 and 549.8 R and 708 psia for C_2H_6 (Table A-1E). The constant-pressure specific heats of these gases at room temperature are 0.532 and 0.427 Btu/lbm·R, respectively (Table A-2Ea).

Analysis We consider 100 lbm of this mixture. Then the mole numbers of each component are

$$N_{\text{CH4}} = \frac{m_{\text{CH4}}}{M_{\text{CH4}}} = \frac{75\,\text{lbm}}{16\,\text{lbm/lbmol}} = 4.6875\,\text{lbmol}$$

$$N_{\text{C2H6}} = \frac{m_{\text{C2H6}}}{M_{\text{C2H6}}} = \frac{25\,\text{lbm}}{30\,\text{lbm/lbmol}} = 0.8333\,\text{lbmol}$$

75% CH_4
25% C_2H_6
(by mass)
2000 psia
300°F

The mole number of the mixture and the mole fractions are

$$N_m = 4.6875 + 0.8333 = 5.5208\,\text{lbmol}$$

$$y_{\text{CH4}} = \frac{N_{\text{CH4}}}{N_m} = \frac{4.6875\,\text{lbmol}}{5.5208\,\text{lbmol}} = 0.8491$$

$$y_{\text{C2H6}} = \frac{N_{\text{C2H6}}}{N_m} = \frac{0.8333\,\text{lbmol}}{5.5208\,\text{lbmol}} = 0.1509$$

Then the apparent molecular weight of the mixture becomes

$$M_m = \frac{m_m}{N_m} = \frac{100\,\text{lbm}}{5.5208\,\text{lbmol}} = 18.11\,\text{lbm/lbmol}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{1.9858\,\text{Btu/lbmol}\cdot\text{R}}{18.11\,\text{lbm/lbmol}} = 0.1097\,\text{Btu/lbm}\cdot\text{R}$$

The constant-pressure specific heat of the mixture is determined from

$$c_p = \text{mf}_{\text{CH4}} c_{p,\text{CH4}} + \text{mf}_{\text{C2H6}} c_{p,\text{C2H6}} = 0.75\times 0.532 + 0.25\times 0.427 = 0.506\,\text{Btu/lbm}\cdot\text{R}$$

To use Kay's rule, we need to determine the pseudo-critical temperature and pseudo-critical pressure of the mixture using the critical point properties of gases.

$$T'_{\text{cr},m} = \sum y_i T_{\text{cr},i} = y_{\text{CH4}} T_{\text{cr,Ch4}} + y_{\text{C2H6}} T_{\text{cr,C2H6}}$$
$$= (0.8491)(343.9\,\text{R}) + (0.1509)(549.8\,\text{R}) = 375.0\,\text{R}$$

$$P'_{\text{cr},m} = \sum y_i P_{\text{cr},i} = y_{\text{Ch4}} P_{\text{cr,Ch4}} + y_{\text{C2H6}} P_{\text{cr,C2H6}}$$
$$= (0.8491)(673\,\text{psia}) + (0.1509)(708\,\text{psia}) = 678.3\,\text{psia}$$

The compressibility factor of the gas mixture in the reservoir and the mass of this gas are

$$\left.\begin{aligned} T_R &= \frac{T_m}{T'_{\text{cr},m}} = \frac{760\,\text{R}}{375.0\,\text{R}} = 2.027 \\ P_R &= \frac{P_m}{P'_{\text{cr},m}} = \frac{2000\,\text{psia}}{678.3\,\text{psia}} = 2.949 \end{aligned}\right\} Z_m = 0.963 \qquad \text{(Fig. A-15)}$$

$$m = \frac{P\mathcal{V}}{Z_m RT} = \frac{(2000\,\text{psia})(1\times 10^6\,\text{ft}^3)}{(0.963)(0.5925\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(760\,\text{R})} = 4.612\times 10^6\,\text{lbm}$$

The enthalpy departure factors in the reservoir and the surface are (from EES or Fig. A-29)

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$$\left.\begin{aligned} T_{R1} &= \frac{T_m}{T'_{\text{cr},m}} = \frac{760\text{ R}}{375.0\text{ R}} = 2.027 \\ P_{R1} &= \frac{P_m}{P'_{\text{cr},m}} = \frac{2000\text{ psia}}{678.3\text{ psia}} = 2.949 \end{aligned}\right\} Z_{h1} = 0.703$$

$$\left.\begin{aligned} T_{R2} &= \frac{T_m}{T'_{\text{cr},m}} = \frac{660\text{ R}}{375.0\text{ R}} = 1.76 \\ P_{R2} &= \frac{P_m}{P'_{\text{cr},m}} = \frac{20\text{ psia}}{678.3\text{ psia}} = 0.0295 \end{aligned}\right\} Z_{h2} = 0.0112$$

The enthalpy change for the ideal gas mixture is

$$(h_1 - h_2)_{\text{ideal}} = c_p(T_1 - T_2) = (0.506\text{ Btu/lbm}\cdot\text{R})(760 - 660)\text{R} = 50.6\text{ Btu/lbm}$$

The enthalpy change with departure factors is

$$\begin{aligned} h_1 - h_2 &= (h_1 - h_2)_{\text{ideal}} - RT'_{\text{cr},m}(Z_{h1} - Z_{h2}) \\ &= 50.6 - (0.1096)(375)(0.703 - 0.0112) \\ &= 22.12\text{ Btu/lbm} \end{aligned}$$

The work input is then

$$W_{\text{in}} = m(h_1 - h_2) = (4.612\times10^6\text{ lbm})(22.12\text{ Btu/lbm}) = \mathbf{1.02\times10^8\ Btu}$$

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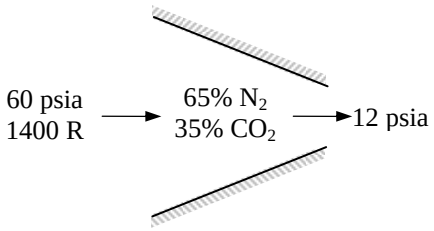
13-65E A gas mixture with known mass fractions is accelerated through a nozzle from a specified state to a specified pressure. For a specified isentropic efficiency, the exit temperature and the exit velocity of the mixture are to be determined.

Assumptions **1** Under specified conditions both N_2 and CO_2 can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The nozzle is adiabatic and thus heat transfer is negligible. **3** This is a steady-flow process. **4** Potential energy changes are negligible.

Properties The specific heats of N_2 and CO_2 are $c_{p,N2}$ = 0.248 Btu/lbm.R, $c_{v,N2}$ = 0.177 Btu/lbm.R, $c_{p,CO2}$ = 0.203 Btu/lbm.R, and $c_{v,CO2}$ = 0.158 Btu/lbm.R. (Table A-2E).

Analysis (*a*) Under specified conditions both N_2 and CO_2 can be treated as ideal gases, and the mixture as an ideal gas mixture. The c_p, c_v, and k values of this mixture are determined from

$$\begin{aligned} c_{p,m} &= \sum \mathrm{mf}_i c_{p,i} = \mathrm{mf}_{\mathrm{N_2}} c_{p,\mathrm{N_2}} + \mathrm{mf}_{\mathrm{CO_2}} c_{p,\mathrm{CO_2}} \\ &= (0.65)(0.248) + (0.35)(0.203) \\ &= 0.2323 \text{ Btu/lbm} \cdot \text{R} \\ c_{v,m} &= \sum \mathrm{mf}_i c_{v,i} = \mathrm{mf}_{\mathrm{N_2}} c_{v,\mathrm{N_2}} + \mathrm{mf}_{\mathrm{CO_2}} c_{v,\mathrm{CO_2}} \\ &= (0.65)(0.177) + (0.35)(0.158) \\ &= 0.1704 \text{ Btu/lbm} \cdot \text{R} \\ k_m &= \frac{c_{p,m}}{c_{v,m}} = \frac{0.2323 \text{ Btu/lbm} \cdot \text{R}}{0.1704 \text{ Btu/lbm} \cdot \text{R}} = 1.363 \end{aligned}$$



Therefore, the N_2-CO_2 mixture can be treated as a single ideal gas with above properties. Then the isentropic exit temperature can be determined from

$$T_{2s} = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (1400 \text{ R}) \left(\frac{12 \text{ psia}}{60 \text{ psia}} \right)^{0.363/1.363} = 911.7 \text{ R}$$

From the definition of isentropic efficiency,

$$\eta_N = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{c_p (T_1 - T_2)}{c_p (T_1 - T_{2s})} \longrightarrow 0.88 = \frac{1400 - T_2}{1400 - 911.7} \longrightarrow T_2 = \mathbf{970.3\ R}$$

(*b*) Noting that, $q = w = 0$, from the steady-flow energy balance relation,

$$\begin{aligned} \dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \\ \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ h_1 + V_1^2/2 &= h_2 + V_2^2/2 \\ 0 &= c_p (T_2 - T_1) + \frac{V_2^2 - V_1^{2\,\nearrow 0}}{2} \\ V_2 &= \sqrt{2 c_p (T_1 - T_2)} = \sqrt{2(0.2323 \text{ Btu/lbm} \cdot \text{R})(1400 - 970.3)\text{R} \left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right)} = \mathbf{2236\ ft/s} \end{aligned}$$

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13-66E Problem 13-65E is reconsidered. The problem is first to be solved and then, for all other conditions being the same, the problem is to be resolved to determine the composition of the nitrogen and carbon dioxide that is required to have an exit velocity of 2200 ft/s at the nozzle exit.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
mf_N2=0.65
mf_CO2=1-mf_N2
P1=60 [psia]
T1=1400 [R]
Vel1=0 [ft/s]
P2=12 [psia]
eta_N=0.88
"Vel2=2200 [ft/s]"

"Properties"
c_p_N2=0.248 [Btu/lbm-R]
c_v_N2=0.177 [Btu/lbm-R]
c_p_CO2=0.203 [Btu/lbm-R]
c_v_CO2=0.158 [Btu/lbm-R]
MM_N2=28 [lbm/lbmol]
MM_CO2=44 [lbm/lbmol]

"Analysis"
c_p_m=mf_N2*c_p_N2+mf_CO2*c_p_CO2
c_v_m=mf_N2*c_v_N2+mf_CO2*c_v_CO2
k_m=c_p_m/c_v_m
T2_s=T1*(P2/P1)^((k_m-1)/k_m)
eta_N=(T1-T2)/(T1-T2_s)
0=c_p_m*(T2-T1)+(Vel2^2-Vel1^2)/2*Convert(ft^2/s^2, Btu/lbm)
N_N2=mf_N2/MM_N2
N_CO2=mf_CO2/MM_CO2
N_total=N_N2+N_CO2
y_N2=N_N2/N_total
y_CO2=N_CO2/N_total
```

SOLUTION of the stated problem

c_p_CO2=0.203 [Btu/lbm-R]	c_p_m=0.2323 [Btu/lbm-R]	c_p_N2=0.248 [Btu/lbm-R]
c_v_CO2=0.158 [Btu/lbm-R]	c_v_m=0.1704 [Btu/lbm-R]	c_v_N2=0.177 [Btu/lbm-R]
eta_N=0.88	k_m=1.363	mf_CO2=0.35
mf_N2=0.65	MM_CO2=44 [lbm/lbmol]	MM_N2=28 [lbm/lbmol]
N_CO2=0.007955	N_N2=0.02321	N_total=0.03117
P1=60 [psia]	P2=12 [psia]	T1=1400 [R]
T2=970.3 [R]	T2_s=911.7 [R]	Vel1=0 [ft/s]
Vel2=2236 [ft/s]	**y_CO2=0.2552**	**y_N2=0.7448**

SOLUTION of the problem with exit velocity of 2200 ft/s

c_p_CO2=0.203 [Btu/lbm-R]	c_p_m=0.2285 [Btu/lbm-R]	c_p_N2=0.248 [Btu/lbm-R]
c_v_CO2=0.158 [Btu/lbm-R]	c_v_m=0.1688 [Btu/lbm-R]	c_v_N2=0.177 [Btu/lbm-R]
eta_N=0.88	k_m=1.354	mf_CO2=0.434
mf_N2=0.566	MM_CO2=44 [lbm/lbmol]	MM_N2=28 [lbm/lbmol]
N_CO2=0.009863	N_N2=0.02022	N_total=0.03008
P1=60 [psia]	P2=12 [psia]	T1=1400 [R]
T2=976.9 [R]	T2_s=919.3 [R]	Vel1=0 [ft/s]
Vel2=2200 [ft/s]	**y_CO2=0.3279**	**y_N2=0.6721**

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13-67 A mixture of hydrogen and oxygen is considered. The entropy change of this mixture between the two specified states is to be determined.

Assumptions Hydrogen and oxygen are ideal gases.

Properties The gas constants of hydrogen and oxygen are 4.124 and 0.2598 kJ/kg·K, respectively (Table A-1).

Analysis The effective gas constant of this mixture is

$R = \text{mf}_{H2} R_{H2} + \text{mf}_{O2} R_{O2} = (0.33)(4.1240) + (0.67)(0.2598) = 1.5350 \text{ kJ/kg} \cdot \text{K}$ Since the temperature of the two states is the same, the entropy change is determined from

$$s_2 - s_1 = -R \ln \frac{P_2}{P_1} = -(1.5350 \text{ kJ/kg} \cdot \text{K}) \ln \frac{150 \text{ kPa}}{750 \text{ kPa}} = \mathbf{2.470 \text{ kJ/kg} \cdot \text{K}}$$

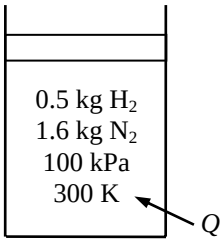
13-68 A piston-cylinder device contains a gas mixture at a given state. Heat is transferred to the mixture. The amount of heat transfer and the entropy change of the mixture are to be determined.

Assumptions **1** Under specified conditions both H_2 and N_2 can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** Kinetic and potential energy changes are negligible.

Properties The constant pressure specific heats of H_2 and N_2 at 450 K are 14.501 kJ/kg.K and 1.049 kJ/kg.K, respectively. (Table A-2b).

Analysis (*a*) Noting that $P_2 = P_1$ and $V_2 = 2V_1$,

$$\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1} \longrightarrow T_2 = \frac{2V_1}{V_1} T_1 = 2T_1 = (2)(300 \text{ K}) = 600 \text{ K}$$



From the closed system energy balance relation,

$$E_{in} - E_{out} = \Delta E_{system}$$
$$Q_{in} - W_{b,out} = \Delta U \quad \rightarrow \quad Q_{in} = \Delta H$$

since W_b and ΔU combine into ΔH for quasi-equilibrium constant pressure processes.

$$\begin{aligned} Q_{in} &= \Delta H = \Delta H_{H_2} + \Delta H_{N_2} = \left[mc_{p,avg}(T_2 - T_1)\right]_{H_2} + \left[mc_{p,avg}(T_2 - T_1)\right]_{N_2} \\ &= (0.5 \text{ kg})(14.501 \text{ kJ/kg} \cdot \text{K})(600 - 300)\text{K} + (1.6 \text{ kg})(1.049 \text{ kJ/kg} \cdot \text{K})(600 - 300)\text{K} \\ &= \mathbf{2679 \text{ kJ}} \end{aligned}$$

(*b*) Noting that the total mixture pressure, and thus the partial pressure of each gas, remains constant, the entropy change of the mixture during this process is

$$\Delta S_{H_2} = \left[m(s_2 - s_1)\right]_{H_2} = m_{H_2}\left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}^{\nearrow 0}\right)_{H_2} = m_{H_2}\left(c_p \ln \frac{T_2}{T_1}\right)_{H_2}$$
$$= (0.5 \text{ kg})(14.501 \text{ kJ/kg} \cdot \text{K}) \ln \frac{600 \text{ K}}{300 \text{ K}}$$
$$= 5.026 \text{ kJ/K}$$

$$\Delta S_{N_2} = \left[m(s_2 - s_1)\right]_{N_2} = m_{N_2}\left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}^{\nearrow 0}\right)_{N_2} = m_{N_2}\left(c_p \ln \frac{T_2}{T_1}\right)_{N_2}$$
$$= (1.6 \text{ kg})(1.049 \text{ kJ/kg} \cdot \text{K}) \ln \frac{600 \text{ K}}{300 \text{ K}}$$
$$= 1.163 \text{ kJ/K}$$

$$\Delta S_{total} = \Delta S_{H_2} + \Delta S_{N_2} = 5.026 \text{ kJ/K} + 1.163 \text{ kJ/K} = \mathbf{6.19 \text{ kJ/K}}$$

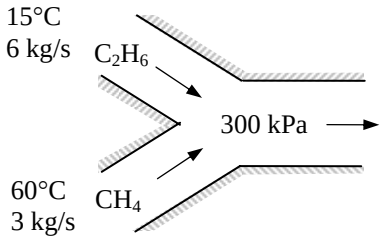
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13-69 The temperatures and pressures of two gases forming a mixture in a mixing chamber are given. The mixture temperature and the rate of entropy generation are to be determined.

Assumptions **1** Under specified conditions both C_2H_6 and CH_4 can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The mixing chamber is insulated and thus there is no heat transfer. **3** There are no other forms of work involved. **3** This is a steady-flow process. **4** The kinetic and potential energy changes are negligible.

Properties The specific heats of C_2H_6 and CH_4 are 1.7662 kJ/kg.°C and 2.2537 kJ/kg.°C, respectively. (Table A-2b).

Analysis (*a*) The enthalpy of ideal gases is independent of pressure, and thus the two gases can be treated independently even after mixing. Noting that $\dot{W} = \dot{Q} = 0$, the steady-flow energy balance equation reduces to



$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i = \dot{m}_{C_2H_6}(h_e - h_i)_{C_2H_6} + \dot{m}_{CH_4}(h_e - h_i)_{CH_4}$$

$$0 = \left[\dot{m} c_p (T_e - T_i)\right]_{C_2H_6} + \left[\dot{m} c_p (T_e - T_i)\right]_{CH_4}$$

Using c_p values at room temperature and substituting, the exit temperature of the mixture becomes

$$0 = (6\text{ kg/s})(1.7662\text{ kJ/kg}\cdot\text{°C})(T_m - 15\text{°C}) + (3\text{ kg/s})(2.2537\text{ kJ/kg}\cdot\text{°C})(T_m - 60\text{°C})$$

$$T_m = \mathbf{32.5°C}\ \ (305.5\text{ K})$$

(*b*) The rate of entropy change associated with this process is determined from an entropy balance on the mixing chamber,

$$\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = \Delta \dot{S}_{\text{system}}^{\nearrow 0} = 0$$

$$[\dot{m}(s_1 - s_2)]_{C_2H_6} + [\dot{m}(s_1 - s_2)]_{CH_4} + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = [\dot{m}(s_2 - s_1)]_{C_2H_6} + [\dot{m}(s_2 - s_1)]_{CH_4}$$

The molar flow rate of the two gases in the mixture is

$$\dot{N}_{C_2H_6} = \left(\frac{\dot{m}}{M}\right)_{C_2H_6} = \frac{6\text{ kg/s}}{30\text{ kg/kmol}} = 0.2\text{ kmol/s}$$

$$\dot{N}_{CH_4} = \left(\frac{\dot{m}}{M}\right)_{CH_4} = \frac{4.5\text{ kg/s}}{16\text{ kg/kmol}} = 0.1875\text{ kmol/s}$$

Then the mole fraction of each gas becomes

$$y_{C_2H_6} = \frac{0.2}{0.2 + 0.1875} = 0.5161$$

$$y_{CH_4} = \frac{0.1875}{0.2 + 0.1875} = 0.4839$$

Thus,

$$(s_2 - s_1)_{C_2H_6} = \left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{yP_{m,2}}{P_1}\right)_{C_2H_6} = \left(c_p \ln\frac{T_2}{T_1} - R\ln y\right)_{C_2H_6}$$

$$= (1.7662\text{ kJ/kg}\cdot\text{K})\ln\frac{305.5\text{ K}}{288\text{ K}} - (0.2765\text{ kJ/kg}\cdot\text{K})\ln(0.5161) = 0.2872\text{ kJ/kg}\cdot\text{K}$$

$$(s_2 - s_1)_{CH_4} = \left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{yP_{m,2}}{P_1}\right)_{CH_4} = \left(c_p \ln\frac{T_2}{T_1} - R\ln y\right)_{CH_4}$$

$$= (2.2537\text{ kJ/kg}\cdot\text{K})\ln\frac{305.5\text{ K}}{333\text{ K}} - (0.5182\text{ kJ/kg}\cdot\text{K})\ln(0.4839) = 0.1821\text{ kJ/kg}\cdot\text{K}$$

Noting that $P_{m,2} = P_{i,1} = 300\text{ kPa}$ and substituting,

$$\dot{S}_{\text{gen}} = (6\text{ kg/s})(0.2872\text{ kJ/kg}\cdot\text{K}) + (3\text{ kg/s})(0.1821\text{ kJ/kg}\cdot\text{K}) = \mathbf{2.27\ kW/K}$$

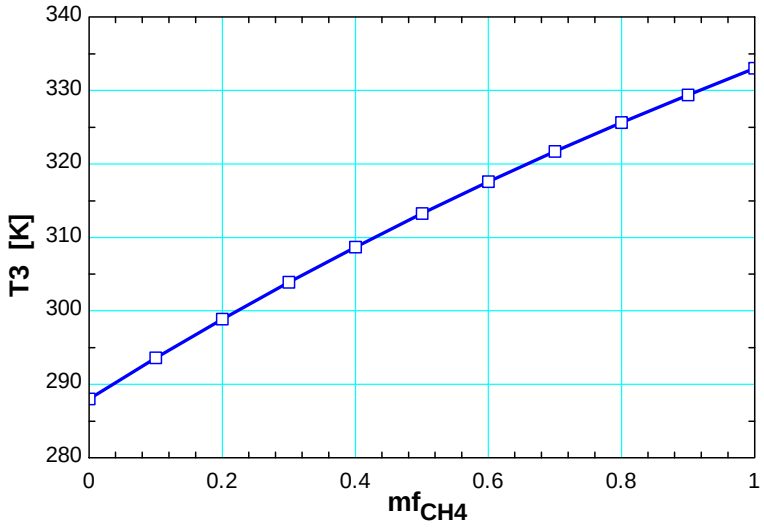
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13-70 Problem 13-69 is reconsidered. The effect of the mass fraction of methane in the mixture on the mixture temperature and the rate of exergy destruction is to be investigated.

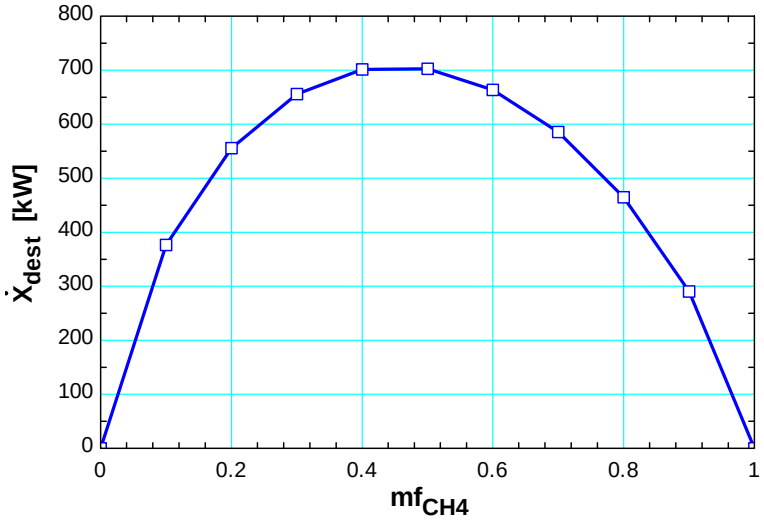
Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
"1: C2H6, 2: CH4"
m_dot_total=9 [kg/s]
"mf_CH4=0.3333"
mf_C2H6=1-mf_CH4
m_dot_1=mf_C2H6*m_dot_total
m_dot_2=mf_CH4*m_dot_total
T1=(15+273) [K]
T2=(60+273) [K]
P=300 [kPa]
T0=(25+273) [K]
"Properties"
c_p_1=1.7662 [kJ/kg-K]
c_p_2=2.2537 [kJ/kg-K]
R_1=0.2765 [kJ/kg-K]
R_2=0.5182 [kJ/kg-K]
MM_1=30 [kg/kmol]
MM_2=16 [kg/kmol]
"Analysis"
0=m_dot_1*c_p_1*(T3-T1)+m_dot_2*c_p_2*(T3-T2)
N_dot_1=m_dot_1/MM_1
N_dot_2=m_dot_2/MM_2
N_dot_total=N_dot_1+N_dot_2
y_1=N_dot_1/N_dot_total
y_2=N_dot_2/N_dot_total
DELTAs_1=c_p_1*ln(T3/T1)-R_1*ln(y_1)
DELTAs_2=c_p_2*ln(T3/T2)-R_2*ln(y_2)
S_dot_gen=m_dot_1*DELTAs_1+m_dot_2*DELTAs_2
X_dot_dest=T0*S_dot_gen
```

mf_{F2}	T3 [K]	X_{dest} [kW]
0	288	0
0.1	293.6	376.4
0.2	298.9	555.4
0.3	303.9	655.8
0.4	308.7	701.4
0.5	313.2	702.5
0.6	317.6	663.6
0.7	321.7	585.4
0.8	325.6	464.6
0.9	329.4	290.2
1	333	0.09793



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13-71E In an air-liquefaction plant, it is proposed that the pressure and temperature of air be adiabatically reduced. It is to be determined whether this process is possible and the work produced is to be determined using Kay's rule and the departure charts.

Assumptions Air is a gas mixture with 21% O_2 and 79% N_2, by mole.

Properties The molar masses of O_2 and N_2 are 32.0 and 28.0 lbm/lbmol, respectively. The critical properties are 278.6 R, 736 psia for O_2 and 227.1 R and 492 psia for N_2 (Table A-1E).

Analysis To use Kay's rule, we need to determine the pseudo-critical temperature and pseudo-critical pressure of the mixture using the critical point properties of gases.

$$T'_{\text{cr},m} = \sum y_i T_{\text{cr},i} = y_{\text{O2}} T_{\text{cr,O2}} + y_{\text{N2}} T_{\text{cr,N2}}$$
$$= (0.21)(278.6\text{ R}) + (0.79)(227.1\text{ R}) = 237.9\text{ R}$$
$$P'_{\text{cr},m} = \sum y_i P_{\text{cr},i} = y_{\text{O2}} P_{\text{cr,O2}} + y_{\text{N2}} P_{\text{cr,N2}}$$
$$= (0.21)(736\text{ psia}) + (0.79)(492\text{ psia}) = 543.2\text{ psia}$$

21% O_2
79% N_2
(by mole)
1500 psia
40°F

The enthalpy and entropy departure factors at the initial and final states are (from EES)

$$T_{R1} = \frac{T_{m1}}{T'_{\text{cr},m}} = \frac{500\text{ R}}{237.9\text{ R}} = 2.102 \Bigg\} Z_{h1} = 0.725$$
$$P_{R1} = \frac{P_{m1}}{P'_{\text{cr},m}} = \frac{1500\text{ psia}}{432.2\text{ psia}} = 3.471 \Bigg\} Z_{s1} = 0.339$$

$$T_{R2} = \frac{T_{m2}}{T'_{\text{cr},m}} = \frac{360\text{ R}}{237.9\text{ R}} = 1.513 \Bigg\} Z_{h2} = 0.0179$$
$$P_{R2} = \frac{P_{m2}}{P'_{\text{cr},m}} = \frac{15\text{ psia}}{432.2\text{ psia}} = 0.0347 \Bigg\} Z_{s2} = 0.00906$$

The enthalpy and entropy changes of the air under the ideal gas assumption is (Properties are from Table A-17E)

$$(h_2 - h_1)_{\text{ideal}} = 85.97 - 119.48 = -33.5\text{ Btu/lbm}$$

$$(s_2 - s_1)_{\text{ideal}} = s_2^{\text{o}} - s_1^{\text{o}} - R\ln\frac{P_2}{P_1} = 0.50369 - 0.58233 - (0.06855)\ln\frac{15}{1500} = 0.2370\text{ Btu/lbm}\cdot\text{R}$$

With departure factors, the enthalpy change (i.e., the work output) and the entropy change are

$$w_{\text{out}} = h_1 - h_2 = (h_1 - h_2)_{\text{ideal}} - RT'_{\text{cr}}(Z_{h1} - Z_{h2})$$
$$= 33.5 - (0.06855)(237.9)(0.725 - 0.0179) = \mathbf{22.0\ Btu/lbm}$$

$$s_2 - s_1 = (s_2 - s_1)_{\text{ideal}} - R(Z_{s2} - Z_{s1})$$
$$= 0.2370 - (0.06855)(0.00906 - 0.339) = \mathbf{0.2596\ Btu/lbm\cdot R}$$

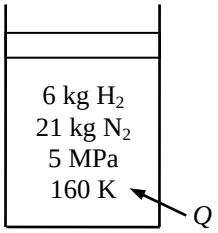
The entropy change in this case is equal to the entropy generation during the process since the process is adiabatic. The positive value of entropy generation shows that this process is possible.

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13-72 Heat is transferred to a gas mixture contained in a piston cylinder device. The initial state and the final temperature are given. The heat transfer is to be determined for the ideal gas and non-ideal gas cases.

Properties The molar masses of H_2 and N_2 are 2.0, and 28.0 kg/kmol. (Table A-1).

Analysis From the energy balance relation,



$$E_{\text{in}} - E_{\text{out}} = \Delta E$$
$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U$$
$$Q_{\text{in}} = \Delta H = \Delta H_{\text{H}_2} + \Delta H_{\text{N}_2} = N_{\text{H}_2}\left(\bar{h}_2 - \bar{h}_1\right)_{\text{H}_2} + N_{\text{N}_2}\left(\bar{h}_2 - \bar{h}_1\right)_{\text{N}_2}$$

since W_b and ΔU combine into ΔH for quasi-equilibrium constant pressure processes

$$N_{\text{H}_2} = \frac{m_{\text{H}_2}}{M_{\text{H}_2}} = \frac{6\text{ kg}}{2\text{ kg/kmol}} = 3\text{ kmol}$$
$$N_{\text{N}_2} = \frac{m_{\text{N}_2}}{M_{\text{N}_2}} = \frac{21\text{ kg}}{28\text{ kg/kmol}} = 0.75\text{ kmol}$$

(*a*) Assuming ideal gas behavior, the inlet and exit enthalpies of H_2 and N_2 are determined from the ideal gas tables to be

$$H_2: \quad \bar{h}_1 = \bar{h}_{@160\text{ K}} = 4{,}535.4\text{ kJ/kmol}, \qquad \bar{h}_2 = \bar{h}_{@200\text{ K}} = 5{,}669.2\text{ kJ/kmol}$$
$$N_2: \quad \bar{h}_1 = \bar{h}_{@160\text{ K}} = 4{,}648\text{ kJ/kmol}, \qquad \bar{h}_2 = \bar{h}_{@200\text{ K}} = 5{,}810\text{ kJ/kmol}$$

Thus, $\quad Q_{\text{ideal}} = 3\times(5{,}669.2 - 4{,}535.4) + 0.75\times(5{,}810 - 4{,}648) = \mathbf{4273\ kJ}$

(*b*) Using Amagat's law and the generalized enthalpy departure chart, the enthalpy change of each gas is determined to be

H_2:
$$\left.\begin{aligned} T_{R_1,\text{H}_2} &= \frac{T_{m,1}}{T_{\text{cr,H}_2}} = \frac{160}{33.3} = 4.805 \\ P_{R_1,\text{H}_2} = P_{R_2,\text{H}_2} &= \frac{P_m}{P_{\text{cr,H}_2}} = \frac{5}{1.30} = 3.846 \\ T_{R_2,\text{H}_2} &= \frac{T_{m,2}}{T_{\text{cr,H}_2}} = \frac{200}{33.3} = 6.006 \end{aligned}\right\} \begin{aligned} Z_{h_1} &\cong 0 \\ Z_{h_2} &\cong 0 \end{aligned} \qquad \text{(Fig. A-29)}$$

Thus H_2 can be treated as an ideal gas during this process.

N_2:
$$\left.\begin{aligned} T_{R_1,\text{N}_2} &= \frac{T_{m,1}}{T_{\text{cr,N}_2}} = \frac{160}{126.2} = 1.27 \\ P_{R_1,\text{N}_2} = P_{R_2,\text{N}_2} &= \frac{P_m}{P_{\text{cr,N}_2}} = \frac{5}{3.39} = 1.47 \\ T_{R_2,\text{N}_2} &= \frac{T_{m,2}}{T_{\text{cr,N}_2}} = \frac{200}{126.2} = 1.58 \end{aligned}\right\} \begin{aligned} Z_{h_1} &= 1.3 \\ Z_{h_2} &= 0.7 \end{aligned} \qquad \text{(Fig. A-29)}$$

Therefore,

$$\left(\bar{h}_2 - \bar{h}_1\right)_{\text{H}_2} = \left(\bar{h}_2 - \bar{h}_1\right)_{\text{H}_2,\text{ideal}} = 5{,}669.2 - 4{,}535.4 = 1{,}133.8\text{kJ/kmol}$$
$$\left(\bar{h}_2 - \bar{h}_1\right)_{\text{N}_2} = R_u T_{cr}\left(Z_{h_1} - Z_{h_2}\right) + \left(\bar{h}_2 - \bar{h}_1\right)_{\text{ideal}}$$
$$= (8.314\text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(126.2\text{K})(1.3 - 0.7) + (5{,}810 - 4{,}648)\text{kJ/kmol} = 1{,}791.5\text{kJ/kmol}$$

Substituting,

$$Q_{\text{in}} = (3\text{ kmol})(1{,}133.8\text{ kJ/kmol}) + (0.75\text{ kmol})(1{,}791.5\text{ kJ/kmol}) = \mathbf{4745\ kJ}$$

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13-73 Heat is transferred to a gas mixture contained in a piston cylinder device discussed in previous problem. The total entropy change and the exergy destruction are to be determined for two cases.

Analysis The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the piston-cylinder device and its immediate surroundings so that the boundary temperature of the extended system is the environment temperature at all times. It gives

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$

$$\frac{Q_{\text{in}}}{T_{\text{boundary}}} + S_{\text{gen}} = \Delta S_{\text{water}} \quad \rightarrow \quad S_{\text{gen}} = m(s_2 - s_1) - \frac{Q_{\text{in}}}{T_{\text{surr}}}$$

Then the exergy destroyed during a process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$.

(*a*) Noting that the total mixture pressure, and thus the partial pressure of each gas, remains constant, the entropy change of a component in the mixture during this process is

$$\Delta S_i = m_i \left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}^{\nearrow 0} \right)_i = m_i c_{p,i} \ln \frac{T_2}{T_1}$$

Assuming ideal gas behavior and using c_p values at the average temperature, the ΔS of H_2 and N_2 are determined from

$$\Delta S_{\text{H}_2,\text{ideal}} = (6 \text{ kg})(13.60 \text{ kJ/kg} \cdot \text{K}) \ln \frac{200 \text{ K}}{160 \text{ K}} = 18.21 \text{ kJ/K}$$

$$\Delta S_{\text{N}_2,\text{ideal}} = (21 \text{ kg})(1.039 \text{ kJ/kg} \cdot \text{K}) \ln \frac{200 \text{ K}}{160 \text{ K}} = 4.87 \text{ kJ/K}$$

and

$$S_{\text{gen}} = 18.21 \text{ kJ/K} + 4.87 \text{ kJ/K} - \frac{4273 \text{ kJ}}{303 \text{ K}} = \mathbf{8.98 \text{ kJ/K}}$$

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (303 \text{ K})(8.98 \text{ kJ/K}) = \mathbf{2721 \text{ kJ}}$$

(*b*) Using Amagat's law and the generalized entropy departure chart, the entropy change of each gas is determined to be

H_2:

$$\left.\begin{aligned} T_{R_1,\text{H}_2} &= \frac{T_{m,1}}{T_{\text{cr},\text{H}_2}} = \frac{160}{33.3} = 4.805 \\ P_{R_1,\text{H}_2} &= P_{R_2,\text{H}_2} = \frac{P_m}{P_{\text{cr},\text{H}_2}} = \frac{5}{1.30} = 3.846 \\ T_{R_2,\text{H}_2} &= \frac{T_{m,2}}{T_{\text{cr},\text{H}_2}} = \frac{200}{33.3} = 6.006 \end{aligned}\right\} \begin{aligned} Z_{s_1} &\cong 1 \\ Z_{s_2} &\cong 1 \end{aligned}$$

(Table A-30)

Thus H_2 can be treated as an ideal gas during this process.

N_2:

$$\left.\begin{aligned} T_{R_1,\text{N}_2} &= \frac{T_{m,1}}{T_{\text{cr},\text{N}_2}} = \frac{160}{126.2} = 1.268 \\ P_{R_1,\text{N}_2} &= P_{R_2,\text{N}_2} = \frac{P_m}{P_{\text{cr},\text{N}_2}} = \frac{5}{3.39} = 1.475 \\ T_{R_2,\text{N}_2} &= \frac{T_{m,2}}{T_{\text{cr},\text{N}_2}} = \frac{200}{126.2} = 1.585 \end{aligned}\right\} \begin{aligned} Z_{s_1} &= 0.8 \\ Z_{s_2} &= 0.4 \end{aligned}$$

(Table A-30)

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Therefore,

$$\Delta S_{H_2} = \Delta S_{H_2,\text{ideal}} = 18.21 \text{ kJ/K}$$

$$\begin{aligned}\Delta S_{N_2} &= N_{N_2} R_u \left(Z_{s_1} - Z_{s_2}\right) + \Delta S_{N_2,\text{ideal}} \\ &= (0.75 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(0.8 - 0.4) + (4.87 \text{ kJ/K}) \\ &= 7.37 \text{ kJ/K}\end{aligned}$$

$$\Delta S_{\text{surr}} = \frac{Q_{\text{surr}}}{T_0} = \frac{-4745 \text{ kJ}}{303 \text{ K}} = -15.66 \text{ kJ/K}$$

and

$$S_{\text{gen}} = 18.21 \text{ kJ/K} + 7.37 \text{ kJ/K} - \frac{4745 \text{ kJ}}{303 \text{ K}} = \mathbf{9.92 \text{ kJ/K}}$$

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (303 \text{ K})(9.92 \text{ kJ/K}) = \mathbf{3006 \text{ kJ}}$$

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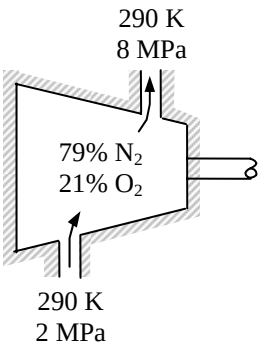
13-74 Air is compressed isothermally in a steady-flow device. The power input to the compressor and the rate of heat rejection are to be determined for ideal and non-ideal gas cases.

Assumptions **1** This is a steady-flow process. **2** The kinetic and potential energy changes are negligible.

Properties The molar mass of air is 28.97 kg/kmol. (Table A-1).

Analysis The mass flow rate of air can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{1.75 \text{ kg/s}}{28.97 \text{ kg/kmol}} = 0.06041 \text{ kmol/s}$$



(*a*) Assuming ideal gas behavior, the Δh and Δs of air during this process is

$$\Delta\bar{h} = 0 \text{ (isothermal process)}$$

$$\Delta\bar{s} = \bar{c}_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R_u \ln\frac{P_2}{P_1} = -R_u \ln\frac{P_2}{P_1}$$

$$= -(8.314 \text{ kJ/kg} \cdot \text{K}) \ln\frac{8 \text{ MPa}}{2 \text{ MPa}} = -11.53 \text{ kJ/kmol} \cdot \text{K}$$

Disregarding any changes in kinetic and potential energies, the steady-flow energy balance equation for the isothermal process of an ideal gas reduces to

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{N}\bar{h}_1 = \dot{Q}_{\text{out}} + \dot{N}\bar{h}_2$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{N}\Delta\bar{h}^{\nearrow 0} = 0 \longrightarrow \dot{W}_{\text{in}} = \dot{Q}_{\text{out}}$$

Also for an isothermal, internally reversible process the heat transfer is related to the entropy change by

$Q = T\Delta S = NT\Delta\bar{s}$,

$$\dot{Q} = \dot{N}T\Delta\bar{s} = (0.06041 \text{ kmol/s})(290 \text{ K})(-11.53 \text{ kJ/kmol} \cdot \text{K}) = -201.9 \text{ kW} \rightarrow \dot{Q}_{\text{out}} = 201.9 \text{ kW}$$

Therefore,

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}} = \mathbf{201.9 \text{ kW}}$$

(*b*) Using Amagat's law and the generalized charts, the enthalpy and entropy changes of each gas are determined from

$$\bar{h}_2 - \bar{h}_1 = R_u T_{cr}(Z_{h_1} - Z_{h_2}) + (\bar{h}_2 - \bar{h}_1)_{\text{ideal}}^{\nearrow 0}$$

$$\bar{s}_2 - \bar{s}_1 = R_u(Z_{s_1} - Z_{s_2}) + (\bar{s}_2 - \bar{s}_1)_{\text{ideal}}$$

where

N_2:

$$\left.\begin{aligned} P_{R_1} &= \frac{P_{m,1}}{P_{\text{cr,N}_2}} = \frac{2}{3.39} = 0.59 \\ T_{R_1} = T_{R_2} &= \frac{T_m}{T_{\text{cr,N}_2}} = \frac{290}{126.2} = 2.298 \\ P_{R_2} &= \frac{P_{m,2}}{P_{\text{cr,N}_2}} = \frac{8}{3.39} = 2.36 \end{aligned}\right\} \begin{aligned} &Z_{h_1} = 0.1154,\ Z_{s_1} = 0.05136 \\ &Z_{h_2} = 0.4136,\ Z_{s_2} = 0.1903 \end{aligned}$$

(Figures A-29 and A-30 or EES)

Note that we used EES to obtain enthalpy and entropy departure factors. The accurate readings like these are not possible with Figures A-29 and A-30. EES has built-in functions for enthalpy departure and entropy departure factors in the following format:

```
Z_h1=ENTHDEP(T_R1, P_R1) "the function that returns enthalpy departure factor"
Z_s1=ENTRDEP(T_R1, P_R1) "the function that returns entropy departure factor"
```

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O_2: $P_{R_1} = \dfrac{P_{m,1}}{P_{\text{cr},O_2}} = \dfrac{2}{5.08} = 0.3937$

$T_{R_1} = T_{R_2} = \dfrac{T_m}{T_{\text{cr},O_2}} = \dfrac{290}{154.8} = 1.873$

$P_{R_2} = \dfrac{P_{m,2}}{P_{\text{cr},O_2}} = \dfrac{8}{5.08} = 1.575$

$Z_{h_1} = 0.1296,\ Z_{s_1} = 0.05967$

$Z_{h_2} = 0.4956,\ Z_{s_2} = 0.2313$

(Figures A-29 and A-30 or EES)

Then,

$$\begin{aligned}\bar{h}_2 - \bar{h}_1 &= y_i \Delta \bar{h}_i = y_{N_2}(\bar{h}_2 - \bar{h}_1)_{N_2} + y_{O_2}(\bar{h}_2 - \bar{h}_1)_{O_2} \\ &= (0.79)(8.314)(126.2)(0.1154 - 0.4136) + (0.21)(8.314)(154.8)(0.1296 - 0.4956) + 0 \\ &= -346.1 \text{ kJ/kmol}\end{aligned}$$

$$\begin{aligned}\bar{s}_2 - \bar{s}_1 &= y_i \Delta \bar{s}_i = y_{N_2}(\bar{s}_2 - \bar{s}_1)_{N_2} + y_{O_2}(\bar{s}_2 - \bar{s}_1)_{O_2} \\ &= (0.79)(8.314)(0.05136 - 0.1903) + (0.21)(8.314)(0.05967 - 0.2313) + (-11.53) \\ &= -12.74 \text{ kJ/kmol} \cdot \text{K}\end{aligned}$$

Thus,

$$\dot{Q}_{\text{out}} = -\dot{N}T\Delta\bar{s} = -(0.06041 \text{ kmol/s})(290 \text{ K})(-12.74 \text{ kJ/kmol} \cdot \text{K}) = \mathbf{223.2\ kW}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{N}\bar{h}_1 = \dot{Q}_{\text{out}} + \dot{N}\bar{h}_2$$

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}} + \dot{N}(\bar{h}_2 - \bar{h}_1) \longrightarrow \dot{W}_{\text{in}} = 223.2 \text{ kW} + (0.06041 \text{ kmol/s})(-346.1 \text{ kJ/kmol}) = \mathbf{202.3\ kW}$$

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13-75 Problem 13-74 is reconsidered. The results obtained by assuming ideal behavior, real gas behavior with Amagat's law, and real gas behavior with EES data are to be compared.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
y_N2=0.79
y_O2=0.21
T=290 [K]
P1=2000 [kPa]
P2=8000 [kPa]
m_dot=1.75 [kg/s]

"Properties"
R_u=8.314 [kPa-m^3/kmol-K]
M_air=molarmass(air)
T_cr_N2=126.2 [K]
T_cr_O2=154.8 [K]
P_cr_N2=3390 [kPa]
P_cr_O2=5080 [kPa]

"Analysis"

"Ideal gas"
N_dot=m_dot/M_air
DELTAh_ideal=0 "isothermal process"
DELTAs_ideal=-R_u*ln(P2/P1) "isothermal process"
Q_dot_in_ideal=N_dot*T*DELTAs_ideal
W_dot_in_ideal=-Q_dot_in_ideal

"Amagad's law"
T_R1_N2=T/T_cr_N2
P_R1_N2=P1/P_cr_N2
Z_h1_N2=ENTHDEP(T_R1_N2, P_R1_N2) "the function that returns enthalpy departure factor"
Z_s1_N2=ENTRDEP(T_R1_N2, P_R1_N2) "the function that returns entropy departure factor"
T_R2_N2=T/T_cr_N2
P_R2_N2=P2/P_cr_N2
Z_h2_N2=ENTHDEP(T_R2_N2, P_R2_N2) "the function that returns enthalpy departure factor"
Z_s2_N2=ENTRDEP(T_R2_N2, P_R2_N2) "the function that returns entropy departure factor"
T_R1_O2=T/T_cr_O2
P_R1_O2=P1/P_cr_O2
Z_h1_O2=ENTHDEP(T_R1_O2, P_R1_O2) "the function that returns enthalpy departure factor"
Z_s1_O2=ENTRDEP(T_R1_O2, P_R1_O2) "the function that returns entropy departure factor"
T_R2_O2=T/T_cr_O2
P_R2_O2=P2/P_cr_O2
Z_h2_O2=ENTHDEP(T_R2_O2, P_R2_O2) "the function that returns enthalpy departure factor"
Z_s2_O2=ENTRDEP(T_R2_O2, P_R2_O2) "the function that returns entropy departure factor"
DELTAh=DELTAh_ideal-(y_N2*R_u*T_cr_N2*(Z_h2_N2-Z_h1_N2)+y_O2*R_u*T_cr_O2*(Z_h2_O2-Z_h1_O2))
DELTAs=DELTAs_ideal-(y_N2*R_u*(Z_s2_N2-Z_s1_N2)+y_O2*R_u*(Z_s2_O2-Z_s1_O2))
Q_dot_in_Amagad =N_dot*T*DELTAs
W_dot_in_Amagad=-Q_dot_in_Amagad +N_dot*DELTAh

"EES"
h_EES[1] = y_N2*enthalpy(Nitrogen,T=T, P=P1)+ y_O2*enthalpy(Oxygen,T=T,P=P1)
h_EES[2] = y_N2*enthalpy(Nitrogen,T=T, P=P2)+ y_O2*enthalpy(Oxygen,T=T,P=P2)
s_EES[1] = y_N2*entropy(Nitrogen,T=T, P=P1)+ y_O2*entropy(Oxygen,T=T,P=P1)
s_EES[2] = y_N2*entropy(Nitrogen,T=T, P=P2)+ y_O2*entropy(Oxygen,T=T,P=P2)
DELTAh_EES=h_EES[2]-h_EES[1]
```

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DELTAs_EES=s_EES[2]-s_EES[1]
Q_dot_in_EES=N_dot*T*DELTAs_EES
W_dot_in_EES=-Q_dot_in_EES+N_dot*DELTAh_EES

SOLUTION

DELTAh=-346.1 [kJ/kmol]
DELTAh_EES=-384.3 [kJ/kmol]
DELTAh_ideal=0 [kJ/kmol]
DELTAs=-12.74 [kJ/kmol-K]
DELTAs_EES=-12.72 [kJ/kmol-K]
DELTAs_ideal=-11.53 [kJ/kmol-K]
h_EES[1]=6473 [kJ/kmol]
h_EES[2]=6089 [kJ/kmol]
M_air=28.97 [kg/kmol]
m_dot=1.75 [kg/s]
N_dot=0.06041 [kmol/s]
P1=2000 [kPa]
P2=8000 [kPa]
P_cr_N2=3390 [kPa]
P_cr_O2=5080 [kPa]
P_R1_N2=0.59
P_R1_O2=0.3937
P_R2_N2=2.36
P_R2_O2=1.575
Q_dot_in_Amagad=-223.2 [kW]
Q_dot_in_EES=-222.9 [kW]
Q_dot_in_ideal=-201.9 [kW]
R_u=8.314 [kPa-m^3/kmol-K]
s_EES[1]=125.3 [kJ/kmol-K]
s_EES[2]=112.5 [kJ/kmol-K]
T=290 [K]
T_cr_N2=126.2 [K]
T_cr_O2=154.8 [K]
T_R1_N2=2.298
T_R1_O2=1.873
T_R2_N2=2.298
T_R2_O2=1.873
W_dot_in_Amagad=202.3 [kW]
W_dot_in_EES=199.7 [kW]
W_dot_in_ideal=201.9 [kW]
y_N2=0.79
y_O2=0.21
Z_h1_N2=0.1154
Z_h1_O2=0.1296
Z_h2_N2=0.4136
Z_h2_O2=0.4956
Z_s1_N2=0.05136
Z_s1_O2=0.05967
Z_s2_N2=0.1903
Z_s2_O2=0.2313

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13-76 Two mass streams of two different ideal gases are mixed in a steady-flow chamber while receiving energy by heat transfer from the surroundings. Expressions for the final temperature and the exit volume flow rate are to be obtained and two special cases are to be evaluated.

Assumptions Kinetic and potential energy changes are negligible.

Analysis (a) Mass and Energy Balances for the mixing process:

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

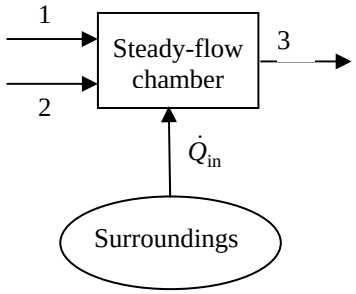
$$\dot{m}_1 h_1 + \dot{m}_2 h_2 + \dot{Q}_{in} = \dot{m}_3 h_3$$

$$h = C_P T$$

$$\dot{m}_1 C_{P,1} T_1 + \dot{m}_2 C_{P,2} T_2 + \dot{Q}_{in} = \dot{m}_3 C_{P,m} T_3$$

$$C_{P,m} = \frac{\dot{m}_1}{\dot{m}_3} C_{P,1} + \frac{\dot{m}_2}{\dot{m}_3} C_{P,2}$$

$$T_3 = \frac{\dot{m}_1 C_{P,1}}{\dot{m}_3 C_{P,m}} T_1 + \frac{\dot{m}_2 C_{P,2}}{\dot{m}_3 C_{P,m}} T_2 + \frac{\dot{Q}_{in}}{\dot{m}_3 C_{P,m}}$$



(b) The expression for the exit volume flow rate is obtained as follows:

$$\dot{V}_3 = \dot{m}_3 v_3 = \dot{m}_3 \frac{R_3 T_3}{P_3}$$

$$\dot{V}_3 = \frac{\dot{m}_3 R_3}{P_3}\left[\frac{\dot{m}_1 C_{P,1}}{\dot{m}_3 C_{P,m}} T_1 + \frac{\dot{m}_2 C_{P,2}}{\dot{m}_3 C_{P,m}} T_2 + \frac{\dot{Q}_{in}}{\dot{m}_3 C_{P,m}}\right]$$

$$\dot{V}_3 = \frac{C_{P,1} R_3}{C_{P,m} R_1}\frac{\dot{m}_1 R_1 T_1}{P_3} + \frac{C_{P,2} R_3}{C_{P,m} R_2}\frac{\dot{m}_2 R_2 T_2}{P_3} + \frac{R_3 \dot{Q}_{in}}{P_3 C_{P,m}}$$

$$P_3 = P_1 = P_2$$

$$\dot{V}_3 = \frac{C_{P,1} R_3}{C_{P,m} R_1}\dot{V}_1 + \frac{C_{P,2} R_3}{C_{P,m} R_2}\dot{V}_2 + \frac{R_3 \dot{Q}_{in}}{P_3 C_{P,m}}$$

$$R = \frac{R_u}{M}, \quad \frac{R_3}{R_1} = \frac{R_u}{M_3}\frac{M_1}{R_u} = \frac{M_1}{M_3}, \quad \frac{R_3}{R_2} = \frac{M_2}{M_3}$$

$$\dot{V}_3 = \frac{C_{P,1} M_1}{C_{P,m} M_3}\dot{V}_1 + \frac{C_{P,2} M_2}{C_{P,m} M_3}\dot{V}_2 + \frac{R_u \dot{Q}_{in}}{P_3 M_3 C_{P,m}}$$

The mixture molar mass M_3 is found as follows:

$$M_3 = \sum y_i M_i, \quad y_i = \frac{m_{fi}/M_i}{\sum m_{fi}/M_i}, \quad m_{fi} = \frac{\dot{m}_i}{\sum \dot{m}_i}$$

(c) For adiabatic mixing $\dot{Q}_{in}$ is zero, and the mixture volume flow rate becomes

$$\dot{V}_3 = \frac{C_{P,1} M_1}{C_{P,m} M_3}\dot{V}_1 + \frac{C_{P,2} M_2}{C_{P,m} M_3}\dot{V}_2$$

(d) When adiabatically mixing the same two ideal gases, the mixture volume flow rate becomes

$$M_3 = M_1 = M_2$$

$$C_{P,3} = C_{P,1} = C_{P,2}$$

$$\dot{V}_3 = \dot{V}_1 + \dot{V}_2$$

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Special Topic: Chemical Potential and the Separation Work of Mixtures

13-77C No, a process that separates a mixture into its components without requiring any work (exergy) input is impossible since such a process would violate the 2nd law of thermodynamics.

13-78C Yes, the volume of the mixture can be more or less than the sum of the initial volumes of the mixing liquids because of the attractive or repulsive forces acting between dissimilar molecules.

13-79C The person who claims that the temperature of the mixture can be higher than the temperatures of the components is right since the total enthalpy of the mixture of two components at the same pressure and temperature, in general, is not equal to the sum of the total enthalpies of the individual components before mixing, the difference being the enthalpy (or heat) of mixing, which is the heat released or absorbed as two or more components are mixed isothermally.

13-80C Mixtures or solutions in which the effects of molecules of different components on each other are negligible are called ideal solutions (or ideal mixtures). The ideal-gas mixture is just one category of ideal solutions. For ideal solutions, the enthalpy change and the volume change due to mixing are zero, but the entropy change is not. The chemical potential of a component of an ideal mixture is independent of the identity of the other constituents of the mixture. The chemical potential of a component in an ideal mixture is equal to the Gibbs function of the pure component.

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13-81 Brackish water is used to produce fresh water. The minimum power input and the minimum height the brackish water must be raised by a pump for reverse osmosis are to be determined.

Assumptions **1** The brackish water is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is also 12°C.

Properties The molar masses of water and salt are M_w = 18.0 kg/kmol and M_s = 58.44 kg/kmol. The gas constant of pure water is R_w = 0.4615 kJ/kg·K (Table A-1). The density of fresh water is 1000 kg/m^3.

Analysis First we determine the mole fraction of pure water in brackish water using Eqs. 13-4 and 13-5. Noting that mf_s = 0.00078 and $\text{mf}_w = 1 - \text{mf}_s = 0.99922$,

$$M_{\text{m}} = \frac{1}{\sum \frac{\text{mf}_i}{M_i}} = \frac{1}{\frac{\text{mf}_s}{M_s} + \frac{\text{mf}_w}{M_w}} = \frac{1}{\frac{0.00078}{58.44} + \frac{0.99922}{18.0}} = 18.01\,\text{kg/kmol}$$

$$y_i = \text{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \text{mf}_w \frac{M_m}{M_w} = (0.99922)\frac{18.01\,\text{kg/kmol}}{18.0\,\text{kg/kmol}} = 0.99976$$

The minimum work input required to produce 1 kg of freshwater from brackish water is

$$w_{\text{min, in}} = R_w T_0 \ln(1/y_w) = (0.4615\,\text{kJ/kg}\cdot\text{K})(285.15\,\text{K})\ln(1/0.99976) = 0.03159\,\text{kJ/kg fresh water}$$

Therefore, 0.03159 kJ of work is needed to produce 1 kg of fresh water is mixed with seawater reversibly. Therefore, the required power input to produce fresh water at the specified rate is

$$\dot{W}_{\text{min, in}} = \rho \dot{V} w_{\text{min, in}} = (1000\,\text{kg/m}^3)(0.280\,\text{m}^3/\text{s})(0.03159\,\text{kJ/kg})\left(\frac{1\,\text{kW}}{1\,\text{kJ/s}}\right) = \mathbf{8.85\ kW}$$

The minimum height to which the brackish water must be pumped is

$$\Delta z_{\text{min}} = \frac{w_{\text{min,in}}}{g} = \left(\frac{0.03159\,\text{kJ/kg}}{9.81\,\text{m/s}^2}\right)\left(\frac{1\,\text{kg.m/s}^2}{1\,\text{N}}\right)\left(\frac{1000\,\text{N.m}}{1\,\text{kJ}}\right) = \mathbf{3.22\ m}$$

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13-82 A river is discharging into the ocean at a specified rate. The amount of power that can be generated is to be determined.

Assumptions **1** The seawater is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is also 15°C.

Properties The molar masses of water and salt are M_w = 18.0 kg/kmol and M_s = 58.44 kg/kmol. The gas constant of pure water is R_w = 0.4615 kJ/kg·K (Table A-1). The density of river water is 1000 kg/m^3.

Analysis First we determine the mole fraction of pure water in ocean water using Eqs. 13-4 and 13-5. Noting that mf_s = 0.025 and mf_w = 1- mf_s = 0.975,

$$M_{\text{m}} = \frac{1}{\sum \frac{\text{mf}_i}{M_i}} = \frac{1}{\frac{\text{mf}_s}{M_s} + \frac{\text{mf}_w}{M_w}} = \frac{1}{\frac{0.025}{58.44} + \frac{0.975}{18.0}} = 18.32 \text{ kg/kmol}$$

$$y_i = \text{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \text{mf}_w \frac{M_m}{M_w} = (0.975)\frac{18.32 \text{ kg/kmol}}{18.0 \text{ kg/kmol}} = 0.9922$$

The maximum work output associated with mixing 1 kg of seawater (or the minimum work input required to produce 1 kg of freshwater from seawater) is

$$w_{\max,\text{out}} = R_w T_0 \ln(1/y_w) = (0.4615 \text{ kJ/kg} \cdot \text{K})(288.15 \text{ K})\ln(1/0.9922) = 1.046 \text{ kJ/kg fresh water}$$

Therefore, 1.046 kJ of work can be produced as 1 kg of fresh water is mixed with seawater reversibly. Therefore, the power that can be generated as a river with a flow rate of 400,000 m^3/s mixes reversibly with seawater is

$$\dot{W}_{\max \text{ out}} = \rho \dot{\mathcal{V}} w_{\max \text{ out}} = (1000 \text{ kg/m}^3)(1.5 \times 10^5 \text{ m}^3\text{/s})(1.046 \text{ kJ/kg})\left(\frac{1 \text{ kW}}{1 \text{ kJ/s}}\right) = \mathbf{157 \times 10^6 \text{ kW}}$$

Discussion This is more power than produced by all nuclear power plants (112 of them) in the U.S., which shows the tremendous amount of power potential wasted as the rivers discharge into the seas.

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13-83 Problem 13-82 is reconsidered. The effect of the salinity of the ocean on the maximum power generated is to be investigated.

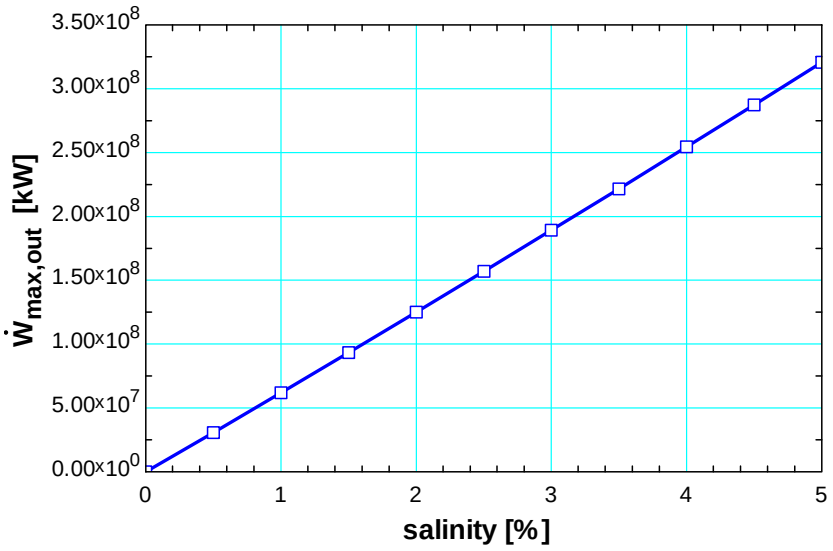
Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
V_dot=150000 [m^3/s]
"salinity=2.5"
T=(15+273.15) [K]

"Properties"
M_w=18 [kg/kmol] "molarmass(H2O)"
M_s=58.44 [kg/kmol] "molar mass of salt"
R_w=0.4615 [kJ/kg-K] "gas constant of water"
rho=1000 [kg/m^3]

"Analysis"
mass_w=100-salinity
mf_s=salinity/100
mf_w=mass_w/100
M_m=1/(mf_s/M_s+mf_w/M_w)
y_w=mf_w*M_m/M_w
w_max_out=R_w*T*ln(1/y_w)
W_dot_max_out=rho*V_dot*w_max_out
```

Salinity [%]	$W_{max,out}$ [kW]
0	0
0.5	3.085E+07
1	6.196E+07
1.5	9.334E+07
2	1.249E+08
2.5	1.569E+08
3	1.891E+08
3.5	2.216E+08
4	2.544E+08
4.5	2.874E+08
5	3.208E+08



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13-84E Brackish water is used to produce fresh water. The mole fractions, the minimum work inputs required to separate 1 lbm of brackish water and to obtain 1 lbm of fresh water are to be determined.

Assumptions **1** The brackish water is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is equal to the water temperature.

Properties The molar masses of water and salt are M_w = 18.0 lbm/lbmol and M_s = 58.44 lbm/lbmol. The gas constant of pure water is R_w = 0.1102 Btu/lbm·R (Table A-1E).

Analysis (*a*) First we determine the mole fraction of pure water in brackish water using Eqs. 13-4 and 13-5. Noting that mf_s = 0.0012 and mf_w = 1- mf_s = 0.9988,

$$M_{\text{m}} = \frac{1}{\sum \frac{\text{mf}_i}{M_i}} = \frac{1}{\frac{\text{mf}_s}{M_s} + \frac{\text{mf}_w}{M_w}} = \frac{1}{\frac{0.0012}{58.44} + \frac{0.9988}{18.0}} = 18.015 \text{ lbm/lbmol}$$

$$y_i = \text{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \text{mf}_w \frac{M_m}{M_w} = (0.9988)\frac{18.015 \text{ lbm/lbmol}}{18.0 \text{ lbm/lbmol}} = \mathbf{0.99963}$$

$$y_s = 1 - y_w = 1 - 0.99963 = \mathbf{0.00037}$$

(*b*) The minimum work input required to separate 1 lbmol of brackish water is

$$\begin{aligned} w_{\text{min,in}} &= -R_w T_0 (y_w \ln y_w + y_s \ln y_s) \\ &= -(0.1102 \text{ Btu/lbmol.R})(525 \text{ R})[0.99963 \ln(0.99963) + 0.00037 \ln(0.00037)] \\ &= -\mathbf{0.191 Btu/lbm} \text{ brackish water} \end{aligned}$$

(*c*) The minimum work input required to produce 1 lbm of freshwater from brackish water is

$$w_{\text{min, in}} = R_w T_0 \ln(1/y_w) = (0.1102 \text{ Btu/lbm} \cdot \text{R})(525 \text{ R})\ln(1/0.99963) = \mathbf{0.0214 \text{ Btu/lbm fresh water}}$$

Discussion Note that it takes about 9 times work to separate 1 lbm of brackish water into pure water and salt compared to producing 1 lbm of fresh water from a large body of brackish water.

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13-85 A desalination plant produces fresh water from seawater. The second law efficiency of the plant is to be determined.

Assumptions **1** The seawater is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is equal to the seawater temperature.

Properties The molar masses of water and salt are M_w = 18.0 kg/kmol and M_s = 58.44 kg/kmol. The gas constant of pure water is R_w = 0.4615 kJ/kg·K (Table A-1). The density of river water is 1000 kg/m^3.

Analysis First we determine the mole fraction of pure water in seawater using Eqs. 13-4 and 13-5. Noting that mf_s = 0.032 and mf_w = 1- mf_s = 0.968,

$$M_{\text{m}} = \frac{1}{\sum \frac{\text{mf}_i}{M_i}} = \frac{1}{\frac{\text{mf}_s}{M_s} + \frac{\text{mf}_w}{M_w}} = \frac{1}{\frac{0.032}{58.44} + \frac{0.968}{18.0}} = 18.41\,\text{kg/kmol}$$

$$y_i = \text{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \text{mf}_w \frac{M_m}{M_w} = (0.968)\frac{18.41\,\text{kg/kmol}}{18.0\,\text{kg/kmol}} = 0.9900$$

The maximum work output associated with mixing 1 kg of seawater (or the minimum work input required to produce 1 kg of freshwater from seawater) is

$$w_{\text{max, out}} = R_w T_0 \ln(1/y_w) = (0.4615\,\text{kJ/kg}\cdot\text{K})(283.15\,\text{K})\ln(1/0.990) = 1.313\,\text{kJ/kg fresh water}$$

The power that can be generated as 1.4 m^3/s fresh water mixes reversibly with seawater is

$$\dot{W}_{\text{max out}} = \rho \dot{V} w_{\text{max out}} = (1000\,\text{kg/m}^3)(1.4\,\text{m}^3/\text{s})(1.313\,\text{kJ/kg})\left(\frac{1\,\text{kW}}{1\,\text{kJ/s}}\right) = 1.84\,\text{kW}$$

Then the second law efficiency of the plant becomes

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{min,in}}}{\dot{W}_{\text{in}}} = \frac{1.83\ \text{MW}}{8.5\,\text{MW}} = 0.216 = \mathbf{21.6\%}$$

13-86 The power consumption and the second law efficiency of a desalination plant are given. The power that can be produced if the fresh water produced is mixed with the seawater reversibly is to be determined.

Assumptions **1** This is a steady-flow process. **2** The kinetic and potential energy changes are negligible.

Analysis From the definition of the second law efficiency

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{actual}}} \quad \rightarrow \quad 0.25 = \frac{\dot{W}_{\text{rev}}}{11{,}500\,\text{kW}} \quad \rightarrow \quad \dot{W}_{\text{rev}} = \mathbf{2875\,kW}$$

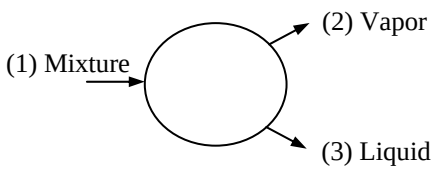
which is the maximum power that can be generated.

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13-87E It is to be determined if it is it possible for an adiabatic liquid-vapor separator to separate wet steam at 100 psia and 90 percent quality, so that the pressure of the outlet streams is greater than 100 psia.

Analysis Because the separator divides the inlet stream into the liquid and vapor portions,

$$\dot{m}_2 = x\dot{m}_1 = 0.9\dot{m}_1$$
$$\dot{m}_3 = (1-x)\dot{m}_1 = 0.1\dot{m}_1$$



According to the water property tables at 100 psia (Table A-5E),

$$s_1 = s_f + xs_{fg} = 0.47427 + 0.9 \times 1.12888 = 1.4903\,\text{Btu/lbm} \cdot \text{R}$$

When the increase in entropy principle is adapted to this system, it becomes

$$\dot{m}_2 s_2 + \dot{m}_3 s_3 \geq \dot{m}_1 s_1$$
$$x\dot{m}_1 s_2 + (1-x)\dot{m}_1 s_3 \geq \dot{m}_1 s_1$$
$$0.9s_2 + 0.1s_3 \geq s_1$$
$$\geq 1.4903\,\text{Btu/lbm} \cdot \text{R}$$

To test this hypothesis, let's assume the outlet pressures are 110 psia. Then,

$$s_2 = s_g = 1.5954\,\text{Btu/lbm} \cdot \text{R}$$
$$s_3 = s_f = 0.48341\,\text{Btu/lbm} \cdot \text{R}$$

The left-hand side of the above equation is

$$0.9s_2 + 0.1s_3 = 0.9 \times 1.5954 + 0.1 \times 0.48341 = 1.4842\,\text{Btu/lbm} \cdot \text{R}$$

which is less than the minimum possible specific entropy. Hence, the outlet pressure cannot be 110 psia. Inspection of the water table in light of above equation proves that the pressure at the separator outlet cannot be greater than that at the inlet.

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Review Problems

13-88 Using Dalton's law, it is to be shown that $Z_m = \sum_{i=1}^{k} y_i Z_i$ for a real-gas mixture.

Analysis Using the compressibility factor, the pressure of a component of a real-gas mixture and of the pressure of the gas mixture can be expressed as

$$P_i = \frac{Z_i N_i R_u T_m}{V_m} \quad \text{and} \quad P_m = \frac{Z_m N_m R_u T_m}{V_m}$$

Dalton's law can be expressed as $P_m = \sum P_i(T_m, V_m)$. Substituting,

$$\frac{Z_m N_m R_u T_m}{V_m} = \sum \frac{Z_i N_i R_u T_m}{V_m}$$

Simplifying,

$$Z_m N_m = \sum Z_i N_i$$

Dividing by N_m,

$$Z_m = \sum y_i Z_i$$

where Z_i is determined at the mixture temperature and volume.

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13-89 The volume fractions of components of a gas mixture are given. The mole fractions, the mass fractions, the partial pressures, the mixture molar mass, apparent gas constant, and constant-pressure specific heat are to be determined and compared to the values in Table A-2a.

Properties The molar masses of N_2, O_2 and Ar are 28.0, 32.0, and 40.0 kg/kmol, respectively (Table A-1). The constant-pressure specific heats of these gases at 300 K are 1.039, 0.918, and 0.5203 kJ/kg·K, respectively (Table A-2a).

Analysis The volume fractions are equal to the mole fractions:

$$y_{N2} = \mathbf{0.78},\ y_{O2} = \mathbf{0.21},\ y_{Ar} = \mathbf{0.01}$$

78% N_2
21% O_2
1% Ar
(by volume)

The volume fractions are equal to the pressure fractions. The partial pressures are then

$$P_{N2} = y_{N2} P_{total} = (0.78)(100\ \text{kPa}) = \mathbf{78\ kPa}$$
$$P_{O2} = y_{O2} P_{total} = (0.21)(100\ \text{kPa}) = \mathbf{21 kPa}$$
$$P_{Ar} = y_{Ar} P_{total} = (0.01)(100\ \text{kPa}) = \mathbf{1 kPa}$$

We consider 100 kmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$m_{N2} = N_{N2} M_{N2} = (78\ \text{kmol})(28\text{kg/kmol}) = 2184\ \text{kg}$$
$$m_{O2} = N_{O2} M_{O2} = (21\ \text{kmol})(32\ \text{kg/kmol}) = 672\ \text{kg}$$
$$m_{Ar} = N_{Ar} M_{Ar} = (1\ \text{kmol})(40\ \text{kg/kmol}) = 40\ \text{kg}$$

The total mass is

$$m_m = m_{N2} + m_{O2} + m_{Ar} = 2184 + 672 + 40 = 2896\ \text{kg}$$

Then the mass fractions are

$$\text{mf}_{N2} = \frac{m_{N2}}{m_m} = \frac{2184\ \text{kg}}{2896\ \text{kg}} = \mathbf{0.7541}$$
$$\text{mf}_{O2} = \frac{m_{O2}}{m_m} = \frac{672\ \text{kg}}{2896\ \text{kg}} = \mathbf{0.2320}$$
$$\text{mf}_{Ar} = \frac{m_{Ar}}{m_m} = \frac{40\ \text{kg}}{2896\ \text{kg}} = \mathbf{0.0138}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{2896\ \text{kg}}{100\ \text{kmol}} = \mathbf{28.96\ kg/kmol}$$

The constant-pressure specific heat of the mixture is determined from

$$\begin{aligned} c_p &= \text{mf}_{N2} c_{p,N2} + \text{mf}_{O2} c_{p,O2} + \text{mf}_{Ar} c_{p,Ar} \\ &= 0.7541 \times 1.039 + 0.2320 \times 0.918 + 0.0138 \times 0.5203 \\ &= \mathbf{1.004\ kJ/kg \cdot K} \end{aligned}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol} \cdot \text{K}}{28.96\ \text{kg/kmol}} = \mathbf{0.2871 kJ/kg \cdot K}$$

This mixture closely correspond to the air, and the mixture properies determined (mixture molar mass, mixture gas constant and mixture specific heat) are practically the same as those listed for air in Tables A-1 and A-2a.

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13-90 The mole numbers of combustion gases are given. The partial pressure of water vapor and the condensation temperature of water vapor are to be determined.

Properties The molar masses of CO_2, H_2O, O_2 and N_2 are 44.0, 18.0, 32.0, and 28.0 kg/kmol, respectively (Table A-1).

Analysis The total mole of the mixture and the mole fraction of water vapor are

$$N_{\text{total}} = 8 + 9 + 12.5 + 94 = 123.5 \text{ kmol}$$

$$y_{\text{H2O}} = \frac{N_{\text{H2O}}}{N_{\text{total}}} = \frac{9}{123.5} = 0.07287$$

Noting that molar fraction is equal to pressure fraction, the partial pressure of water vapor is

$$P_{\text{H2O}} = y_{\text{H2O}} P_{\text{total}} = (0.07287)(100 \text{ kPa}) = \mathbf{7.29 \text{ kPa}}$$

The temperature at which the condensation starts is the saturation temperature of water at this pressure. This is called the dew-point temperature. Then,

$$T_{\text{cond}} = T_{\text{sat@7.29 kPa}} = \mathbf{39.7°C} \qquad \text{(Table A-5)}$$

Water vapor in the combustion gases will start to condense when the temperature of the combustion gases drop to 39.7°C.

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13-91 The masses of gases forming a mixture at a specified pressure and temperature are given. The mass of the gas mixture is to be determined using four methods.

Properties The molar masses of O_2, CO_2, and He are 32.0, 44.0, and 4.0 kg/kmol, respectively (Table A-1).

Analysis (*a*) The given total mass of the mixture is

$$m_m = m_{O2} + m_{CO2} + m_{He} = 0.1 + 1 + 0.5 = 1.6 \text{ kg}$$

The mole numbers of each component are

$$N_{O2} = \frac{m_{O2}}{M_{O2}} = \frac{0.1 \text{ kg}}{32 \text{ kg/kmol}} = 0.003125 \text{ kmol}$$

$$N_{CO2} = \frac{m_{CO2}}{M_{CO2}} = \frac{1 \text{ kg}}{44 \text{ kg/kmol}} = 0.02273 \text{ kmol}$$

$$N_{He} = \frac{m_{He}}{M_{He}} = \frac{0.5 \text{ kg}}{4 \text{ kg/kmol}} = 0.125 \text{ kmol}$$

0.1 kg O_2
1 kg CO_2
0.5 kg He

The mole number of the mixture is

$$N_m = N_{O2} + N_{CO2} + N_{He} = 0.003125 + 0.02273 + 0.125 = 0.1509 \text{ kmol}$$

Then the apparent molecular weight of the mixture becomes

$$M_m = \frac{m_m}{N_m} = \frac{1.6 \text{ kg}}{0.1509 \text{ kmol}} = 10.61 \text{ kg/kmol}$$

The mass of this mixture in a 0.3 m^3 tank is

$$m = \frac{M_m P \boldsymbol{V}}{R_u T} = \frac{(10.61 \text{ kg/kmol})(17{,}500 \text{ kPa})(0.3 \text{ m}^3)}{(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(293 \text{ K})} = \mathbf{22.87 \text{ kg}}$$

(*b*) To use the Amagat's law for this real gas mixture, we first need the mole fractions and the *Z* of each component at the mixture temperature and pressure.

$$y_{O2} = \frac{N_{O2}}{N_m} = \frac{0.003125 \text{ kmol}}{0.1509 \text{ kmol}} = 0.02071$$

$$y_{CO2} = \frac{N_{CO2}}{N_m} = \frac{0.02273 \text{ kmol}}{0.1509 \text{ kmol}} = 0.1506$$

$$y_{He} = \frac{N_{He}}{N_m} = \frac{0.125 \text{ kmol}}{0.1509 \text{ kmol}} = 0.8284$$

$$\left.\begin{aligned} T_{R,O2} &= \frac{T_m}{T_{cr,O2}} = \frac{293 \text{ K}}{154.8 \text{ K}} = 1.893 \\ P_{R,O2} &= \frac{P_m}{P_{cr,O2}} = \frac{17.5 \text{ MPa}}{5.08 \text{ MPa}} = 3.445 \end{aligned}\right\} Z_{O2} = 0.93 \qquad \text{(Fig. A-15)}$$

$$\left.\begin{aligned} T_{R,CO2} &= \frac{T_m}{T_{cr,CO2}} = \frac{293 \text{ K}}{304.2 \text{ K}} = 0.963 \\ P_{R,CO2} &= \frac{P_m}{P_{cr,CO2}} = \frac{17.5 \text{ MPa}}{7.39 \text{ MPa}} = 2.368 \end{aligned}\right\} Z_{CO2} = 0.33 \qquad \text{(Fig. A-15)}$$

$$\left.\begin{aligned} T_{R,He} &= \frac{T_m}{T_{cr,He}} = \frac{293 \text{ K}}{5.3 \text{ K}} = 55.3 \\ P_{R,He} &= \frac{P_m}{P_{cr,He}} = \frac{17.5 \text{ MPa}}{0.23 \text{ MPa}} = 76.1 \end{aligned}\right\} Z_{He} = 1.04 \qquad \text{(from EES)}$$

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Then,

$$Z_m = \sum y_i Z_i = y_{O2} Z_{O2} + y_{CO2} Z_{CO2} + y_{He} Z_{He}$$
$$= (0.02071)(0.93) + (0.1506)(0.33) + (0.8284)(1.04) = 0.9305$$

$$m = \frac{M_m P \mathcal{V}}{Z_m R_u T} = \frac{(10.61\,\text{kg/kmol})(17{,}500\,\text{kPa})(0.3\,\text{m}^3)}{(0.9305)(8.314\,\text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293\,\text{K})} = \mathbf{24.57\ kg}$$

(*c*) To use Dalton's law with compressibility factors:

$$\left.\begin{aligned} T_{R,O2} &= 1.893 \\ \nu_{R,O2} &= \frac{\mathcal{V}_m/m_{O2}}{R_{O2}T_{cr,O2}/P_{cr,O2}} = \frac{(0.3\,\text{m}^3)/(22.87\times 0.1/1.6\,\text{kg})}{(0.2598\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(154.8\,\text{K})/(5080\,\text{kPa})} = 26.5 \end{aligned}\right\} Z_{O2} = 1.0$$

$$\left.\begin{aligned} T_{R,CO2} &= 0.963 \\ \nu_{R,CO2} &= \frac{\mathcal{V}_m/m_{CO2}}{R_{CO2}T_{cr,CO2}/P_{cr,CO2}} = \frac{(0.3\,\text{m}^3)/(22.87\times 1.0/1.6\,\text{kg})}{(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(304.2\,\text{K})/(7390\,\text{kPa})} = 2.70 \end{aligned}\right\} Z_{CO2} = 0.86$$

$$\left.\begin{aligned} T_{R,He} &= 55.3 \\ \nu_{R,He} &= \frac{\mathcal{V}_m/m_{He}}{R_{He}T_{cr,He}/P_{cr,He}} = \frac{(0.3\,\text{m}^3)/(22.87\times 0.5/1.6\,\text{kg})}{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(5.3\,\text{K})/(230\,\text{kPa})} = 0.88 \end{aligned}\right\} Z_{He} = 1.0$$

Note that we used m = 22.87 kg in above calculations, the value obtained by ideal gas behavior. The solution normally requires iteration until the assumed and calculated mass values match. The mass of the component gas is obtained by multiplying the mass of the mixture by its mass fraction. Then,

$$Z_m = \sum y_i Z_i = y_{O2} Z_{O2} + y_{CO2} Z_{CO2} + y_{He} Z_{He}$$
$$= (0.02071)(1.0) + (0.1506)(0.86) + (0.8284)(1.0) = 0.9786$$

$$m = \frac{M_m P \mathcal{V}}{Z_m R_u T} = \frac{(10.61\,\text{kg/kmol})(17{,}500\,\text{kPa})(0.3\,\text{m}^3)}{(0.9786)(8.314\,\text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293\,\text{K})} = \mathbf{23.37\ kg}$$

This mass is sufficiently close to the mass value 22.87 kg. Therefore, there is no need to repeat the calculations at this calculated mass.

(*d*) To use Kay's rule, we need to determine the pseudo-critical temperature and pseudo-critical pressure of the mixture using the critical point properties of O_2, CO_2 and He.

$$T'_{cr,m} = \sum y_i T_{cr,i} = y_{O2} T_{cr,O2} + y_{CO2} T_{cr,CO2} + y_{He} T_{cr,He}$$
$$= (0.02071)(154.8\,\text{K}) + (0.1506)(304.2\,\text{K}) + (0.8284)(5.3\,\text{K}) = 53.41\,\text{K}$$
$$P'_{cr,m} = \sum y_i P_{cr,i} = y_{O2} P_{cr,O2} + y_{CO2} P_{cr,CO2} + y_{He} P_{cr,He}$$
$$= (0.02071)(5.08\,\text{MPa}) + (0.1506)(7.39\,\text{MPa}) + (0.8284)(0.23\,\text{MPa}) = 1.409\,\text{MPa}$$

Then,

$$\left.\begin{aligned} T_R &= \frac{T_m}{T'_{cr,m}} = \frac{293\,\text{K}}{53.41\,\text{K}} = 5.486 \\ P_R &= \frac{P_m}{P'_{cr,m}} = \frac{17.5\,\text{MPa}}{1.409\,\text{MPa}} = 12.42 \end{aligned}\right\} Z_m = 1.194 \qquad \text{(from EES)}$$

$$m = \frac{M_m P \mathcal{V}}{Z_m R_u T} = \frac{(10.61\,\text{kg/kmol})(17{,}500\,\text{kPa})(0.3\,\text{m}^3)}{(1.194)(8.314\,\text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293\,\text{K})} = \mathbf{19.15\ kg}$$

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13-92 A mixture of carbon dioxide and nitrogen flows through a converging nozzle. The required make up of the mixture on a mass basis is to be determined.

Assumptions Under specified conditions CO_2 and N_2 can be treated as ideal gases, and the mixture as an ideal gas mixture.

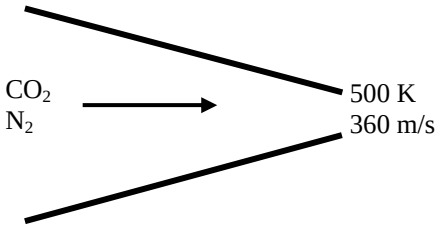
Properties The molar masses of CO_2 and N_2 are 44.0 and 28.0 kg/kmol, respectively (Table A-1). The specific heat ratios of CO_2 and N_2 at 500 K are $k_{CO2} = 1.229$ and $k_{N2} = 1.391$ (Table A-2).

Analysis The molar mass of the mixture is determined from

$$M_m = y_{CO_2} M_{CO_2} + y_{N_2} M_{N_2}$$

The molar fractions are related to each other by

$$y_{CO_2} + y_{N_2} = 1$$



The gas constant of the mixture is given by

$$R_m = \frac{R_u}{M_m}$$

The specific heat ratio of the mixture is expressed as

$$k = \text{mf}_{CO_2} k_{CO_2} + \text{mf}_{N_2} k_{N_2}$$

The mass fractions are

$$\text{mf}_{CO_2} = y_{CO_2} \frac{M_{CO_2}}{M_m}$$

$$\text{mf}_{N_2} = y_{N_2} \frac{M_{N_2}}{M_m}$$

The exit velocity equals the speed of sound at 500 K

$$V_{\text{exit}} = \sqrt{kR_m T \left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}} \right)}$$

Substituting the given values and known properties and solving the above equations simultaneously using EES, we find

$$\text{mf}_{CO_2} = \mathbf{0.838}$$

$$\text{mf}_{N_2} = \mathbf{0.162}$$

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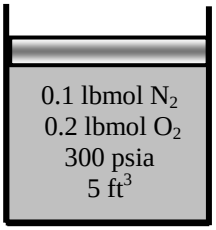
13-93E A mixture of nitrogen and oxygen is expanded isothermally. The work produced is to be determined.

Assumptions **1** Nitrogen and oxygen are ideal gases. **2** The process is reversible.

Properties The mole numbers of nitrogen and oxygen are 28.0 and 32.0 lbm/lbmol, respectively (Table A-1E).

Analysis The mole fractions are

$$y_{N2} = \frac{N_{N2}}{N_{total}} = \frac{0.1\,\text{lbmol}}{0.3\,\text{lbmol}} = 0.3333$$
$$y_{O2} = \frac{N_{O2}}{N_{total}} = \frac{0.2\,\text{kmol}}{0.3\,\text{kmol}} = 0.6667$$



The gas constant for this mixture is then

$$R = \frac{R_u}{y_{N2}M_{N2} + y_{O2}M_{O2}}$$
$$= \frac{1.9858\,\text{Btu/lbmol}\cdot\text{R}}{(0.3333\times 28 + 0.6667\times 32)\,\text{lbm/lbmol}}$$
$$= 0.06475\,\text{Btu/lbm}\cdot\text{R}$$
$$= (0.06475\,\text{Btu/lbm}\cdot\text{R})\left(\frac{5.404\,\text{psia}\cdot\text{ft}^3}{1\,\text{Btu}}\right)$$
$$= 0.3499\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$$

The mass of this mixture of gases is

$$m = N_{N2}M_{N2} + N_{O2}M_{O2} = 0.1\times 28 + 0.2\times 32 = 9.2\,\text{lbm}$$

The temperature of the mixture is

$$T_1 = \frac{P_1 V_1}{mR} = \frac{(300\,\text{psia})(5\,\text{ft}^3)}{(9.2\,\text{lbm})(0.3499\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})} = 466.0\,\text{R}$$

Noting that $Pv = RT$ for an ideal gas, the work done for this process is then

$$W_{out} = m\int_1^2 P\,dv = mRT\int_1^2 \frac{dv}{v} = mRT\ln\frac{v_2}{v_1}$$
$$= (9.2\,\text{lbm})(0.06475\,\text{Btu/lbm}\cdot\text{R})(466\,\text{R})\ln\frac{10\,\text{ft}^3}{5\,\text{ft}^3}$$
$$= \mathbf{192.4\,Btu}$$

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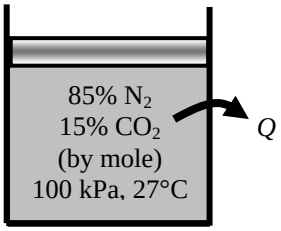
13-94 A mixture of nitrogen and carbon dioxide is compressed at constant temperature in a closed system. The work required is to be determined.

Assumptions **1** Nitrogen and carbon dioxide are ideal gases. **2** The process is reversible.

Properties The mole numbers of nitrogen and carbon dioxide are 28.0 and 44.0 kg/kmol, respectively (Table A-1).

Analysis The effective molecular weight of this mixture is

$$\begin{aligned} M &= y_{N2} M_{N2} + y_{CO2} M_{CO2} \\ &= (0.85)(28) + (0.15)(44) \\ &= 30.4 \text{ kg/kmol} \end{aligned}$$



The work done is determined from

$$w = \int_1^2 P dV = RT \int_1^2 \frac{dv}{v} = RT \ln \frac{v_2}{v_1} = RT \ln \frac{P_2}{P_1} = \frac{R_u}{M} RT \ln \frac{P_2}{P_1}$$

$$= \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{30.4 \text{ kg/kmol}} (300 \text{ K}) \ln \frac{500 \text{ kPa}}{100 \text{ kPa}}$$

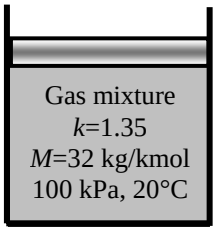
$$= \mathbf{132.0 \text{ kJ/kg}}$$

13-95 The specific heat ratio and an apparent molecular weight of a mixture of ideal gases are given. The work required to compress this mixture isentropically in a closed system is to be determined.

Analysis For an isentropic process of an ideal gas with constant specific heats, the work is expressed as

$$w_{out} = \int_1^2 P dv = P_1 v_1^k \int_1^2 v^{-k} dv$$

$$= \frac{P_1 v_1^k}{1-k} (v_2^{1-k} - v_1^{1-k}) = \frac{P_1 v_1^k}{1-k} \left[\left(\frac{v_2}{v_1} \right)^{1-k} - 1 \right]$$



since $P_1 v_1^k = P v^k$ for an isentropic process. Also,

$$P_1 v_1 = RT_1$$

$$(v_2 / v_1)^k = P_1 / P_2$$

Substituting, we obtain

$$w_{out} = \frac{R_u T_1}{M(1-k)} \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right]$$

$$= \frac{(8.314 \text{ kJ/kmol} \cdot \text{K})(293 \text{ K})}{(32 \text{ kg/kmol})(1-1.35)} \left[\left(\frac{1000 \text{ kPa}}{100 \text{ kPa}} \right)^{(1.35-1)/1.35} - 1 \right]$$

$$= \mathbf{-177.6 \text{ kJ/kg}}$$

The negative sign shows that the work is done on the system.

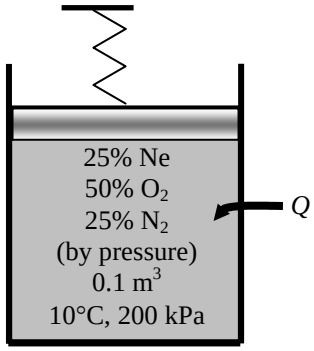
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13-96 A mixture of gases is placed in a spring-loaded piston-cylinder device. The device is now heated until the pressure rises to a specified value. The total work and heat transfer for this process are to be determined.

Properties The molar masses of Ne, O_2, and N_2 are 20.18, 32.0, 28.0 kg/kmol, respectively and the gas constants are 0.4119, 0.2598, and 0.2968 kJ/kg·K, respectively (Table A-1). The constant-volume specific heats are 0.6179, 0.658, and 0.743 kJ/kg·K, respectively (Table A-2a).

Analysis The total pressure is 200 kPa and the partial pressures are

$$P_{\text{Ne}} = y_{\text{Ne}} P_m = (0.25)(200\ \text{kPa}) = 50\ \text{kPa}$$
$$P_{\text{O2}} = y_{\text{O2}} P_m = (0.50)(200\ \text{kPa}) = 100\ \text{kPa}$$
$$P_{\text{N2}} = y_{\text{N2}} P_m = (0.25)(200\ \text{kPa}) = 50\ \text{kPa}$$



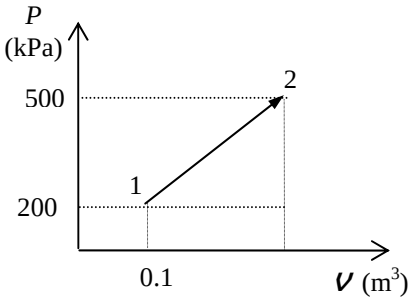
The mass of each constituent for a volume of 0.1 m^3 and a temperature of 10°C are

$$m_{\text{Ne}} = \frac{P_{\text{Ne}} V_m}{R_{\text{Ne}} T} = \frac{(50\ \text{kPa})(0.1\ \text{m}^3)}{(0.4119\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\ \text{K})} = 0.04289\ \text{kg}$$

$$m_{\text{O2}} = \frac{P_{\text{O2}} V_m}{R_{\text{O2}} T} = \frac{(100\ \text{kPa})(0.1\ \text{m}^3)}{(0.2598\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\ \text{K})} = 0.1360\ \text{kg}$$

$$m_{\text{N2}} = \frac{P_{\text{N2}} V_m}{R_{\text{N2}} T} = \frac{(50\ \text{kPa})(0.1\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\ \text{K})} = 0.05953\ \text{kg}$$

$$m_{\text{total}} = 0.04289 + 0.1360 + 0.05953 = 0.2384\ \text{kg}$$



The mass fractions are

$$\text{mf}_{\text{Ne}} = \frac{m_{\text{Ne}}}{m_m} = \frac{0.04289\ \text{kg}}{0.2384\ \text{kg}} = 0.1799$$

$$\text{mf}_{\text{O2}} = \frac{m_{\text{O2}}}{m_m} = \frac{0.1360\ \text{kg}}{0.2384\ \text{kg}} = 0.5705$$

$$\text{mf}_{\text{N2}} = \frac{m_{\text{N2}}}{m_m} = \frac{0.05953\ \text{kg}}{0.2384\ \text{kg}} = 0.2497$$

The constant-volume specific heat of the mixture is determined from

$$c_v = \text{mf}_{\text{Ne}} c_{v,\text{Ne}} + \text{mf}_{\text{O2}} c_{v,\text{O2}} + \text{mf}_{\text{N2}} c_{v,\text{N2}}$$
$$= 0.1799 \times 0.6179 + 0.5705 \times 0.658 + 0.2497 \times 0.743 = 0.672\ \text{kJ/kg}\cdot\text{K}$$

The moles are

$$N_{\text{Ne}} = \frac{m_{\text{Ne}}}{M_{\text{Ne}}} = \frac{0.04289\ \text{kg}}{20.18\ \text{kg/kmol}} = 0.002126\ \text{kmol}$$

$$N_{\text{O2}} = \frac{m_{\text{O2}}}{M_{\text{O2}}} = \frac{0.1360\ \text{kg}}{32\ \text{kg/kmol}} = 0.00425\ \text{kmol}$$

$$N_{\text{N2}} = \frac{m_{\text{N2}}}{M_{\text{N2}}} = \frac{0.05953\ \text{kg}}{28\ \text{kg/kmol}} = 0.002126\ \text{kmol}$$

$$N_{\text{m}} = N_{\text{Ne}} + N_{\text{O2}} + N_{\text{N2}} = 0.008502\ \text{kmol}$$

Then the apparent molecular weight of the mixture becomes

$$M_m = \frac{m_m}{N_m} = \frac{0.2384\ \text{kg}}{0.008502\ \text{kmol}} = 28.04\ \text{kg/kmol}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol}\cdot\text{K}}{28.05\ \text{kg/kmol}} = 0.2964\ \text{kJ/kg}\cdot\text{K}$$

The mass contained in the system is

$$m = \frac{P_1 V_1}{R T_1} = \frac{(200\ \text{kPa})(0.1\ \text{m}^3)}{(0.2964\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\ \text{K})} = 0.2384\ \text{kg}$$

Noting that the pressure changes linearly with volume, the final volume is determined by linear interpolation to be

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$$\frac{500-200}{1000-200}=\frac{V_2-0.1}{1.0-0.1}\longrightarrow V_2=0.4375\,\text{m}^3$$

The final temperature is

$$T_2=\frac{P_2V_2}{mR}=\frac{(500\,\text{kPa})(0.4375\,\text{m}^3)}{(0.2384\,\text{kg})(0.2964\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})}=3096\,\text{K}$$

The work done during this process is

$$W_{\text{out}}=\frac{P_1+P_2}{2}(V_2-V_1)=\frac{(500+200)\,\text{kPa}}{2}(0.4375-0.1)\,\text{m}^3=\mathbf{118\,kJ}$$

An energy balance on the system gives

$$Q_{\text{in}}=W_{\text{out}}+mc_v(T_2-T_1)=118+(0.2384\,\text{kg})(0.672\,\text{kJ/kg}\cdot\text{K})(3096-283)\,\text{K}=\mathbf{569\,kJ}$$

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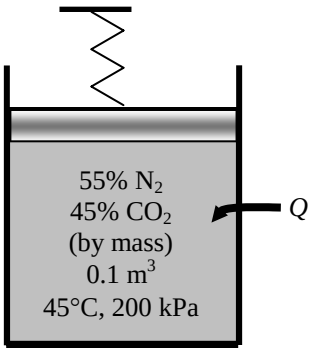
13-97 A spring-loaded piston-cylinder device is filled with a mixture of nitrogen and carbon dioxide whose mass fractions are given. The gas is heated until the volume has doubled. The total work and heat transfer for this process are to be determined.

Properties The molar masses of N_2 and CO_2 are 28.0 and 44.0 kg/kmol, respectively (Table A-1). The constant-volume specific heats of these gases at room temperature are 0.743 and 0.657 kJ/kg·K, respectively (Table A-2a).

Analysis We consider 100 kg of this mixture. The mole numbers of each component are

$$N_{N2} = \frac{m_{N2}}{M_{N2}} = \frac{55\text{ kg}}{28\text{ kg/kmol}} = 1.964\text{ kmol}$$

$$N_{CO2} = \frac{m_{CO2}}{M_{CO2}} = \frac{45\text{ kg}}{44\text{ kg/kmol}} = 1.023\text{ kmol}$$



The mole number of the mixture is

$$N_m = N_{N2} + N_{CO2} = 1.964 + 1.023 = 2.987\text{ kmol}$$

The apparent molecular weight of the mixture is

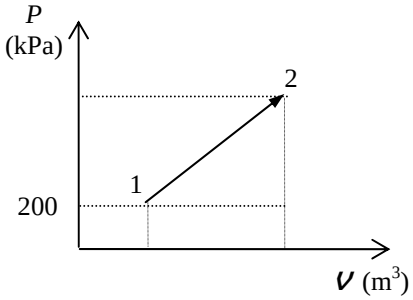
$$M_m = \frac{m_m}{N_m} = \frac{100\text{ kg}}{2.987\text{ kmol}} = 33.48\text{ kg/kmol}$$

The constant-volume specific heat of the mixture is determined from

$$c_v = \text{mf}_{N2} c_{v,N2} + \text{mf}_{CO2} c_{v,CO2} = 0.55 \times 0.743 + 0.45 \times 0.657 = 0.7043\text{ kJ/kg} \cdot \text{K}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.134\text{ kJ/kmol} \cdot \text{K}}{33.48\text{ kg/kmol}} = 0.2483\text{ kJ/kg} \cdot \text{K}$$



Noting that the pressure changes linearly with volume, the initial volume is determined by linear interpolation using the data of the previous problem to be

$$\frac{200 - 200}{1000 - 200} = \frac{V_1 - 0.1}{1.0 - 0.1} \longrightarrow V_1 = 0.1\text{ m}^3$$

The final volume is

$$V_2 = 2V_1 = 2(0.1\text{ m}^3) = 0.2\text{ m}^3$$

The final pressure is similarly determined by linear interpolation using the data of the previous problem to be

$$\frac{P_2 - 200}{1000 - 200} = \frac{0.2 - 0.1}{1.0 - 0.1} \longrightarrow P_2 = 288.9\text{ kPa}$$

The mass contained in the system is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(200\text{ kPa})(0.1\text{ m}^3)}{(0.2483\text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(318\text{ K})} = 0.2533\text{ kg}$$

The final temperature is

$$T_2 = \frac{P_2 V_2}{mR} = \frac{(288.9\text{ kPa})(0.2\text{ m}^3)}{(0.2533\text{ kg})(0.2483\text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})} = 918.7\text{ K}$$

The work done during this process is

$$W_{out} = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 288.9)\text{ kPa}}{2}(0.2 - 0.1)\text{ m}^3 = \mathbf{24.4\text{ kJ}}$$

An energy balance on the system gives

$$Q_{in} = W_{out} + mc_v(T_2 - T_1) = 24.4 + (0.2533\text{ kg})(0.7043\text{ kJ/kg} \cdot \text{K})(918.7 - 318)\text{ K} = \mathbf{132\text{ kJ}}$$

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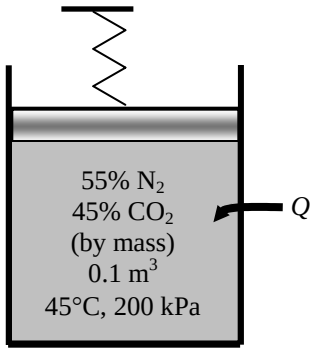
13-98 A spring-loaded piston-cylinder device is filled with a mixture of nitrogen and carbon dioxide whose mass fractions are given. The gas is heated until the pressure has tripled. The total work and heat transfer for this process are to be determined.

Properties The molar masses of N_2 and CO_2 are 28.0 and 44.0 kg/kmol, respectively (Table A-1). The constant-volume specific heats of these gases at room temperature are 0.743 and 0.657 kJ/kg·K, respectively (Table A-2a).

Analysis We consider 100 kg of this mixture. The mole numbers of each component are

$$N_{N2} = \frac{m_{N2}}{M_{N2}} = \frac{55\text{ kg}}{28\text{ kg/kmol}} = 1.964\text{ kmol}$$

$$N_{CO2} = \frac{m_{CO2}}{M_{CO2}} = \frac{45\text{ kg}}{44\text{ kg/kmol}} = 1.023\text{ kmol}$$



The mole number of the mixture is

$$N_m = N_{N2} + N_{CO2} = 1.964 + 1.023 = 2.987\text{ kmol}$$

The apparent molecular weight of the mixture is

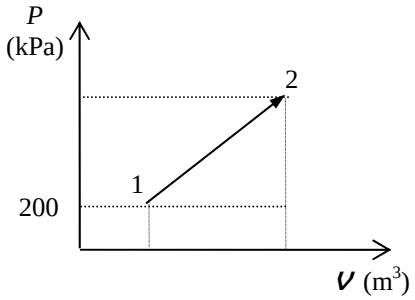
$$M_m = \frac{m_m}{N_m} = \frac{100\text{ kg}}{2.987\text{ kmol}} = 33.48\text{ kg/kmol}$$

The constant-volume specific heat of the mixture is determined from

$$c_v = \text{mf}_{N2} c_{v,N2} + \text{mf}_{CO2} c_{v,CO2} = 0.55 \times 0.743 + 0.45 \times 0.657 = 0.7043\text{ kJ/kg} \cdot \text{K}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.134\text{ kJ/kmol} \cdot \text{K}}{33.48\text{ kg/kmol}} = 0.2483\text{ kJ/kg} \cdot \text{K}$$



Noting that the pressure changes linearly with volume, the initial volume is determined by linear interpolation using the data of the previous problem to be

$$\frac{200 - 200}{1000 - 200} = \frac{V_1 - 0.1}{1.0 - 0.1} \longrightarrow V_1 = 0.1\text{ m}^3$$

The final pressure is

$$P_2 = 3P_1 = 3(200\text{ kPa}) = 600\text{ kPa}$$

The final volumee is similarly determined by linear interpolation using the data of the previous problem to be

$$\frac{600 - 200}{1000 - 200} = \frac{V_2 - 0.1}{1.0 - 0.1} \longrightarrow V_2 = 0.55\text{ m}^3$$

The mass contained in the system is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(200\text{ kPa})(0.1\text{ m}^3)}{(0.2483\text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(318\text{ K})} = 0.2533\text{ kg}$$

The final temperature is

$$T_2 = \frac{P_2 V_2}{mR} = \frac{(600\text{ kPa})(0.55\text{ m}^3)}{(0.2533\text{ kg})(0.2483\text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})} = 5247\text{ K}$$

The work done during this process is

$$W_{out} = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 600)\text{ kPa}}{2}(0.55 - 0.1)\text{ m}^3 = \mathbf{180\text{ kJ}}$$

An energy balance on the system gives

$$Q_{in} = W_{out} + mc_v(T_2 - T_1) = 180 + (0.2533\text{ kg})(0.7043\text{ kJ/kg} \cdot \text{K})(5247 - 318)\text{ K} = \mathbf{1059\text{ kJ}}$$

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13-99 The masses, pressures, and temperatures of the constituents of a gas mixture in a tank are given. Heat is transferred to the tank. The final pressure of the mixture and the heat transfer are to be determined.

Assumptions He is an ideal gas and O_2 is a nonideal gas.

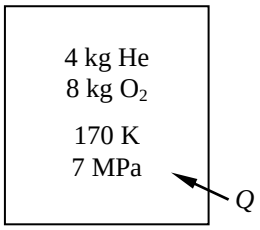
Properties The molar masses of He and O_2 are 4.0 and 32.0 kg/kmol. (Table A-1)

Analysis (*a*) The number of moles of each gas is

$$N_{\text{He}} = \frac{m_{\text{He}}}{M_{\text{He}}} = \frac{4 \text{ kg}}{4.0 \text{ kg/kmol}} = 1 \text{ kmol}$$

$$N_{\text{O}_2} = \frac{m_{\text{O}_2}}{M_{\text{O}_2}} = \frac{8 \text{ kg}}{32 \text{ kg/kmol}} = 0.25 \text{ kmol}$$

$$N_m = N_{\text{He}} + N_{\text{O}_2} = 1 \text{ kmol} + 0.25 \text{ kmol} = 1.25 \text{ kmol}$$



Then the partial volume of each gas and the volume of the tank are

He:

$$V_{\text{He}} = \frac{N_{\text{He}} R_u T_1}{P_{m,1}} = \frac{(1 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(170 \text{ K})}{7000 \text{ kPa}} = 0.202 \text{ m}^3$$

O_2:

$$\left.\begin{aligned} P_{R_1} &= \frac{P_{m,1}}{P_{\text{cr,O}_2}} = \frac{7}{5.08} = 1.38 \\ T_{R_1} &= \frac{T_1}{T_{\text{cr,O}_2}} = \frac{170}{154.8} = 1.10 \end{aligned}\right\} Z_1 = 0.53 \qquad \text{(Fig. A-15)}$$

$$V_{\text{O}_2} = \frac{Z N_{\text{O}_2} R_u T_1}{P_{m,1}} = \frac{(0.53)(0.25 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(170 \text{ K})}{7000 \text{ kPa}} = 0.027 \text{ m}^3$$

$$V_{\text{tank}} = V_{\text{He}} + V_{\text{O}_2} = 0.202 \text{ m}^3 + 0.027 \text{ m}^3 = 0.229 \text{ m}^3$$

The partial pressure of each gas and the total final pressure is

He:

$$P_{\text{He},2} = \frac{N_{\text{He}} R_u T_2}{V_{\text{tank}}} = \frac{(1 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(220 \text{ K})}{0.229 \text{ m}^3} = 7987 \text{ kPa}$$

O_2:

$$\left.\begin{aligned} T_{R_2} &= \frac{T_2}{T_{\text{cr,O}_2}} = \frac{220}{154.8} = 1.42 \\ v_{R,\text{O}_2} &= \frac{\bar{v}_{\text{O}_2}}{R_u T_{\text{cr,O}_2} / P_{\text{cr,O}_2}} = \frac{V_m / N_{\text{O}_2}}{R_u T_{\text{cr,O}_2} / P_{\text{cr,O}_2}} \\ &= \frac{(0.229 \text{ m}^3)/(0.25 \text{ kmol})}{(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(154.8 \text{ K})/(5080 \text{ kPa})} = 3.616 \end{aligned}\right\} P_R = 0.39 \qquad \text{(Fig. A-15)}$$

$$P_{\text{O}_2} = (P_R P_{\text{cr}})_{\text{O}_2} = (0.39)(5080 \text{ kPa}) = 1981 \text{ kPa} = 1.981 \text{ MPa}$$

$$P_{\text{m},2} = P_{\text{He}} + P_{\text{O}_2} = 7.987 \text{ MPa} + 1.981 \text{ MPa} = \mathbf{9.97 \text{ MPa}}$$

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(*b*) We take both gases as the system. No work or mass crosses the system boundary, therefore this is a closed system with no work interactions. Then the energy balance for this closed system reduces to

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$
$$Q_{\text{in}} = \Delta U = \Delta U_{\text{He}} + \Delta U_{\text{O}_2}$$

He:

$$\Delta U_{\text{He}} = mc_{\nu}(T_m - T_1) = (4\text{ kg})(3.1156\text{ kJ/kg}\cdot\text{K})(220-170)\text{K} = 623.1\text{ kJ}$$

O_2:

$$\left.\begin{aligned} T_{R_1} &= 1.10 \\ P_{R_1} &= 1.38 \end{aligned}\right\} Z_{h_1} = 2.2$$

$$\left.\begin{aligned} T_{R_2} &= 1.42 \\ P_{R_2} &= \frac{9.97}{5.08} = 1.963 \end{aligned}\right\} Z_{h_2} = 1.2 \qquad \text{(Fig. A-29)}$$

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= R_u T_{\text{cr}}(Z_{h_1} - Z_{h_2}) + (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} \\ &= (8.314\text{ kJ/kmol}\cdot\text{K})(154.8\text{ K})(2.2-1.2) + (6404-4949)\text{kJ/kmol} \\ &= 2742\text{ kJ/kmol} \end{aligned}$$

Also,

$$P_{\text{He},1} = \frac{N_{\text{He}} R_u T_1}{V_{\text{tank}}} = \frac{(1\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(170\text{ K})}{0.229\text{ m}^3} = 6{,}172\text{ kPa}$$
$$P_{\text{O}_2,1} = P_{m,1} - P_{\text{He},1} = 7000\text{ kPa} - 6172\text{ kPa} = 828\text{ kPa}$$

Thus,

$$\begin{aligned} \Delta U_{\text{O}_2} &= N_{\text{O}_2}(\bar{h}_2 - \bar{h}_1) - (P_2 V_2 - P_1 V_1) = N_{\text{O}_2}(\bar{h}_2 - \bar{h}_1) - (P_{\text{O}_2,2} - P_{\text{O}_2,1})V_{\text{tank}} \\ &= (0.25\text{ kmol})(2742\text{ kJ/kmol}) - (1981-828)(0.229)\text{kPa}\cdot\text{m}^3 \\ &= 421.5\text{ kJ} \end{aligned}$$

Substituting,

$$Q_{\text{in}} = 623.1\text{ kJ} + 421.5\text{ kJ} = \mathbf{1045\text{ kJ}}$$

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13-100 A mixture of carbon dioxide and methane expands through a turbine. The power produced by the mixture is to be determined using ideal gas approximation and Kay's rule.

Assumptions The expansion process is reversible and adiabatic (isentropic).

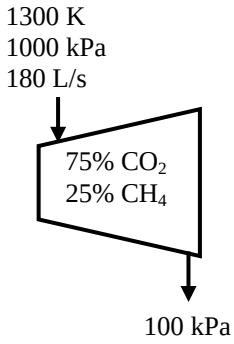
Properties The molar masses of CO_2 and CH_4 are 44.0 and 16.0 kg/kmol and respectively. The critical properties are 304.2 K, 7390 kPa for CO_2 and 191.1 K and 4640 kPa for CH_4 (Table A-1). EES may use slightly different values.

Analysis The molar mass of the mixture is determined to be

$$M_m = y_{CO_2} M_{CO_2} + y_{CH_4} M_{CH_4} = (0.75)(44) + (0.25)(16) = 37.0 \text{ kg/kmol}$$

The gas constant is

$$R = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol.K}}{37.0 \text{ kg/kmol}} = 0.2246 \text{ kJ/kg.K}$$



The mass fractions are

$$\text{mf}_{CO_2} = y_{CO_2} \frac{M_{CO_2}}{M_m} = (0.75) \frac{44 \text{ kg/kmol}}{37.0 \text{ kg/kmol}} = 0.8917$$

$$\text{mf}_{CH_4} = y_{CH_4} \frac{M_{CH_4}}{M_m} = (0.25) \frac{16 \text{ kg/kmol}}{37.0 \text{ kg/kmol}} = 0.1083$$

Ideal gas solution:

Using Dalton's law to find partial pressures, the entropies at the initial state are determined from EES to be:

$$T = 1300 \text{ K},\ P = (0.75 \times 1000) = 750 \text{ kPa} \longrightarrow s_{CO_2,1} = 6.068 \text{ kJ/kg.K}$$

$$T = 1600 \text{ K},\ P = (0.25 \times 1000) = 250 \text{ kPa} \longrightarrow s_{CH_4,1} = 16.22 \text{ kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final temperature is not known. However, for an isentropic process, the entropy change is zero and the final temperature may be determined from

$$\Delta s_{\text{total}} = mf_{CO_2} \Delta s_{CO_2} + mf_{CH_4} \Delta s_{CH_4}$$
$$0 = mf_{CO_2} (s_{CO_2,2} - s_{CO_2,1}) + mf_{CH_4} (s_{CH_4,2} - s_{CH_4,1})$$

The solution is obtained using EES to be

$$T_2 = 947.1 \text{ K}$$

The initial and final enthalpies and the changes in enthalpy are (from EES)

$$T_1 = 1300 \text{ K} \longrightarrow \begin{matrix} h_{CO_2,1} = -7803 \text{ kJ/kg} \\ h_{CH_4,1} = -831 \text{ kJ/kg} \end{matrix} \qquad T_2 = 947.1 \text{ K} \longrightarrow \begin{matrix} h_{CO_2,2} = -8248 \text{ kJ/kg} \\ h_{CH_4,2} = -2503 \text{ kJ/kg} \end{matrix}$$

Noting that the heat transfer is zero, an energy balance on the system gives

$$\dot{Q}_{\text{in}} - \dot{W}_{\text{out}} = \dot{m} \Delta h_m \longrightarrow \dot{W}_{\text{out}} = -\dot{m} \Delta h_m$$

where

$$\Delta h_m = \text{mf}_{CO_2} (h_{CO_2,2} - h_{CO_2,1}) + \text{mf}_{CH_4} (h_{CH_4,2} - h_{CH_4,1})$$
$$= (0.8917)\big[(-8248) - (-7803)\big] + (0.1083)\big[(-2503) - (-831)\big] = -577.7 \text{ kJ/kg}$$

The mass flow rate is

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(1000 \text{ kPa})(0.180 \text{ m}^3/\text{s})}{(0.2246 \text{ kJ/kg.K})(1300 \text{ K})} = 0.6165 \text{ kg/s}$$

Substituting, $\dot{W}_{\text{out}} = \dot{m} \Delta h_m = -(0.6165)(-577.7 \text{ kJ/kg}) = \mathbf{356 \text{ kW}}$

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Kay's rule solution:

The critical temperature and pressure of the mixture is

$$T_{cr} = y_{CO_2} T_{cr,CO_2} + y_{CH_4} T_{cr,CH_4} = (0.75)(304.2\text{ K}) + (0.25)(191.1\text{ K}) = 276\text{ K}$$
$$P_{cr} = y_{CO_2} P_{cr,CO_2} + y_{CH_4} P_{cr,CH_4} = (0.75)(7390\text{ kPa}) + (0.25)(4640\text{ kPa}) = 6683\text{ kPa}$$

State 1 properties:

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{1300\text{ K}}{276\text{ K}} = 4.715 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{1000\text{ kPa}}{6683\text{ kPa}} = 0.1496 \end{aligned}\right\} \begin{aligned} Z_1 &= 1.003 \\ Z_{h1} &= -0.0064 \\ Z_{s1} &= 0.001197 \end{aligned} \quad \text{(from EES)}$$

$$\Delta h_1 = Z_{h1} R T_{cr} = (-0.0064)(0.2246\text{ kJ/kg.K})(276\text{ K}) = -0.399\text{ kJ/kg}$$

$$h_1 = mf_{CO_2} h_{CO_2,1} + mf_{CH_4} h_{CH_4,1} - \Delta h_1$$
$$= (0.8917)(-7803) + (0.1083)(-831) - (-0.399) = -7047\text{ kJ/kg}$$
$$\Delta s_1 = Z_{s1} R = (0.001197)(0.2246\text{ kJ/kg.K}) = 0.0002688\text{ kJ/kg.K}$$

$$s_1 = \text{mf}_{CO_2} s_{CO_2,1} + \text{mf}_{CH_4} s_{CH_4,1} - \Delta s_1$$
$$= (0.8917)(6.068) + (0.1083)(16.22) - (0.0002688) = 7.1679\text{ kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final temperature is not known. However, for an isentropic process, the entropy change is zero and the final temperature may be determined from

$$\Delta s_{total} = \text{mf}_{CO_2} \Delta s_{CO_2} + \text{mf}_{CH_4} \Delta s_{CH_4}$$
$$0 = \text{mf}_{CO_2} (s_{CO_2,2} - s_{CO_2,1}) + \text{mf}_{CH_4} (s_{CH_4,2} - s_{CH_4,1})$$

The solution is obtained using EES to be

$$T_2 = 947\text{ K}$$

The initial and final enthalpies and the changes in enthalpy are

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{cr}} = \frac{947\text{ K}}{276\text{ K}} = 3.434 \\ P_{R2} &= \frac{P_2}{P_{cr}} = \frac{100\text{ kPa}}{6683\text{ kPa}} = 0.015 \end{aligned}\right\} \begin{aligned} Z_{h2} &= -0.0004869 \\ Z_{s2} &= 0.0004057 \end{aligned} \quad \text{(from EES)}$$

$$\Delta h_2 = Z_{h2} R T_{cr} = (-0.0004869)(0.2246\text{ kJ/kg.K})(276\text{ K}) = -0.03015\text{ kJ/kg}$$

$$h_2 = \text{mf}_{CO_2} h_{CO_2,2} + \text{mf}_{CH_4} h_{CH_4,2} - \Delta h_2$$
$$= (0.8917)(-8248) + (0.1083)(-2503) - (-0.03015) = -7625\text{ kJ/kg}$$

Noting that the heat transfer is zero, an energy balance on the system gives

$$\dot{Q}_{in} - \dot{W}_{out} = \dot{m}\Delta h_m \longrightarrow \dot{W}_{out} = -\dot{m}(h_2 - h_1)$$

where the mass flow rate is

$$\dot{m} = \frac{P_1 \dot{V}_1}{Z_1 R T_1} = \frac{(1000\text{ kPa})(0.180\text{ m}^3/\text{s})}{(1.003)(0.2246\text{ kJ/kg.K})(1300\text{ K})} = 0.6149\text{ kg/s}$$

Substituting,

$$\dot{W}_{out} = -(0.6149\text{ kg/s})\left[(-7625) - (-7047)\text{ kJ/kg}\right] = \mathbf{356\ kW}$$

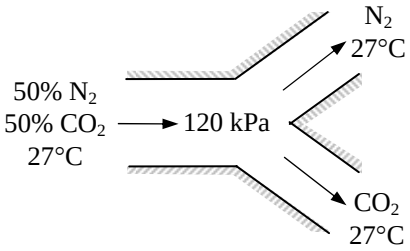
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13-101 A stream of gas mixture at a given pressure and temperature is to be separated into its constituents steadily. The minimum work required is to be determined.

Assumptions **1** Both the N_2 and CO_2 gases and their mixture are ideal gases. **2** This is a steady-flow process. **3** The kinetic and potential energy changes are negligible.

Properties The molar masses of N_2 and CO_2 are 28.0 and 44.0 kg/kmol. (Table A-1).

Analysis The minimum work required to separate a gas mixture into its components is equal to the reversible work associated with the mixing process, which is equal to the exergy destruction (or irreversibility) associated with the mixing process since



$$X_{\text{destroyed}} = W_{\text{rev,out}} - \cancel{W_{\text{act},u}}^{\,0} = W_{\text{rev,out}} = T_0 S_{\text{gen}}$$

where S_{gen} is the entropy generation associated with the steady-flow mixing process. The entropy change associated with a constant pressure and temperature adiabatic mixing process is determined from

$$\begin{aligned}\bar{s}_{\text{gen}} &= \sum \Delta \bar{s}_i = -R_u \sum y_i \ln y_i = -(8.314\ \text{kJ/kmol}\cdot\text{K})[0.5\ln(0.5) + 0.5\ln(0.5)] \\ &= 5.763\ \text{kJ/kmol}\cdot\text{K}\end{aligned}$$

$$M_m = \sum y_i M_i = (0.5)(28\ \text{kg/kmol}) + (0.5)(44\ \text{kg/kmol}) = 36\ \text{kg/kmol}$$

$$s_{\text{gen}} = \frac{\bar{s}_{\text{gen}}}{M_m} = \frac{5.763\ \text{kJ/kmol}\cdot\text{K}}{36\ \text{kg/kmol}} = 0.1601\ \text{kJ/kg}\cdot\text{K}$$

$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = (300\ \text{K})(0.1601\ \text{kJ/kg}\cdot\text{K}) = \mathbf{48.0\ kJ/kg}$$

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13-102E The mass percentages of a gas mixture are given. This mixture is expanded in an adiabatic, steady-flow turbine of specified isentropic efficiency. The second law efficiency and the exergy destruction during this expansion process are to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

Properties The molar masses of N_2, He, CH_4, and C_2H_6 are 28.0, 4.0, 16.0, and 30.0 lbm/lbmol, respectively (Table A-1E). The constant-pressure specific heats of these gases at room temperature are 0.248, 1.25, 0.532, and 0.427 Btu/lbm·R, respectively (Table A-2Ea).

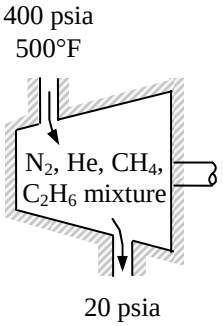
Analysis For 1 lbm of mixture, the mole numbers of each component are

$$N_{N2} = \frac{m_{N2}}{M_{N2}} = \frac{0.15\,\text{lbm}}{28\,\text{lbm/lbmol}} = 0.005357\,\text{lbmol}$$

$$N_{N2} = \frac{m_{He}}{M_{He}} = \frac{0.05\,\text{lbm}}{4\,\text{lbm/lbmol}} = 0.0125\,\text{lbmol}$$

$$N_{N2} = \frac{m_{CH4}}{M_{Ch4}} = \frac{0.6\,\text{lbm}}{16\,\text{lbm/lbmol}} = 0.0375\,\text{lbmol}$$

$$N_{N2} = \frac{m_{C2H6}}{M_{C2H6}} = \frac{0.20\,\text{lbm}}{30\,\text{lbm/lbmol}} = 0.006667\,\text{lbmol}$$



The mole number of the mixture is

$$N_m = N_{O2} + N_{CO2} + N_{He} = 0.005357 + 0.0125 + 0.0375 + 0.006667 = 0.06202\,\text{lbmol}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{1\,\text{lbm}}{0.065202\,\text{lbmol}} = 16.12\,\text{lbm/lbmol}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{1.9858\,\text{lbm/lbmol}\cdot\text{R}}{16.12\,\text{lbm/lbmol}} = 0.1232\,\text{Btu/lbm}\cdot\text{R}$$

The constant-pressure specific heat of the mixture is determined from

$$\begin{aligned} c_p &= \text{mf}_{N2}c_{p,N2} + \text{mf}_{He}c_{p,He} + \text{mf}_{CH4}c_{p,CH4} + \text{mf}_{C2H6}c_{p,C2H6} \\ &= 0.15\times 0.248 + 0.05\times 1.25 + 0.60\times 0.532 + 0.20\times 0.427 \\ &= 0.5043\,\text{Btu/lbm}\cdot\text{R} \end{aligned}$$

Then the constant-volume specific heat is

$$c_v = c_p - R = 0.5043 - 0.1232 = 0.3811\,\text{Btu/lbm}\cdot\text{R}$$

The specific heat ratio is

$$k = \frac{c_p}{c_v} = \frac{0.5043}{0.3811} = 1.323$$

The temperature at the end of the expansion for the isentropic process is

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (960\,\text{R})\left(\frac{20\,\text{psia}}{400\,\text{psia}}\right)^{0.323/1.323} = 462.0\,\text{R}$$

Using the definition of turbine isentropic efficiency, the actual outlet temperature is

$$T_2 = T_1 - \eta_{turb}(T_1 - T_{2s}) = (960\,\text{R}) - (0.85)(960 - 462.0) = 536.7\,\text{R}$$

The entropy change of the gas mixture is

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (0.5043)\ln\frac{536.7}{960} - (0.1232)\ln\frac{20}{400} = 0.07583\,\text{Btu/lbm}\cdot\text{R}$$

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The actual work produced is

$$w_{\text{out}} = h_1 - h_2 = c_p(T_1 - T_2) = (0.5043\ \text{Btu/lbm} \cdot \text{R})(960 - 536.7)\ \text{R} = 213.5\ \text{Btu/lbm}$$

The reversible work output is

$$w_{\text{rev,out}} = h_1 - h_2 - T_0(s_1 - s_2) = 213.5\ \text{Btu/lbm} - (537\ \text{R})(-0.07583\ \text{Btu/lbm} \cdot \text{R}) = 254.2\ \text{Btu/lbm}$$

The second-law efficiency and the exergy destruction are then

$$\eta_{\text{II}} = \frac{w_{\text{out}}}{w_{\text{rev,out}}} = \frac{213.5}{254.2} = 0.840 = \mathbf{84.0\%}$$

$$x_{\text{dest}} = w_{\text{rev,out}} - w_{\text{out}} = 254.2 - 213.5 = \mathbf{40.7\ Btu/lbm}$$

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13-103 A program is to be written to determine the mole fractions of the components of a mixture of three gases with known molar masses when the mass fractions are given, and to determine the mass fractions of the components when the mole fractions are given. Also, the program is to be run for a sample case.

Analysis The problem is solved using EES, and the solution is given below.

```
Procedure Fractions(Type$,A$,B$,C$,A,B,C:mf_A,mf_B,mf_C,y_A,y_B,y_C)
{If  Type$  <>  ('mass fraction'  OR  'mole fraction' ) then
Call ERROR('Type$ must be set equal to "mass fraction" or "mole fraction".')
GOTO 10
endif}
Sum = A+B+C
If ABS(Sum - 1) > 0 then goto 20
MM_A = molarmass(A$)
MM_B = molarmass(B$)
MM_C = molarmass(C$)
If Type$ = 'mass fraction' then
mf_A = A
mf_B = B
mf_C = C
sumM_mix = mf_A/MM_A+ mf_B/MM_B+ mf_C/MM_C
y_A = mf_A/MM_A/sumM_mix
y_B = mf_B/MM_B/sumM_mix
y_C = mf_C/MM_C/sumM_mix
GOTO 10
endif
if Type$ = 'mole fraction' then
y_A = A
y_B = B
y_C = C
MM_mix = y_A*MM_A+ y_B*MM_B+ y_C*MM_C
mf_A = y_A*MM_A/MM_mix
mf_B = y_B*MM_B/MM_mix
mf_C = y_C*MM_C/MM_mix
GOTO 10
Endif
Call ERROR('Type$ must be either mass fraction or mole fraction.')
GOTO 10
20:
Call ERROR('The sum of the mass or mole fractions must be 1')
10:
END

"Either the mole fraction y_i or the mass fraction mf_i may be given by setting the parameter Type$='mole fraction' when the
mole fractions are given or Type$='mass fraction' is given"
{Input Data in the Diagram Window}
{Type$='mole fraction'
A$ = 'N2'
B$ = 'O2'
C$ = 'Argon'
A = 0.71  "When Type$='mole fraction' A, B, C are the mole fractions"
B = 0.28  "When Type$='mass fraction' A, B, C are the mass fractions"
C = 0.01}
Call Fractions(Type$,A$,B$,C$,A,B,C:mf_A,mf_B,mf_C,y_A,y_B,y_C)
```

SOLUTION

A=0.71	A$='N2'	B=0.28	B$='O2'
C=0.01	C$='Argon'	mf_A=0.680	mf_B=0.306
mf_C=0.014	Type$='mole fraction'		y_A=0.710
y_B=0.280	y_C=0.010		

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13-104 A program is to be written to determine the entropy change of a mixture of 3 ideal gases when the mole fractions and other properties of the constituent gases are given. Also, the program is to be run for a sample case.

Analysis The problem is solved using EES, and the solution is given below.

```
T1=300 [K]
T2=600 [K]
P1=100 [kPa]
P2=500 [kPa]
A$ = 'N2'
B$ = 'O2'
C$ = 'Argon'
y_A = 0.71
y_B = 0.28
y_C = 0.01
MM_A = molarmass(A$)
MM_B = molarmass(B$)
MM_C = molarmass(C$)
MM_mix = y_A*MM_A+ y_B*MM_B+ y_C*MM_C
mf_A = y_A*MM_A/MM_mix
mf_B = y_B*MM_B/MM_mix
mf_C = y_C*MM_C/MM_mix
DELTAs_mix=mf_A*(entropy(A$,T=T2,P=y_B*P2)-
entropy(A$,T=T1,P=y_A*P1))+mf_B*(entropy(B$,T=T2,P=y_B*P2)-
entropy(B$,T=T1,P=y_B*P1))+mf_C*(entropy(C$,T=T2,P=y_C*P2)-entropy(C$,T=T1,P=y_C*P1))
```

SOLUTION

```
A$='N2'
B$='O2'
C$='Argon'
DELTAs_mix=12.41 [kJ/kg-K]
mf_A=0.68
mf_B=0.3063
mf_C=0.01366
MM_A=28.01 [kg/kmol]
MM_B=32 [kg/kmol]
MM_C=39.95 [kg/kmol]
MM_mix=29.25 [kJ/kmol]
P1=100 [kPa]
P2=500 [kPa]
T1=300 [K]
T2=600 [K]
y_A=0.71
y_B=0.28
y_C=0.01
```

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Fundamentals of Engineering (FE) Exam Problems

13-105 An ideal gas mixture whose apparent molar mass is 20 kg/kmol consists of nitrogen N_2 and three other gases. If the mole fraction of nitrogen is 0.55, its mass fraction is

(a) 0.15 (b) 0.23 (c) 0.39 (d) 0.55 (e) 0.77

Answer (e) 0.77

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
M_mix=20 "kg/kmol"
M_N2=28 "kg/kmol"
y_N2=0.55
mf_N2=(M_N2/M_mix)*y_N2

"Some Wrong Solutions with Common Mistakes:"
W1_mf = y_N2 "Taking mass fraction to be equal to mole fraction"
W2_mf= y_N2*(M_mix/M_N2) "Using the molar mass ratio backwords"
W3_mf= 1-mf_N2 "Taking the complement of the mass fraction"
```

13-106 An ideal gas mixture consists of 2 kmol of N_2 and 6 kmol of CO_2. The mass fraction of CO_2 in the mixture is

(a) 0.175 (b) 0.250 (c) 0.500 (d) 0.750 (e) 0.825

Answer (e) 0.825

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
N1=2 "kmol"
N2=6 "kmol"
N_mix=N1+N2
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
m_mix=N1*MM1+N2*MM2
mf2=N2*MM2/m_mix

"Some Wrong Solutions with Common Mistakes:"
W1_mf = N2/N_mix "Using mole fraction"
W2_mf = 1-mf2 "The wrong mass fraction"
```

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13-107 An ideal gas mixture consists of 2 kmol of N_2 and 4 kmol of CO_2. The apparent gas constant of the mixture is

(a) 0.215 kJ/kg·K (b) 0.225 kJ/kg·K (c) 0.243 kJ/kg·K (d) 0.875 kJ/kg·K (e) 1.24 kJ/kg·K

Answer (a) 0.215 kJ/kg·K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Ru=8.314 "kJ/kmol.K"
N1=2 "kmol"
N2=4 "kmol"
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
R1=Ru/MM1
R2=Ru/MM2
N_mix=N1+N2
y1=N1/N_mix
y2=N2/N_mix
MM_mix=y1*MM1+y2*MM2
R_mix=Ru/MM_mix

"Some Wrong Solutions with Common Mistakes:"
W1_Rmix =(R1+R2)/2 "Taking the arithmetic average of gas constants"
W2_Rmix= y1*R1+y2*R2 "Using wrong relation for Rmixture"
```

13-108 A rigid tank is divided into two compartments by a partition. One compartment contains 3 kmol of N_2 at 400 kPa pressure and the other compartment contains 7 kmol of CO_2 at 200 kPa. Now the partition is removed, and the two gases form a homogeneous mixture at 250 kPa. The partial pressure of N_2 in the mixture is

(a) 75 kPa (b) 90 kPa (c) 125 kPa (d) 175 kPa (e) 250 kPa

Answer (a) 75 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1 = 400 "kPa"
P2 = 200 "kPa"
P_mix=250 "kPa"
N1=3 "kmol"
N2=7 "kmol"
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
N_mix=N1+N2
y1=N1/N_mix
y2=N2/N_mix
P_N2=y1*P_mix

"Some Wrong Solutions with Common Mistakes:"
W1_P1= P_mix/2 "Assuming equal partial pressures"
W2_P1= mf1*P_mix; mf1=N1*MM1/(N1*MM1+N2*MM2) "Using mass fractions"
W3_P1 = P_mix*N1*P1/(N1*P1+N2*P2) "Using some kind of weighed averaging"
```

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13-109 An 80-L rigid tank contains an ideal gas mixture of 5 g of N_2 and 5 g of CO_2 at a specified pressure and temperature. If N_2 were separated from the mixture and stored at mixture temperature and pressure, its volume would be

(a) 32 L (b) 36 L (c) 40 L (d) 49 L (e) 80 L

Answer (d) 49 L

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_mix=80 "L"
m1=5 "g"
m2=5 "g"
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
N1=m1/MM1
N2=m2/MM2
N_mix=N1+N2
y1=N1/N_mix
V1=y1*V_mix  "L"

"Some Wrong Solutions with Common Mistakes:"
W1_V1=V_mix*m1/(m1+m2)  "Using mass fractions"
W2_V1= V_mix "Assuming the volume to be the mixture volume"
```

13-110 An ideal gas mixture consists of 3 kg of Ar and 6 kg of CO_2 gases. The mixture is now heated at constant volume from 250 K to 350 K. The amount of heat transfer is

(a) 374 kJ (b) 436 kJ (c) 488 kJ (d) 525 kJ (e) 664 kJ

Answer (c) 488 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=250 "K"
T2=350 "K"
Cv1=0.3122; Cp1=0.5203 "kJ/kg.K"
Cv2=0.657; Cp2=0.846 "kJ/kg.K"
m1=3 "kg"
m2=6 "kg"
MM1=39.95 "kg/kmol"
MM2=44 "kg/kmol"
"Applying Energy balance gives Q=DeltaU=DeltaU_Ar+DeltaU_CO2"
Q=(m1*Cv1+m2*Cv2)*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
W1_Q = (m1+m2)*(Cv1+Cv2)/2*(T2-T1)  "Using arithmetic average of properties"
W2_Q = (m1*Cp1+m2*Cp2)*(T2-T1)"Using Cp instead of Cv"
W3_Q = (m1*Cv1+m2*Cv2)*T2 "Using T2 instead of T2-T1"
```

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13-111 An ideal gas mixture consists of 60% helium and 40% argon gases by mass. The mixture is now expanded isentropically in a turbine from 400°C and 1.2 MPa to a pressure of 200 kPa. The mixture temperature at turbine exit is

(a) 56°C (b) 195°C (c) 130°C (d) 112°C (e) 400°C

Answer (a) 56°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=400+273"K"
P1=1200 "kPa"
P2=200 "kPa"
mf_He=0.6
mf_Ar=0.4
k1=1.667
k2=1.667
"The specific heat ratio k of the mixture is also 1.667 since k=1.667 for all componet gases"
k_mix=1.667
T2=T1*(P2/P1)^((k_mix-1)/k_mix)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T2 = (T1-273)*(P2/P1)^((k_mix-1)/k_mix) "Using C for T1 instead of K"
W2_T2 = T1*(P2/P1)^((k_air-1)/k_air)-273; k_air=1.4 "Using k value for air"
W3_T2 = T1*P2/P1 "Assuming T to be proportional to P"
```

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13-112 One compartment of an insulated rigid tank contains 2 kmol of CO_2 at 20°C and 150 kPa while the other compartment contains 5 kmol of H_2 gas at 35°C and 300 kPa. Now the partition between the two gases is removed, and the two gases form a homogeneous ideal gas mixture. The temperature of the mixture is

(a) 25°C (b) 29°C (c) 22°C (d) 32°C (e) 34°C

Answer (b) 29°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
N_H2=5 "kmol"
T1_H2=35 "C"
P1_H2=300 "kPa"
N_CO2=2 "kmol"
T1_CO2=20 "C"
P1_CO2=150 "kPa"
Cv_H2=10.183; Cp_H2=14.307 "kJ/kg.K"
Cv_CO2=0.657; Cp_CO2=0.846 "kJ/kg.K"
MM_H2=2 "kg/kmol"
MM_CO2=44 "kg/kmol"
m_H2=N_H2*MM_H2
m_CO2=N_CO2*MM_CO2
"Applying Energy balance gives 0=DeltaU=DeltaU_H2+DeltaU_CO2"
0=m_H2*Cv_H2*(T2-T1_H2)+m_CO2*Cv_CO2*(T2-T1_CO2)

"Some Wrong Solutions with Common Mistakes:"
0=m_H2*Cp_H2*(W1_T2-T1_H2)+m_CO2*Cp_CO2*(W1_T2-T1_CO2) "Using Cp instead of Cv"
0=N_H2*Cv_H2*(W2_T2-T1_H2)+N_CO2*Cv_CO2*(W2_T2-T1_CO2) "Using N instead of mass"
W3_T2 = (T1_H2+T1_CO2)/2 "Assuming averate temperature"
```

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13-113 A piston-cylinder device contains an ideal gas mixture of 3 kmol of He gas and 7 kmol of Ar gas at 50°C and 400 kPa. Now the gas expands at constant pressure until its volume doubles. The amount of heat transfer to the gas mixture is

(a) 6.2 MJ (b) 42 MJ (c) 27 MJ (d) 10 MJ (e) 67 MJ

Answer (e) 67 MJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
N_He=3 "kmol"
N_Ar=7 "kmol"
T1=50+273 "C"
P1=400 "kPa"
P2=P1
"T2=2T1 since PV/T=const for ideal gases and it is given that P=constant"
T2=2*T1 "K"
MM_He=4 "kg/kmol"
MM_Ar=39.95 "kg/kmol"
m_He=N_He*MM_He
m_Ar=N_Ar*MM_Ar
Cp_Ar=0.5203; Cv_Ar = 3122 "kJ/kg.C"
Cp_He=5.1926; Cv_He = 3.1156 "kJ/kg.K"
"For a P=const process, Q=DeltaH since DeltaU+Wb is DeltaH"
Q=m_Ar*Cp_Ar*(T2-T1)+m_He*Cp_He*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
W1_Q =m_Ar*Cv_Ar*(T2-T1)+m_He*Cv_He*(T2-T1) "Using Cv instead of Cp"
W2_Q=N_Ar*Cp_Ar*(T2-T1)+N_He*Cp_He*(T2-T1) "Using N instead of mass"
W3_Q=m_Ar*Cp_Ar*(T22-T1)+m_He*Cp_He*(T22-T1); T22=2*(T1-273)+273 "Using C for T1"
W4_Q=(m_Ar+m_He)*0.5*(Cp_Ar+Cp_He)*(T2-T1) "Using arithmetic averate of Cp"
```

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13-114 An ideal gas mixture of helium and argon gases with identical mass fractions enters a turbine at 1500 K and 1 MPa at a rate of 0.12 kg/s, and expands isentropically to 100 kPa. The power output of the turbine is

(a) 253 kW (b) 310 kW (c) 341 kW (d) 463 kW (e) 550 kW

Answer (b) 310 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=0.12 "kg/s"
T1=1500 "K"
P1=1000 "kPa"
P2=100 "kPa"
mf_He=0.5
mf_Ar=0.5
k_He=1.667
k_Ar=1.667
Cp_Ar=0.5203
Cp_He=5.1926
Cp_mix=mf_He*Cp_He+mf_Ar*Cp_Ar
"The specific heat ratio k of the mixture is also 1.667 since k=1.667 for all componet gases"
k_mix=1.667
T2=T1*(P2/P1)^((k_mix-1)/k_mix)
-W_out=m*Cp_mix*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
W1_Wout= - m*Cp_mix*(T22-T1); T22 = (T1-273)*(P2/P1)^((k_mix-1)/k_mix)+273 "Using C for T1 instead of K"
W2_Wout= - m*Cp_mix*(T222-T1); T222 = T1*(P2/P1)^((k_air-1)/k_air)-273; k_air=1.4 "Using k value for air"
W3_Wout= - m*Cp_mix*(T2222-T1); T2222 = T1*P2/P1 "Assuming T to be proportional to P"
W4_Wout= - m*0.5*(Cp_Ar+Cp_He)*(T2-T1) "Using arithmetic average for Cp"
```

13-115 … 13-118 Design and Essay Problem

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 14
GAS-VAPOR MIXTURES AND AIR CONDITIONING

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Dry and Atmospheric Air: Specific and Relative Humidity

14-1C Dry air does not contain any water vapor, but atmospheric air does.

14-2C Yes; by cooling the air at constant pressure.

14-3C Yes.

14-4C Specific humidity will decrease but relative humidity will increase.

14-5C Yes, the water vapor in the air can be treated as an ideal gas because of its very low partial pressure.

14-6C The same. This is because water vapor behaves as an ideal gas at low pressures, and the enthalpy of an ideal gas depends on temperature only.

14-7C Specific humidity is the amount of water vapor present in a unit mass of dry air. Relative humidity is the ratio of the actual amount of vapor in the air at a given temperature to the maximum amount of vapor air can hold at that temperature.

14-8C The specific humidity will remain constant, but the relative humidity will decrease as the temperature rises in a well-sealed room.

14-9C The specific humidity will remain constant, but the relative humidity will decrease as the temperature drops in a well-sealed room.

14-10C A tank that contains moist air at 3 atm is located in moist air that is at 1 atm. The driving force for moisture transfer is the vapor pressure difference, and thus it is possible for the water vapor to flow into the tank from surroundings if the vapor pressure in the surroundings is greater than the vapor pressure in the tank.

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14-11C Insulations on *chilled water lines* are always wrapped with *vapor barrier jackets* to eliminate the possibility of vapor entering the insulation. This is because moisture that migrates through the insulation to the cold surface will condense and remain there indefinitely with no possibility of vaporizing and moving back to the outside.

14-12C When the temperature, total pressure, and the relative humidity are given, the vapor pressure can be determined from the psychrometric chart or the relation $P_v = \phi P_{\text{sat}}$ where P_{sat} is the saturation (or boiling) pressure of water at the specified temperature and ϕ is the relative humidity.

14-13 A tank contains dry air and water vapor at specified conditions. The specific humidity, the relative humidity, and the volume of the tank are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The specific humidity can be determined form its definition,

$$\omega = \frac{m_v}{m_a} = \frac{0.3\,\text{kg}}{21\,\text{kg}} = \mathbf{0.0143\ kg\ H_2O/kg\ dry\ air}$$

21 kg dry air
0.3 kg H_2O vapor
30°C
100 kPa

(*b*) The saturation pressure of water at 30°C is

$$P_g = P_{\text{sat @ 30°C}} = 4.2469\,\text{kPa}$$

Then the relative humidity can be determined from

$$\phi = \frac{\omega P}{(0.622 + \omega) P_g} = \frac{(0.0143)(100\ \text{kPa})}{(0.622 + 0.0143)(4.2469\ \text{kPa})} = \mathbf{52.9\%}$$

(*c*) The volume of the tank can be determined from the ideal gas relation for the dry air,

$$P_v = \phi P_g = (0.529)(4.2469\ \text{kPa}) = 2.245\,\text{kPa}$$

$$P_a = P - P_v = 100 - 2.245 = 97.755\,\text{kPa}$$

$$V = \frac{m_a R_a T}{P_a} = \frac{(21\,\text{kg})(0.287\,\text{kJ/kg}\cdot\text{K})(303\,\text{K})}{97.755\,\text{kPa}} = \mathbf{18.7\ m^3}$$

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14-14 A tank contains dry air and water vapor at specified conditions. The specific humidity, the relative humidity, and the volume of the tank are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The specific humidity can be determined form its definition,

$$\omega = \frac{m_v}{m_a} = \frac{0.3\,\text{kg}}{21\,\text{kg}} = \mathbf{0.0143\,kg\,H_2O/kg\,dry\,air}$$

21 kg dry air
0.3 kg H_2O vapor
20°C
100 kPa

(*b*) The saturation pressure of water at 24°C is

$$P_g = P_{\text{sat @20°C}} = 2.339\,\text{kPa}$$

Then the relative humidity can be determined from

$$\phi = \frac{\omega P}{(0.622 + \omega)P_g} = \frac{(0.0143)(100\,\text{kPa})}{(0.622 + 0.0143)2.339\,\text{kPa}} = 0.960 = \mathbf{96.0\%}$$

(*c*) The volume of the tank can be determined from the ideal gas relation for the dry air,

$$P_v = \phi P_g = (0.960)(2.339\,\text{kPa}) = 2.245\,\text{kPa}$$
$$P_a = P - P_v = 100 - 2.245 = 97.755\,\text{kPa}$$
$$V = \frac{m_a R_a T}{P_a} = \frac{(21\,\text{kg})(0.287\,\text{kJ/kg}\cdot\text{K})(293\,\text{K})}{97.755\,\text{kPa}} = \mathbf{18.1\,m^3}$$

14-15 A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The partial pressure of dry air can be determined from

$$P_v = \phi P_g = \phi P_{\text{sat @ 20°C}} = (0.85)(2.3392\,\text{kPa}) = 1.988\,\text{kPa}$$
$$P_a = P - P_v = 98 - 1.988 = \mathbf{96.01\,kPa}$$

AIR
20°C
98 kPa
85% RH

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(1.988\,\text{kPa})}{(98 - 1.988)\,\text{kPa}} = \mathbf{0.0129\,kg\,H_2O/kg\,dry\,air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$
$$= (1.005\,\text{kJ/kg}\cdot\text{°C})(20\text{°C}) + (0.0129)(2537.4\,\text{kJ/kg})$$
$$= \mathbf{52.78\,kJ/kg\,dry\,air}$$

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14-16 A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The partial pressure of dry air can be determined from

$$P_v = \phi P_g = \phi P_{\text{sat @ 20°C}} = (0.85)(2.3392\ \text{kPa}) = 1.988\ \text{kPa}$$
$$P_a = P - P_v = 85 - 1.988 = \mathbf{83.01\ kPa}$$

AIR
20°C
85 kPa
85% RH

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(1.988\ \text{kPa})}{(85 - 1.988)\ \text{kPa}} = \mathbf{0.0149\ kg\ H_2O/kg\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$
$$= (1.005\ \text{kJ/kg} \cdot \text{°C})(20\text{°C}) + (0.0149)(2537.4\ \text{kJ/kg})$$
$$= \mathbf{57.90\ kJ/kg\ dry\ air}$$

14-17E A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The partial pressure of dry air can be determined from

$$P_v = \phi P_g = \phi P_{\text{sat @ 85°F}} = (0.60)(0.5966\ \text{psia}) = 0.358\ \text{psia}$$
$$P_a = P - P_v = 13.5 - 0.358 = \mathbf{13.14\ psia}$$

AIR
85°F
13.5 psia
60% RH

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(0.358\ \text{psia})}{(13.5 - 0.358)\ \text{psia}} = \mathbf{0.0169\ lbm\ H_2O/lbm\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$
$$= (0.24\ \text{Btu/lbm} \cdot \text{°F})(85\text{°F}) + (0.0169)(1098.3\ \text{Btu/lbm})$$
$$= \mathbf{39.01\ Btu/lbm\ dry\ air}$$

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14-18 The masses of dry air and the water vapor contained in a room at specified conditions and relative humidity are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis The partial pressure of water vapor and dry air are determined to be

$$P_v = \phi P_g = \phi P_{\text{sat @ 23°C}} = (0.50)(2.811\,\text{kPa}) = 1.41\,\text{kPa}$$
$$P_a = P - P_v = 98 - 1.41 = 96.59\,\text{kPa}$$

ROOM
240 m^3
23°C
98 kPa
50% RH

The masses are determined to be

$$m_a = \frac{P_a V}{R_a T} = \frac{(96.59\,\text{kPa})(240\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\,\text{K})} = \mathbf{272.9\,kg}$$
$$m_v = \frac{P_v V}{R_v T} = \frac{(1.41\,\text{kPa})(240\,\text{m}^3)}{(0.4615\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\,\text{K})} = \mathbf{2.47\,kg}$$

14-19E Humid air is expanded in an isentropic nozzle. The amount of water vapor that has condensed during the process is to be determined.

Assumptions The air and the water vapor are ideal gases.

Properties The specific heat ratio of air at room temperature is k = 1.4 (Table A-2a). The saturation properties of water are to be obtained from water tables.

Analysis Since the mole fraction of the water vapor in this mixture is very small,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (960\,\text{R})\left(\frac{15\,\text{psia}}{75\,\text{psia}}\right)^{0.4/1.4} = 606.1\,\text{R}$$

75 psia
500°F
ω_1=0.018
AIR
15 psia

We will assume that the air leaves the nozzle at a relative humidity of 100% (will be verified later). The vapor pressure and specific humidity at the outlet are then

$$P_{v,2} = \phi_2 P_{g,2} = \phi_2 P_{\text{sat @ 146.1°F}} = (1.0)(3.381\,\text{psia}) = 3.381\,\text{psia}$$

$$\omega_2 = \frac{0.622 P_{v,2}}{P - P_{v,2}} = \frac{(0.622)(3.381\,\text{psia})}{(15 - 3.381)\,\text{psia}} = 0.181\,\text{lbm H}_2\text{O/lbm dry air}$$

This is greater than the inlet specific humidity (0.018 lbm/lbm dry air), and thus there will be no condensation of water vapor:

$$\Delta\omega = \omega_1 - \omega_2 = \mathbf{0\,lbm\,H_2O/lbm\,dry\,air}$$

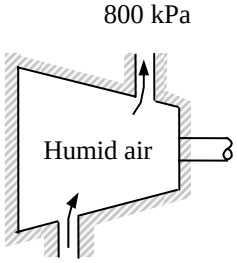
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14-20 Humid air is compressed in an isentropic compressor. The relative humidity of the air at the compressor outlet is to be determined.

Assumptions The air and the water vapor are ideal gases.

Properties The specific heat ratio of air at room temperature is $k = 1.4$ (Table A-2a). The saturation properties of water are to be obtained from water tables.

Analysis At the inlet,



$$P_{v,1} = \phi_1 P_{g,1} = \phi_1 P_{\text{sat @ 20°C}} = (0.90)(2.3392\,\text{kPa}) = 2.105\,\text{kPa}$$

$$\omega_2 = \omega_1 = \frac{0.622 P_{v,1}}{P - P_{v,1}} = \frac{(0.622)(2.105\,\text{kPa})}{(100 - 2.105)\,\text{kPa}} = 0.0134\,\text{kg H}_2\text{O/kg dry air}$$

Since the mole fraction of the water vapor in this mixture is very small,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (293\,\text{K})\left(\frac{800\,\text{kPa}}{100\,\text{kPa}}\right)^{0.4/1.4} = 531\,\text{K}$$

The saturation pressure at this temperature is

$$P_{g,2} = P_{\text{sat @ 258°C}} = 4542\,\text{kPa} \quad \text{(from EES)}$$

The vapor pressure at the exit is

$$P_{v,2} = \frac{\omega_2 P_2}{\omega_2 + 0.622} = \frac{(0.0134)(800)}{0.0134 + 0.622} = 16.87\,\text{kPa}$$

The relative humidity at the exit is then

$$\phi_2 = \frac{P_{v,2}}{P_{g,2}} = \frac{16.87}{4542} = \mathbf{0.0037} = \mathbf{0.37\%}$$

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Dew-Point, Adiabatic Saturation, and Wet-bulb Temperatures

14-21C Dew-point temperature is the temperature at which condensation begins when air is cooled at constant pressure.

14-22C Andy's. The temperature of his glasses may be below the dew-point temperature of the room, causing condensation on the surface of the glasses.

14-23C The outer surface temperature of the glass may drop below the dew-point temperature of the surrounding air, causing the moisture in the vicinity of the glass to condense. After a while, the condensate may start dripping down because of gravity.

14-24C When the temperature falls below the dew-point temperature, dew forms on the outer surfaces of the car. If the temperature is below 0°C, the dew will freeze. At very low temperatures, the moisture in the air will freeze directly on the car windows.

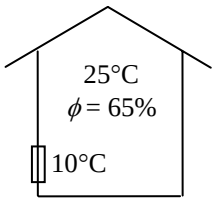
14-25C When the air is saturated (100% relative humidity).

14-26C These two are approximately equal at atmospheric temperatures and pressure.

14-27 A house contains air at a specified temperature and relative humidity. It is to be determined whether any moisture will condense on the inner surfaces of the windows when the temperature of the window drops to a specified value.

Assumptions The air and the water vapor are ideal gases.

Analysis The vapor pressure P_v is uniform throughout the house, and its value can be determined from



$$P_v = \phi P_{g\,@\,25°C} = (0.65)(3.1698\ \text{kPa}) = 2.06\ \text{kPa}$$

The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,2.06\ \text{kPa}} = \mathbf{18.0°C}$$

That is, the moisture in the house air will start condensing when the temperature drops below 18.0°C. Since the windows are at a lower temperature than the dew-point temperature, some moisture **will condense** on the window surfaces.

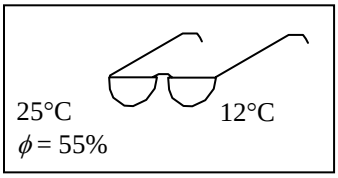
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14-28 A person wearing glasses enters a warm room at a specified temperature and relative humidity from the cold outdoors. It is to be determined whether the glasses will get fogged.

Assumptions The air and the water vapor are ideal gases.

Analysis The vapor pressure P_v of the air in the house is uniform throughout, and its value can be determined from

$$P_v = \phi P_{g\,@\,25°C} = (0.55)(3.1698\,\text{kPa}) = 1.743\,\text{kPa}$$



The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,1.743\,\text{kPa}} = \mathbf{15.3°C} \quad \text{(Table A-5 or EES)}$$

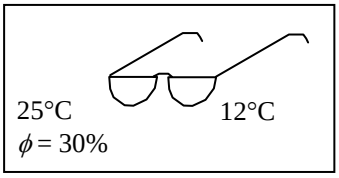
That is, the moisture in the house air will start condensing when the air temperature drops below 15.3°C. Since the glasses are at a lower temperature than the dew-point temperature, some moisture will condense on the glasses, and thus they **will get fogged**.

14-29 A person wearing glasses enters a warm room at a specified temperature and relative humidity from the cold outdoors. It is to be determined whether the glasses will get fogged.

Assumptions The air and the water vapor are ideal gases.

Analysis The vapor pressure P_v of the air in the house is uniform throughout, and its value can be determined from

$$P_v = \phi P_{g\,@\,25°C} = (0.30)(3.1698\,\text{kPa}) = 0.9509\,\text{kPa}$$



The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,0.9509\,\text{kPa}} = \mathbf{6.2°C} \quad \text{(Table A-5 or EES)}$$

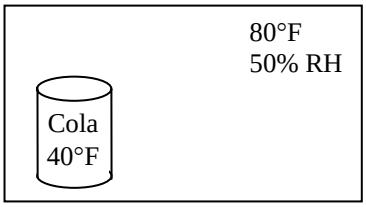
That is, the moisture in the house air will start condensing when the air temperature drops below 6.2°C. Since the glasses are at a higher temperature than the dew-point temperature, moisture will not condense on the glasses, and thus they **will not get fogged**.

14-30E A woman drinks a cool canned soda in a room at a specified temperature and relative humidity. It is to be determined whether the can will sweat.

Assumptions The air and the water vapor are ideal gases.

Analysis The vapor pressure P_v of the air in the house is uniform throughout, and its value can be determined from

$$P_v = \phi P_{g\,@\,80°F} = (0.50)(0.50745\,\text{psia}) = 0.254\,\text{psia}$$



The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,0.254\,\text{psia}} = \mathbf{59.7°F} \quad \text{(from EES)}$$

That is, the moisture in the house air will start condensing when the air temperature drops below 59.7°C. Since the canned drink is at a lower temperature than the dew-point temperature, some moisture will condense on the can, and thus it **will sweat.**

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14-31 The dry- and wet-bulb temperatures of atmospheric air at a specified pressure are given. The specific humidity, the relative humidity, and the enthalpy of air are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) We obtain the properties of water vapor from EES. The specific humidity ω_1 is determined from

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

95 kPa
25°C
T_{wb} = 17°C

where T_2 is the wet-bulb temperature, and ω_2 is determined from

$$\omega_2 = \frac{0.622P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(1.938\text{ kPa})}{(95 - 1.938)\text{ kPa}} = 0.01295\text{ kg H}_2\text{O/kg dry air}$$

Thus,

$$\omega_1 = \frac{(1.005\text{ kJ/kg}\cdot\text{°C})(17 - 25)\text{°C} + (0.01295)(2460.6\text{ kJ/kg})}{(2546.5 - 71.36)\text{ kJ/kg}} = \mathbf{0.00963\text{ kg H}_2\text{O/kg dry air}}$$

(*b*) The relative humidity ϕ_1 is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1)P_{g1}} = \frac{(0.00963)(95\text{ kPa})}{(0.622 + 0.00963)(3.1698\text{ kPa})} = 0.457 \quad \text{or} \quad \mathbf{45.7\%}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h_1 = h_{a1} + \omega_1 h_{v1} \cong c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot\text{°C})(25\text{°C}) + (0.00963)(2546.5\text{ kJ/kg})$$
$$= \mathbf{49.65\text{ kJ/kg dry air}}$$

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14-32 The dry- and wet-bulb temperatures of air in room at a specified pressure are given. The specific humidity, the relative humidity, and the dew-point temperature are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) We obtain the properties of water vapor from EES. The specific humidity ω_1 is determined from

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

where T_2 is the wet-bulb temperature, and ω_2 is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(2.488\ \text{kPa})}{(100 - 2.488)\ \text{kPa}} = 0.01587\ \text{kg H}_2\text{O/kg dry air}$$

100 kPa
26°C
T_{wb} = 21°C

Thus,

$$\omega_1 = \frac{(1.005\ \text{kJ/kg} \cdot °\text{C})(21 - 26)°\text{C} + (0.01587)(2451.2\ \text{kJ/kg})}{(2548.3 - 88.10)\ \text{kJ/kg}} = \mathbf{0.01377\ kg\ H_2O/kg\ dry\ air}$$

(*b*) The relative humidity ϕ_1 is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1) P_{g1}} = \frac{(0.01377)(100\ \text{kPa})}{(0.622 + 0.01377)(3.3638\ \text{kPa})} = 0.644 \quad \text{or} \quad \mathbf{64.4\%}$$

(*c*) The vapor pressure at the inlet conditions is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 26°C}} = (0.644)(3.3638\ \text{kPa}) = 2.166\ \text{kPa}$$

Thus the dew-point temperature of the air is

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 2.166 kPa}} = \mathbf{18.8°C}$$

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14-33 EES Problem 14-32 is reconsidered. The required properties are to be determined using EES at 100 and 300 kPa pressures.

Analysis The problem is solved using EES, and the solution is given below.

```
Tdb=26 [C]
Twb=21 [C]
P1=100 [kPa]
P2=300 [kPa]

h1=enthalpy(AirH2O,T=Tdb,P=P1,B=Twb)
v1=volume(AirH2O,T=Tdb,P=P1,B=Twb)
Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,B=Twb)
w1=humrat(AirH2O,T=Tdb,P=P1,B=Twb)
Rh1=relhum(AirH2O,T=Tdb,P=P1,B=Twb)

h2=enthalpy(AirH2O,T=Tdb,P=P2,B=Twb)
v2=volume(AirH2O,T=Tdb,P=P2,B=Twb)
Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,B=Twb)
w2=humrat(AirH2O,T=Tdb,P=P2,B=Twb)
Rh2=relhum(AirH2O,T=Tdb,P=P2,B=Twb)
```

SOLUTION

```
h1=61.25 [kJ/kg]
h2=34.16 [kJ/kg]
P1=100 [kPa]
P2=300 [kPa]
Rh1=0.6437
Rh2=0.4475
Tdb=26 [C]
Tdp1=18.76
Tdp2=13.07
Twb=21 [C]
v1=0.8777 [m^3/kg]
v2=0.2877 [m^3/kg]
w1=0.01376 [kg/kg]
w2=0.003136 [kg/kg]
```

Alternative Solution

The following EES routine can also be used to solve this problem. The above EES routine uses built-in psychrometric functions whereas the one blow uses analytical expressions together with steam properties.

```
"Given"
T_db=26 [C]
T_wb=21 [C]
P=100 [kPa]

"Properties"
Fluid$='steam_iapws'
P_g1=pressure(Fluid$, T=T_db, x=1)
P_g2=pressure(Fluid$, T=T_wb, x=1)
h_g1=enthalpy(Fluid$, T=T_db, x=1)
h_g2=enthalpy(Fluid$, T=T_wb, x=1)
h_f2=enthalpy(Fluid$, T=T_wb, x=0)
h_fg2=h_g2-h_f2
c_p=1.005 [kJ/kg-C] "for air"
```

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"Analysis"
"(a)"
w_2=(0.622*P_g2)/(P-P_g2) "kg H2O/kg dry air"
w_1=(c_p*(T_wb-T_db)+w_2*h_fg2)/(h_g1-h_f2)
"(b)"
phi_1=(w_1*P)/((0.622+w_1)*P_g1)
"(c)"
P_v1=phi_1*P_g1
T_dp=temperature(Fluid$, P=P_v1, x=1)

SOLUTION
c_p=1.005 [kJ/kg-C]
Fluid$='steam_iapws'
h_f2=88.1 [kJ/kg]
h_fg2=2451.2 [kJ/kg]
h_g1=2548.3 [kJ/kg]
h_g2=2539.3 [kJ/kg]
P=100 [kPa]
phi_1=0.6439
P_g1=3.3638 [kPa]
P_g2=2.488 [kPa]
P_v1=2.166 [kPa]
T_db=26 [C]
T_dp=18.8 [C]
T_wb=21 [C]
w_1=0.01377
w_2=0.01587

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14-34E The dry- and wet-bulb temperatures of air in room at a specified pressure are given. The specific humidity, the relative humidity, and the dew-point temperature are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The specific humidity ω_1 is determined from

14.7 psia
80°F
T_{wb} = 65°F

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

where T_2 is the wet-bulb temperature, and ω_2 is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(0.30578\,\text{psia})}{(14.7 - 0.30578)\,\text{psia}} = 0.01321\,\text{lbm H}_2\text{O/lbm dry air}$$

Thus,

$$\omega_1 = \frac{(0.24\,\text{Btu/lbm}\cdot°\text{F})(65 - 80)°\text{F} + (0.01321)(1056.5\,\text{Btu/lbm})}{(1096.1 - 33.08)\,\text{Btu/lbm}} = \mathbf{0.00974\,lbm\ H_2O/lbm\ dry\ air}$$

(*b*) The relative humidity ϕ_1 is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1) P_{g1}} = \frac{(0.00974)(14.7\,\text{psia})}{(0.622 + 0.00974)(0.50745\,\text{psia})} = 0.447 \quad \text{or} \quad \mathbf{44.7\%}$$

(*c*) The vapor pressure at the inlet conditions is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 70°F}} = (0.447)(0.50745\,\text{psia}) = 0.2268\,\text{psia}$$

Thus the dew-point temperature of the air is

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 0.2268 psia}} = \mathbf{56.6°F} \qquad \text{(from EES)}$$

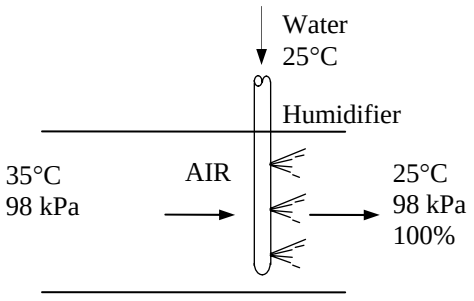
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14-35 Atmospheric air flows steadily into an adiabatic saturation device and leaves as a saturated vapor. The relative humidity and specific humidity of air are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The exit state of the air is completely specified, and the total pressure is 98 kPa. The properties of the moist air at the exit state may be determined from EES to be

$$h_2 = 78.11 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.02079 \text{ kg } H_2O\text{/kg dry air}$$



The enthalpy of makeup water is

$$h_{w2} = h_{f@\,25°C} = 104.83 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

An energy balance on the control volume gives

$$h_1 + (\omega_2 - \omega_1)h_w = h_2$$
$$h_1 + (0.02079 - \omega_1)(104.83 \text{ kJ/kg}) = 78.11 \text{ kJ/kg}$$

Pressure and temperature are known for inlet air. Other properties may be determined from this equation using EES. A hand solution would require a trial-error approach. The results are

$$h_1 = 77.66 \text{ kJ/kg dry air}$$
$$\omega_1 = \mathbf{0.01654 \text{ kg } H_2O\text{/kg dry air}}$$
$$\phi_1 = \mathbf{0.4511}$$

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Psychrometric Chart

14-36C They are very nearly parallel to each other.

14-37C The saturation states (located on the saturation curve).

14-38C By drawing a horizontal line until it intersects with the saturation curve. The corresponding temperature is the dew-point temperature.

14-39C No, they cannot. The enthalpy of moist air depends on ω, which depends on the total pressure.

14-40  The pressure, temperature, and relative humidity of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the wet-bulb temperature, the dew-point temperature, and the specific volume of the air are to be determined.

Analysis From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0087 \text{ kg H}_2\text{O / kg dry air}$

(*b*) $h = 45.4 \text{ kJ/kg dry air}$

(*c*) $T_{\text{wb}} = 16.3°\text{C}$

(*d*) $T_{\text{dp}} = 12.0°\text{C}$

(*e*) $\nu = 0.851 \text{ m}^3 \text{ / kg dry air}$

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14-41 Problem 14-40 is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 2000 m.

Analysis The problem is solved using EES, and the solution is given below.

```
Tdb=23 [C]
Rh=0.50
P1=101.325 [kPa]
Z = 2000 [m]
P2=101.325*(1-0.02256*Z*convert(m,km))^5.256  "Relation giving P as a function of altitude"

    h1=enthalpy(AirH2O,T=Tdb,P=P1,R=Rh)
    v1=volume(AirH2O,T=Tdb,P=P1,R=Rh)
    Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,R=Rh)
    w1=humrat(AirH2O,T=Tdb,P=P1,R=Rh)
    Twb1=wetbulb(AirH2O,T=Tdb,P=P1,R=Rh)

    h2=enthalpy(AirH2O,T=Tdb,P=P2,R=Rh)
    v2=volume(AirH2O,T=Tdb,P=P2,R=Rh)
    Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,R=Rh)
    w2=humrat(AirH2O,T=Tdb,P=P2,R=Rh)
    Twb2=wetbulb(AirH2O,T=Tdb,P=P2,R=Rh)
```

SOLUTION

```
h1=45.4 [kJ/kg]
h2=51.62 [kJ/kg]
P1=101.3 [kPa]
P2=79.49 [kPa]
Rh=0.5
Tdb=23 [C]
Tdp1=12.03 [C]
Tdp2=12.03 [C]
Twb1=16.25 [C]
Twb2=15.67 [C]
v1=0.8507 [m^3/kg]
v2=1.089 [m^3/kg]
w1=0.008747 [kg/kg]
w2=0.01119 [kg/kg]
Z=2000 [m]
```

Discussion The atmospheric pressure for a given elevation can also be obtained from Table A-16 of the book.

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14-42 The pressure and the dry- and wet-bulb temperatures of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the relative humidity, the dew-point temperature, and the specific volume of the air are to be determined.

Analysis From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0092$ kg H_2O / kg dry air

(*b*) $h = 47.6$ kJ / kg dry air

(*c*) $\phi = 49.6\%$

(*d*) $T_{dp} = 12.8°C$

(*e*) $\nu = 0.855\ m^3$ / kg dry air



14-43 Problem 14-42 is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 3000 m.

Analysis The problem is solved using EES, and the solution is given below.

```
Tdb=24 [C]
Twb=17 [C]
P1=101.325 [kPa]
Z = 3000 [m]
P2=101.325*(1-0.02256*Z*convert(m,km))^5.256  "Relation giving P as function of altitude"

    h1=enthalpy(AirH2O,T=Tdb,P=P1,B=Twb)
    v1=volume(AirH2O,T=Tdb,P=P1,B=Twb)
    Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,B=Twb)
    w1=humrat(AirH2O,T=Tdb,P=P1,B=Twb)
    Rh1=relhum(AirH2O,T=Tdb,P=P1,B=Twb)

    h2=enthalpy(AirH2O,T=Tdb,P=P2,B=Twb)
    v2=volume(AirH2O,T=Tdb,P=P2,B=Twb)
    Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,B=Twb)
    w2=humrat(AirH2O,T=Tdb,P=P2,B=Twb)
    Rh2=relhum(AirH2O,T=Tdb,P=P2,B=Twb)
```

SOLUTION

h1=47.61 [kJ/kg]	h2=61.68 [kJ/kg]
P1=101.3 [kPa]	P2=70.11 [kPa]
Rh1=0.4956	Rh2=0.5438
Tdb=24 [C]	Tdp1=12.81 [C]
Tdp2=14.24 [C]	Twb=17 [C]
v1=0.8542 [m^3/kg]	v2=1.245 [m^3/kg]
w1=0.009219 [kg/kg]	w2=0.01475 [kg/kg]
Z=3000 [m]	

Discussion The atmospheric pressure for a given elevation can also be obtained from Table A-16 of the book.

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14-44E The pressure, temperature, and relative humidity of air are specified. Using the psychrometric chart, the wet-bulb temperature, specific humidity, the enthalpy, the dew-point temperature, and the water vapor pressure are to be determined.

Analysis From the psychrometric chart in Fig. A-31E or using EES psychrometric functions we obtain

(*a*) $T_{wb} = 81.6°C$

(*b*) $\omega = 0.0214 \text{ lbm } H_2O / \text{lbm dry air}$

(*c*) $h = 45.2 \text{ Btu/lbm dry air}$

(*d*) $T_{dp} = 78.9°F$

(*e*) $P_v = \phi P_g = \phi P_{sat@90°F} = (0.70)(0.69904 \text{ psia}) = 0.489 \text{ psia}$

Air
1 atm
90°F
70% RH

14-45 The pressure, temperature, and wet-bulb temperature of air are specified. Using the psychrometric chart, the relative humidity, specific humidity, the enthalpy, the dew-point temperature, and the water vapor pressure are to be determined.

Analysis From the psychrometric chart in Fig. A-31 or using EES psychrometric functions we obtain

(*a*) $\phi = 0.242 = 24.2\%$

(*b*) $\omega = 0.0072 \text{ kg } H_2O / \text{kg dry air}$

(*c*) $h = 50.6 \text{ kJ/kg dry air}$

(*d*) $T_{dp} = 9.1°C$

(*e*) $P_v = \phi P_g = \phi P_{sat@32°C} = (0.242)(4.760 \text{ kPa}) = 1.15 \text{ kPa}$

Air
1 atm
32°C
T_{wb}=18°C

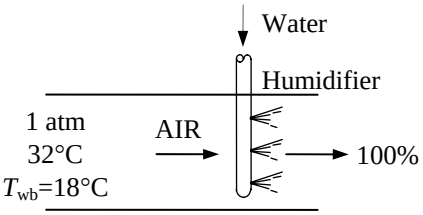
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14-46 The pressure, temperature, and wet-bulb temperature of air are specified. The adiabatic saturation temperature is to be determined.

Analysis For an adiabatic saturation process, we obtained Eq. 14-14 in the text,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

This requires a trial-error solution for the adiabatic saturation temperature, T_2. The inlet state properties are



$$\omega_1 = 0.0072 \text{ kg H}_2\text{O / kg dry air}$$

$$h_{g1} = h_{g@32°C} = 2559.2 \text{ kJ/kg}$$

As a first estimate, let us take T_2 =18°C (the inlet wet-bulb temperature). Also, at the exit, the relative humidity is 100% ($\phi_2 = 1$) and the pressure is 1 atm. Other properties at the exit state are

$$\omega_2 = 0.0130 \text{ kg H}_2\text{O / kg dry air}$$

$$h_{f2} = h_{f@18°C} = 75.54 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

$$h_{fg2} = h_{fg@18°C} = 2458.3 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

Substituting,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}} = \frac{(1.005)(18-32) + (0.0130)(2458.3)}{2559.2 - 75.54} = 0.00720 \text{ kg H}_2\text{O / kg dry air}$$

which is equal to the inlet specific humidity. Therefore, the adiabatic saturation temperature is

T_2 = **18°C**

Discussion This result is not surprising since the wet-bulb and adiabatic saturation temperatures are approximately equal to each other for air-water mixtures at atmospheric pressure.

14-47E The pressure, temperature, and wet-bulb temperature of air are specified. Using the psychrometric chart, the relative humidity, specific humidity, the enthalpy, the dew-point temperature, and the water vapor pressure are to be determined.

Analysis From the psychrometric chart in Fig. A-31E or using EES psychrometric functions we obtain

Air
1 atm
90°F
T_{dp}=75°F

(*a*) $\phi = 0.615 = 61.5\%$

(*b*) $\omega = 0.0188 \text{ lbm H}_2\text{O / lbm dry air}$

(*c*) $h = 42.3 \text{ Btu/lbm dry air}$

(*d*) $T_{wb} = 78.9°\text{F}$

(*e*) $P_v = \phi P_g = \phi P_{sat@28°C} = (0.615)(0.69904 \text{ psia}) = 0.430 \text{ psia}$

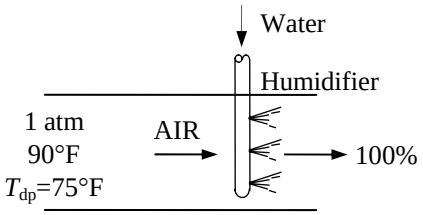
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14-48E The pressure, temperature, and wet-bulb temperature of air are specified. The adiabatic saturation temperature is to be determined.

Analysis For an adiabatic saturation process, we obtained Eq. 14-14 in the text,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

This requires a trial-error solution for the adiabatic saturation temperature, T_2. The inlet state properties are



$$\omega_1 = 0.0188 \text{ lbm } H_2O / \text{lbm dry air} \quad \text{(Fig. A-31E)}$$

$$h_{g1} = h_{g@90°F} = 1100.4 \text{ Btu/lbm} \quad \text{(Table A-4E)}$$

As a first estimate, let us take T_2 =78.9°F ≈ 79°F (the inlet wet-bulb temperature). Also, at the exit, the relative humidity is 100% ($\phi_2 = 1$) and the pressure is 1 atm. Other properties at the exit state are

$$\omega_2 = 0.0215 \text{ lbm } H_2O / \text{lbm dry air}$$

$$h_{f2} = h_{f@79°F} = 47.07 \text{ Btu/lbm} \quad \text{(Table A - 4E)}$$

$$h_{fg2} = h_{fg@79°F} = 1048.6 \text{ Btu/lbm} \quad \text{(Table A - 4E)}$$

Substituting,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}} = \frac{(0.240)(79 - 90) + (0.0215)(1048.6)}{1100.4 - 47.07} = 0.0189 \text{ lbm } H_2O / \text{lbm dry air}$$

which is sufficiently close to the inlet specific humidity (0.0188). Therefore, the adiabatic saturation temperature is

$$T_2 \cong \mathbf{79°F}$$

Discussion This result is not surprising since the wet-bulb and adiabatic saturation temperatures are approximately equal to each other for air-water mixtures at atmospheric pressure.

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Human Comfort and Air-Conditioning

14-49C It humidifies, dehumidifies, cleans and even deodorizes the air.

14-50C (*a*) Perspires more, (*b*) cuts the blood circulation near the skin, and (*c*) sweats excessively.

14-51C It is the direct heat exchange between the body and the surrounding surfaces. It can make a person feel chilly in winter, and hot in summer.

14-52C It affects by removing the warm, moist air that builds up around the body and replacing it with fresh air.

14-53C The spectators. Because they have a lower level of activity, and thus a lower level of heat generation within their bodies.

14-54C Because they have a large skin area to volume ratio. That is, they have a smaller volume to generate heat but a larger area to lose it from.

14-55C It affects a body's ability to perspire, and thus the amount of heat a body can dissipate through evaporation.

14-56C Humidification is to add moisture into an environment, dehumidification is to remove it.

14-57C The metabolism refers to the burning of foods such as carbohydrates, fat, and protein in order to perform the necessary bodily functions. The metabolic rate for an average man ranges from 108 W while reading, writing, typing, or listening to a lecture in a classroom in a seated position to 1250 W at age 20 (730 at age 70) during strenuous exercise. The corresponding rates for women are about 30 percent lower. Maximum metabolic rates of trained athletes can exceed 2000 W. We are interested in metabolic rate of the occupants of a building when we deal with heating and air conditioning because the metabolic rate represents the rate at which a body generates heat and dissipates it to the room. This body heat contributes to the heating in winter, but it adds to the cooling load of the building in summer.

14-58C The metabolic rate is proportional to the size of the body, and the metabolic rate of women, in general, is lower than that of men because of their smaller size. Clothing serves as insulation, and the thicker the clothing, the lower the environmental temperature that feels comfortable.

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14-59C Sensible heat is the energy associated with a temperature change. The sensible heat loss from a human body increases as (*a*) the skin temperature increases, (*b*) the environment temperature decreases, and (*c*) the air motion (and thus the convection heat transfer coefficient) increases.

14-60C Latent heat is the energy released as water vapor condenses on cold surfaces, or the energy absorbed from a warm surface as liquid water evaporates. The latent heat loss from a human body increases as (*a*) the skin wetness increases and (*b*) the relative humidity of the environment decreases. The rate of evaporation from the body is related to the rate of latent heat loss by $\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg}$ where h_{fg} is the latent heat of vaporization of water at the skin temperature.

14-61 A department store expects to have a specified number of people at peak times in summer. The contribution of people to the sensible, latent, and total cooling load of the store is to be determined.

Assumptions There is a mix of men, women, and children in the classroom.

Properties The average rate of heat generation from people doing light work is 115 W, and 70% of is in sensible form (see Sec. 14-6).

Analysis The contribution of people to the sensible, latent, and total cooling load of the store are

$$\dot{Q}_{\text{people, total}} = (\text{No. of people}) \times \dot{Q}_{\text{person, total}} = 245 \times (115 \text{ W}) = \mathbf{28{,}180 \text{ W}}$$

$$\dot{Q}_{\text{people, sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person, sensible}} = 245 \times (0.7 \times 115 \text{ W}) = \mathbf{19{,}720 \text{ W}}$$

$$\dot{Q}_{\text{people, latent}} = (\text{No. of people}) \times \dot{Q}_{\text{person, latent}} = 245 \times (0.3 \times 115 \text{ W}) = \mathbf{8450 \text{ W}}$$

14-62E There are a specified number of people in a movie theater in winter. It is to be determined if the theater needs to be heated or cooled.

Assumptions There is a mix of men, women, and children in the classroom.

Properties The average rate of heat generation from people in a movie theater is 105 W, and 70 W of it is in sensible form and 35 W in latent form.

Analysis Noting that only the sensible heat from a person contributes to the heating load of a building, the contribution of people to the heating of the building is

$$\dot{Q}_{\text{people, sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person, sensible}} = 500 \times (70 \text{ W}) = 35{,}000 \text{ W} = \mathbf{119{,}420 \text{ Btu/h}}$$

since 1 W = 3.412 Btu/h. The building needs to be heated since the heat gain from people is less than the rate of heat loss of 130,000 Btu/h from the building.

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14-63 The infiltration rate of a building is estimated to be 0.9 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at room temperature of 24°C. **4** The effect of water vapor on air density is negligible.

Properties The gas constant and the specific heat of air are $R = 0.287$ kPa.m^3/kg.K and $c_p = 1.005$ kJ/kg·°C (Table A-2). The heat of vaporization of water at 24°C is $h_{fg} = h_{fg@24°C} = 2444.1$ kJ/kg (Table A-4). The properties of the ambient and room air are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{aligned} T_{\text{ambient}} &= 38^{\text{o}}\text{C} \\ \phi_{\text{ambient}} &= 35\% \end{aligned}\right\} w_{\text{ambient}} = 0.01458 \text{ kg/kg dryair}$$

$$\left.\begin{aligned} T_{\text{room}} &= 24^{\text{o}}\text{C} \\ \phi_{\text{room}} &= 55\% \end{aligned}\right\} w_{\text{room}} = 0.01024 \text{ kg/kg dryair}$$

Analysis Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 0.9 times every hour, the air will enter the room at a mass flow rate of

$$\rho_{\text{ambient}} = \frac{P_0}{RT_0} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa.m}^3\text{/kg.K})(38 + 273 \text{ K})} = 1.135 \text{ kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_{\text{ambient}} \boldsymbol{V}_{\text{room}} \text{ACH} = (1.135 \text{ kg/m}^3)(20 \times 13 \times 3 \text{ m}^3)(0.9 \text{ h}^{-1}) = 797.0 \text{ kg/h} = 0.2214 \text{ kg/s}$$

Then the sensible, latent, and total infiltration heat loads of the room are determined to be

$$\dot{Q}_{\text{infiltration, sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (0.2214 \text{ kg/s})(1.005 \text{ kJ/kg.°C})(38 - 24)°\text{C} = \mathbf{3.11 \text{ kW}}$$

$$\dot{Q}_{\text{infiltration, latent}} = \dot{m}_{\text{air}} (w_{\text{ambient}} - w_{\text{room}}) h_{fg} = (0.2214 \text{ kg/s})(0.01458 - 0.01024)(2444.1 \text{ kJ/kg}) = \mathbf{2.35 \text{ kW}}$$

$$\dot{Q}_{\text{infiltration, total}} = \dot{Q}_{\text{infiltration, sensible}} + \dot{Q}_{\text{infiltration, latent}} = 3.11 + 2.35 = \mathbf{5.46 \text{ kW}}$$

Discussion The specific volume of the dry air at the ambient conditions could also be determined from the psychrometric chart at ambient conditions.

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14-64 The infiltration rate of a building is estimated to be 1.8 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at room temperature of 24°C. **4** The effect of water vapor on air density is negligible.

Properties The gas constant and the specific heat of air are $R = 0.287$ kPa.m^3/kg.K and $c_p = 1.005$ kJ/kg·°C (Table A-2). The heat of vaporization of water at 24°C is $h_{fg} = h_{fg@24°C} = 2444.1$ kJ/kg (Table A-4). The properties of the ambient and room air are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{aligned} T_{\text{ambient}} &= 38^{\circ}\text{C} \\ \phi_{\text{ambient}} &= 35\% \end{aligned}\right\} w_{\text{ambient}} = 0.01458 \text{ kg/kg dryair}$$

$$\left.\begin{aligned} T_{\text{room}} &= 24^{\circ}\text{C} \\ \phi_{\text{room}} &= 55\% \end{aligned}\right\} w_{\text{room}} = 0.01024 \text{ kg/kg dryair}$$

Analysis Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 1.8 times every hour, the air will enter the room at a mass flow rate of

$$\rho_{\text{ambient}} = \frac{P_0}{RT_0} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa.m}^3\text{/kg.K})(38 + 273 \text{ K})} = 1.135 \text{ kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_{\text{ambient}} \boldsymbol{V}_{\text{room}} \text{ACH} = (1.135 \text{ kg/m}^3)(20 \times 13 \times 3 \text{ m}^3)(1.8 \text{ h}^{-1}) = 1594 \text{ kg/h} = 0.4427 \text{ kg/s}$$

Then the sensible, latent, and total infiltration heat loads of the room are determined to be

$$\dot{Q}_{\text{infiltration, sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (0.4427 \text{ kg/s})(1.005 \text{ kJ/kg.°C})(38 - 24)°\text{C} = \mathbf{6.23\ kW}$$

$$\dot{Q}_{\text{infiltration, latent}} = \dot{m}_{\text{air}} (w_{\text{ambient}} - w_{\text{room}}) h_{fg} = (0.4427 \text{ kg/s})(0.01458 - 0.01024)(2444.1 \text{ kJ/kg}) = \mathbf{4.69\ kW}$$

$$\dot{Q}_{\text{infiltration, total}} = \dot{Q}_{\text{infiltration, sensible}} + \dot{Q}_{\text{infiltration, latent}} = 6.23 + 4.69 = \mathbf{10.92\ kW}$$

Discussion The specific volume of the dry air at the ambient conditions could also be determined from the psychrometric chart at ambient conditions.

14-65 An average person produces 0.25 kg of moisture while taking a shower. The contribution of showers of a family of four to the latent heat load of the air-conditioner per day is to be determined.

Assumptions All the water vapor from the shower is condensed by the air-conditioning system.

Properties The latent heat of vaporization of water is given to be 2450 kJ/kg.

Analysis The amount of moisture produced per day is

$$\begin{aligned} \dot{m}_{\text{vapor}} &= (\text{Moisture produced per person})(\text{No. of persons}) \\ &= (0.25 \text{ kg / person})(4 \text{ persons / day}) = 1 \text{ kg / day} \end{aligned}$$

Then the latent heat load due to showers becomes

$$\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg} = (1 \text{ kg / day})(2450 \text{ kJ / kg}) = \mathbf{2450\ kJ\ /\ day}$$

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14-66 There are 100 chickens in a breeding room. The rate of total heat generation and the rate of moisture production in the room are to be determined.

Assumptions All the moisture from the chickens is condensed by the air-conditioning system.

Properties The latent heat of vaporization of water is given to be 2430 kJ/kg. The average metabolic rate of chicken during normal activity is 10.2 W (3.78 W sensible and 6.42 W latent).

Analysis The total rate of heat generation of the chickens in the breeding room is

$$\dot{Q}_{\text{gen, total}} = \dot{q}_{\text{gen, total}}(\text{No. of chickens}) = (10.2\ \text{W / chicken})(100\ \text{chickens}) = \mathbf{1020\ W}$$

The latent heat generated by the chicken and the rate of moisture production are

$$\begin{aligned}\dot{Q}_{\text{gen, latent}} &= \dot{q}_{\text{gen, latent}}(\text{No. of chickens}) \\ &= (6.42\ \text{W/chicken})(100\ \text{chickens}) = 642\ \text{W} \\ &= 0.642\ \text{kW}\end{aligned}$$

$$\dot{m}_{\text{moisture}} = \frac{\dot{Q}_{\text{gen, latent}}}{h_{fg}} = \frac{0.642\ \text{kJ / s}}{2430\ \text{kJ / kg}} = 0.000264\ \text{kg / s} = \mathbf{0.264\ g / s}$$

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Simple Heating and Cooling

14-67C Relative humidity decreases during a simple heating process and increases during a simple cooling process. Specific humidity, on the other hand, remains constant in both cases.

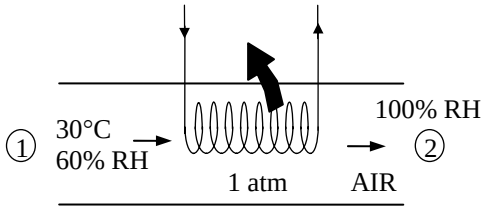
14-68C Because a horizontal line on the psychrometric chart represents a ω = constant process, and the moisture content ω of air remains constant during these processes.

14-69 Humid air at a specified state is cooled at constant pressure to the dew-point temperature. The cooling required for this process is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the cooling section since the process involves no humidification or dehumidification. The inlet and exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 71.3 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0161 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$T_{\text{dp},1} = 21.4°\text{C}$$



The exit state enthalpy is

$$\left.\begin{aligned} P &= 1 \text{ atm} \\ T_2 &= T_{\text{dp},1} = 21.4°\text{C} \\ \phi_2 &= 1 \end{aligned}\right\} \; h_2 = 62.4 \text{ kJ/kg dry air}$$

From the energy balance on air in the cooling section,

$$q_{\text{out}} = h_1 - h_2 = 71.3 - 62.4 = \mathbf{8.9 \text{ kJ/kg dry air}}$$

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14-70E Humid air at a specified state is heated at constant pressure to a specified temperature. The relative humidity at the exit and the amount of heat required are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

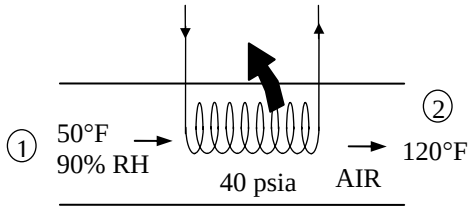
Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 40 psia. The properties of the air at the inlet and exit states are determined to be

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 50°F}} = (0.90)(0.17812\ \text{psia}) = 0.16031\ \text{psia}$$

$$h_{g1} = h_{g\,@\,50°\text{F}} = 1083.1\ \text{Btu/lbm}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.16031\ \text{psia})}{(40 - 0.16031)\ \text{psia}}$$

$$= 0.002503\ \text{lbm H}_2\text{O/lbm dry air}$$



$$h_1 = c_p T_1 + \omega_1 h_{g1}$$

$$= (0.240\ \text{Btu/lbm}\cdot°\text{F})(50°\text{F}) + (0.002503)(1083.1\ \text{Btu/lbm}) = 14.71\ \text{Btu/lbm dry air}$$

$$P_{v2} = P_{v1} = 0.16031\ \text{psia}$$

$$P_{g2} = P_{\text{sat @ 120°F}} = 1.6951\ \text{psia}$$

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{0.16031\ \text{psia}}{1.6951\ \text{psia}} = 0.0946 = \mathbf{9.46\%}$$

$$h_{g2} = h_{g\,@\,120°\text{F}} = 1113.2\ \text{Btu/lbm}$$

$$\omega_2 = \omega_1$$

$$h_2 = c_p T_2 + \omega_2 h_{g2}$$

$$= (0.240\ \text{Btu/lbm}\cdot°\text{F})(120°\text{F}) + (0.002503)(1113.2\ \text{Btu/lbm}) = 31.59\ \text{Btu/lbm dry air}$$

From the energy balance on air in the heating section,

$$q_{\text{in}} = h_2 - h_1 = 31.59 - 14.71 = \mathbf{16.9\ Btu/lbm\ dry\ air}$$

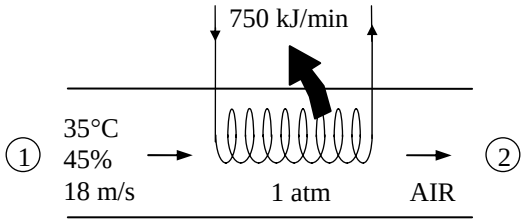
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14-71 Air enters a cooling section at a specified pressure, temperature, velocity, and relative humidity. The exit temperature, the exit relative humidity of the air, and the exit velocity are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the cooling section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31 or EES) to be

$$h_1 = 76.14 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01594 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$\nu_1 = 0.8953 \text{ m}^3\text{/ kg dry air}$$



The mass flow rate of dry air through the cooling section is

$$\dot{m}_a = \frac{1}{\nu_1} V_1 A_1$$
$$= \frac{1}{(0.8953 \text{ m}^3\text{/ kg})} (18 \text{ m/s})(\pi \times 0.3^2/4 \text{ m}^2)$$
$$= 1.421 \text{ kg/s}$$

From the energy balance on air in the cooling section,

$$-\dot{Q}_{\text{out}} = \dot{m}_a (h_2 - h_1)$$
$$-(750/60) \text{ kJ/s} = (1.421 \text{ kg/s})(h_2 - 76.14) \text{ kJ/kg}$$
$$h_2 = 67.35 \text{ kJ/kg dry air}$$

(*b*) The exit state of the air is fixed now since we know both h_2 and ω_2. From the psychrometric chart at this state we read

$$T_2 = \mathbf{26.5°C}$$
$$\phi_2 = \mathbf{73.1\%}$$
$$\nu_2 = 0.8706 \text{ m}^3\text{/ kg dry air}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{\nu_1} = \frac{\dot{V}_2}{\nu_2} \longrightarrow \frac{V_1 A}{\nu_1} = \frac{V_2 A}{\nu_2}$$
$$V_2 = \frac{\nu_2}{\nu_1} V_1 = \frac{0.8706}{0.8953} (18 \text{ m/s}) = \mathbf{17.5 \text{ m/s}}$$

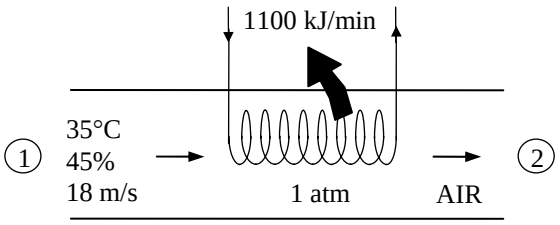
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14-72 Air enters a cooling section at a specified pressure, temperature, velocity, and relative humidity. The exit temperature, the exit relative humidity of the air, and the exit velocity are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the cooling section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31 or EES) to be

$$h_1 = 76.14 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01594 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$\nu_1 = 0.8953 \text{ m}^3 \text{/ kg dry air}$$



The mass flow rate of dry air through the cooling section is

$$\dot{m}_a = \frac{1}{\nu_1} V_1 A_1$$
$$= \frac{1}{(0.8953 \text{ m}^3 \text{/ kg})} (18 \text{ m/s})(\pi \times 0.3^2 / 4 \text{ m}^2)$$
$$= 1.421 \text{ kg/s}$$

From the energy balance on air in the cooling section,

$$-\dot{Q}_{\text{out}} = \dot{m}_a (h_2 - h_1)$$
$$-(1100/60) \text{ kJ/s} = (1.421 \text{ kg/s})(h_2 - 76.14) \text{ kJ/kg}$$
$$h_2 = 63.24 \text{ kJ/kg dry air}$$

(*b*) The exit state of the air is fixed now since we know both h_2 and ω_2. From the psychrometric chart at this state we read

$$T_2 = \mathbf{22.6°C}$$
$$\phi_2 = \mathbf{92.6\%}$$
$$\nu_2 = 0.8591 \text{ m}^3 \text{/ kg dry air}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{\nu_1} = \frac{\dot{V}_2}{\nu_2} \longrightarrow \frac{V_1 A}{\nu_1} = \frac{V_2 A}{\nu_2}$$
$$V_2 = \frac{\nu_2}{\nu_1} V_1 = \frac{0.8591}{0.8953} (18 \text{ m/s}) = \mathbf{17.3 \text{ m/s}}$$

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14-73 Saturated humid air at a specified state is heated to a specified temperature. The relative humidity at the exit and the rate of heat transfer are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

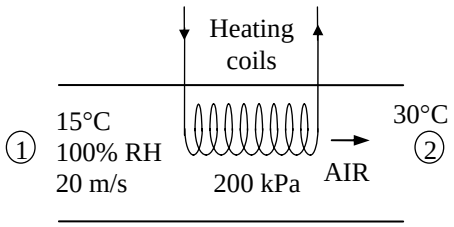
Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 200 kPa. The properties of the air at the inlet and exit states are determined to be

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 15°C}} = (1.0)(1.7057\ \text{kPa}) = 1.7057\ \text{kPa}$$

$$h_{g1} = h_{g\,@15°\text{C}} = 2528.3\ \text{kJ/kg}$$

$$P_{a1} = P_1 - P_{v1} = 200 - 1.7057 = 198.29\ \text{kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{198.29\ \text{kPa}} = 0.4168\ \text{m}^3/\text{kg dry air}$$



$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.7057\ \text{kPa})}{(200 - 1.7057)\ \text{kPa}} = 0.005350\ \text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\ \text{kJ/kg}\cdot°\text{C})(15°\text{C}) + (0.005350)(2528.3\ \text{kJ/kg}) = 28.60\ \text{kJ/kg dry air}$$

$$P_{v2} = P_{v1} = 1.7057\ \text{kPa}$$

$$P_{g2} = P_{\text{sat @ 30°C}} = 4.2469\ \text{kPa}$$

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{1.7057\ \text{kPa}}{4.2469\ \text{kPa}} = 0.402 = \mathbf{40.2\%}$$

$$h_{g2} = h_{g\,@30°\text{C}} = 2555.6\ \text{kJ/kg}$$

$$\omega_2 = \omega_1$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\ \text{kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.005350)(2555.6\ \text{kJ/kg}) = 43.82\ \text{kJ/kg dry air}$$

Then,

$$\dot{V}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (20\ \text{m/s})\left(\frac{\pi(0.04\ \text{m})^2}{4}\right) = 0.02513\ \text{m}^3/\text{s}$$

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{0.02513\ \text{m}^3/\text{s}}{0.4168\ \text{m}^3/\text{kg dry air}} = 0.06029\ \text{kg/s}$$

From the energy balance on air in the heating section,

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (0.06029\ \text{kg/s})(43.82 - 28.60)\text{kJ/kg} = \mathbf{0.918\ kW}$$

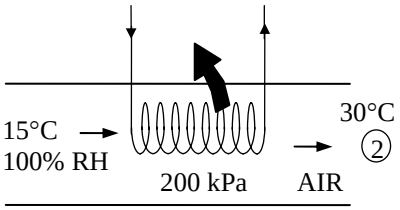
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14-74 Saturated humid air at a specified state is heated to a specified temperature. The rate at which the exergy of the humid air is increased is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 200 kPa. The properties of the air at the inlet and exit states are determined to be

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @15°C}} = (1.0)(1.7057\ \text{kPa}) = 1.7057\ \text{kPa}$$
$$h_{g1} = h_{g\,@15°\text{C}} = 2528.3\ \text{kJ/kg}$$
$$s_{g1} = s_{g\,@15°\text{C}} = 8.7803\ \text{kJ/kg}\cdot\text{K}$$
$$P_{a1} = P_1 - P_{v1} = 200 - 1.7057 = 198.29\ \text{kPa}$$
$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{198.29\ \text{kPa}} = 0.4168\ \text{m}^3/\text{kg dry air}$$



$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.7057\ \text{kPa})}{(200 - 1.7057)\ \text{kPa}} = 0.005350\ \text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\ \text{kJ/kg}\cdot°\text{C})(15°\text{C}) + (0.005350)(2528.3\ \text{kJ/kg}) = 28.60\ \text{kJ/kg dry air}$$

$$P_{v2} = P_{v1} = 1.7057\ \text{kPa}$$
$$P_{g2} = P_{\text{sat @30°C}} = 4.2469\ \text{kPa}$$
$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{1.7057\ \text{kPa}}{4.2469\ \text{kPa}} = 0.402 = \mathbf{40.2\%}$$
$$P_{a2} = P_2 - P_{v2} = 200 - 1.7057 = 198.29\ \text{kPa}$$
$$h_{g2} = h_{g\,@30°\text{C}} = 2555.6\ \text{kJ/kg}$$
$$s_{g2} = s_{g\,@30°\text{C}} = 8.4520\ \text{kJ/kg}\cdot\text{K}$$
$$\omega_2 = \omega_1$$
$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\ \text{kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.005350)(2555.6\ \text{kJ/kg}) = 43.82\ \text{kJ/kg dry air}$$

The entropy change of the dry air is

$$(s_2 - s_1)_{\text{dry air}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_{a2}}{P_{a1}} = (1.005)\ln\frac{303}{288} - (0.287)\ln\frac{198.29}{198.29} = 0.05103\ \text{kJ/kg}\cdot\text{K}$$

The entropy change of the air-water mixture is

$$s_2 - s_1 = (s_2 - s_1)_{\text{dry air}} + \omega(s_2 - s_1)_{\text{water vapor}} = 0.05103 + (0.005350)(8.4520 - 8.7803) = 0.04927\ \text{kJ/kg}\cdot\text{K}$$

The mass flow rate of the dry air is

$$\dot{V}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (20\ \text{m/s})\left(\frac{\pi(0.04\ \text{m})^2}{4}\right) = 0.02513\ \text{m}^3/\text{s}$$
$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{0.02513\ \text{m}^3/\text{s}}{0.4168\ \text{m}^3/\text{kg dry air}} = 0.06029\ \text{kg/s}$$

The exergy increase of the humid air during this process is then,

$$\Delta\Phi = \dot{m}_a(\psi_2 - \psi_1) = \dot{m}_a\left[(h_2 - h_1) - T_0(s_2 - s_1)\right]$$
$$= (0.06029\ \text{kg/s})\left[(43.82 - 28.60)\text{kJ/kg} - (288\ \text{K})(0.04927\ \text{kJ/kg}\cdot\text{K})\right] = \mathbf{0.062\ kW}$$

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Heating with Humidification

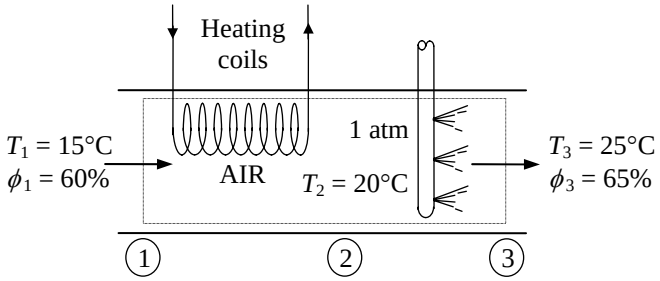
14-75C To achieve a higher level of comfort. Very dry air can cause dry skin, respiratory difficulties, and increased static electricity.

14-76 Air is first heated and then humidified by water vapor. The amount of steam added to the air and the amount of heat transfer to the air are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$h_1 = 31.1$ kJ / kg dry air
$\omega_1 = 0.0064$ kg H_2O / kg dry air ($= \omega_2$)
$h_2 = 36.2$ kJ / kg dry air
$h_3 = 58.1$ kJ / kg dry air
$\omega_3 = 0.0129$ kg H_2O / kg dry air



Analysis (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The amount of steam added to the air in the heating section is

$$\Delta\omega = \omega_3 - \omega_2 = 0.0129 - 0.0064 = \mathbf{0.0065\ kg\ H_2O\ /\ kg\ dry\ air}$$

(*b*) The heat transfer to the air in the heating section per unit mass of air is

$$q_{in} = h_2 - h_1 = 36.2 - 31.1 = \mathbf{5.1\ kJ\ /\ kg\ dry\ air}$$

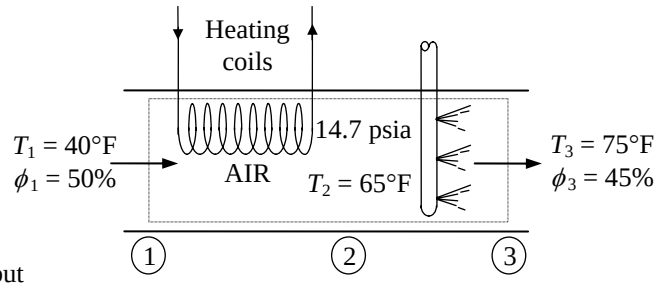
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14-77E Air is first heated and then humidified by water vapor. The amount of steam added to the air and the amount of heat transfer to the air are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31E or EES) to be

$h_1 = 12.40$ Btu/lbm dry air
$\omega_1 = 0.002586$ lbm H_2O/lbm dry air
$h_2 = 18.44$ Btu/lbm dry air
$\omega_2 = \omega_1 = 0.002586$ lbm H_2O/lbm dry air
$h_3 = 27.10$ Btu/lbm dry air
$\omega_3 = 0.008298$ lbm H_2O/lbm dry air



Analysis (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The amount of steam added to the air in the heating section is

$$\Delta\omega = \omega_3 - \omega_2 = 0.008298 - 0.002586 = \mathbf{0.0057\ lbm\ H_2O/lbm\ dry\ air}$$

(*b*) The heat transfer to the air in the heating section per unit mass of air is

$$q_{in} = h_2 - h_1 = 18.44 - 12.40 = \mathbf{6.0\ Btu/lbm\ dry\ air}$$

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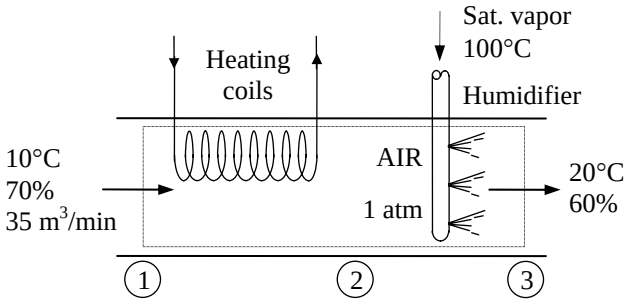
14-78 Air is first heated and then humidified by wet steam. The temperature and relative humidity of air at the exit of heating section, the rate of heat transfer, and the rate at which water is added to the air are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$h_1 = 23.5$ kJ/kg dry air
$\omega_1 = 0.0053$ kg H_2O/kg dry air ($= \omega_2$)
$\upsilon_1 = 0.809$ m^3/kg dry air

$h_3 = 42.3$ kJ/kg dry air
$\omega_3 = 0.0087$ kg H_2O/kg dry air



Analysis (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{\upsilon_1} = \frac{35\ \text{m}^3/\text{min}}{0.809\ \text{m}^3/\text{kg}} = 43.3\ \text{kg/min}$$

Noting that $Q = W = 0$, the energy balance on the humidifying section can be expressed as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0 \text{ (steady)}}{} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad \longrightarrow \quad \dot{m}_w h_w + \dot{m}_{a2} h_2 = \dot{m}_a h_3$$
$$(\omega_3 - \omega_2) h_w + h_2 = h_3$$

Solving for h_2,

$$h_2 = h_3 - (\omega_3 - \omega_2) h_{g\,@100°\text{C}} = 42.3 - (0.0087 - 0.0053)(2675.6) = 33.2\ \text{kJ/kg dry air}$$

Thus at the exit of the heating section we have $\omega_2 = 0.0053$ kg H_2O dry air and $h_2 = 33.2$ kJ/kg dry air, which completely fixes the state. Then from the psychrometric chart we read

$T_2 = \mathbf{19.5°C}$
$\phi_2 = \mathbf{37.8\%}$

(*b*) The rate of heat transfer to the air in the heating section is

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (43.3\ \text{kg/min})(33.2 - 23.5)\ \text{kJ/kg} = \mathbf{420\ kJ/min}$$

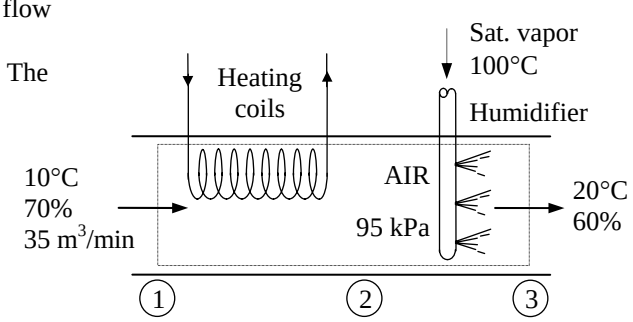
(*c*) The amount of water added to the air in the humidifying section is determined from the conservation of mass equation of water in the humidifying section,

$$\dot{m}_w = \dot{m}_a (\omega_3 - \omega_2) = (43.3\ \text{kg/min})(0.0087 - 0.0053) = \mathbf{0.15\ kg/min}$$

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14-79 Air is first heated and then humidified by wet steam. The temperature and relative humidity of air at the exit of heating section, the rate of heat transfer, and the rate at which water is added to the air are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.



Analysis (*a*) The amount of moisture in the air also remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. The properties of the air at various states are determined to be

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 10°C}} = (0.70)(1.2281\,\text{kPa}) = 0.860\,\text{kPa}\ (= P_{v2})$$

$$P_{a1} = P_1 - P_{v1} = 95 - 0.860 = 94.14\,\text{kPa}$$

$$\nu_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\,\text{K})}{94.14\,\text{kPa}} = 0.863\,\text{m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.86\,\text{kPa})}{(95-0.86)\,\text{kPa}} = 0.00568\,\text{kg H}_2\text{O/kg dry air}\ (= \omega_2)$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\,\text{kJ/kg}\cdot\text{°C})(10\text{°C}) + (0.00568)(2519.2\,\text{kJ/kg}) = 24.36\,\text{kJ/kg dry air}$$

$$P_{v3} = \phi_3 P_{g3} = \phi_3 P_{\text{sat @ 20°C}} = (0.60)(2.3392\,\text{kPa}) = 1.40\,\text{kPa}$$

$$\omega_3 = \frac{0.622\,P_{v3}}{P_3 - P_{v3}} = \frac{0.622(1.40\,\text{kPa})}{(95-1.40)\,\text{kPa}} = 0.00930\,\text{kg H}_2\text{O/kg dry air}$$

$$h_3 = c_p T_3 + \omega_3 h_{g3} = (1.005\,\text{kJ/kg}\cdot\text{°C})(20\text{°C}) + (0.0093)(2537.4\,\text{kJ/kg}) = 43.70\,\text{kJ/kg dry air}$$

Also, $$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{35\,\text{m}^3/\text{min}}{0.863\,\text{m}^3/\text{kg}} = 40.6\,\text{kg/min}$$

Noting that $Q = W = 0$, the energy balance on the humidifying section gives

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \overset{\nearrow 0\ \text{(steady)}}{} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}_w h_w + \dot{m}_{a2} h_2 = \dot{m}_a h_3 \longrightarrow (\omega_3 - \omega_2) h_w + h_2 = h_3$$

$$h_2 = h_3 - (\omega_3 - \omega_2) h_{g\,@\,100\text{°C}} = 43.7 - (0.0093 - 0.00568) \times 2675.6 = 34.0\,\text{kJ/kg dry air}$$

Thus at the exit of the heating section we have $\omega = 0.00568$ kg H_2O dry air and $h_2 = 34.0$ kJ/kg dry air, which completely fixes the state. The temperature of air at the exit of the heating section is determined from the definition of enthalpy,

$$h_2 = c_p T_2 + \omega_2 h_{g2} \cong c_p T_2 + \omega_2 (2500.9 + 1.82 T_2)$$

$$34.0 = (1.005) T_2 + (0.00568)(2500.9 + 1.82 T_2)$$

Solving for h_2, yields

$$T_2 = \mathbf{19.5°C}$$

The relative humidity at this state is

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{P_{v2}}{P_{\text{sat @ 19.5°C}}} = \frac{0.860\,\text{kPa}}{2.2759\,\text{kPa}} = 0.377 \quad \text{or} \quad \mathbf{37.7\%}$$

(*b*) The rate of heat transfer to the air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (40.6\,\text{kg/min})(34.0 - 24.36)\,\text{kJ/kg} = \mathbf{391\,kJ/min}$$

(*c*) The amount of water added to the air in the humidifying section is determined from the conservation of mass equation of water in the humidifying section,

$$\dot{m}_w = \dot{m}_a (\omega_3 - \omega_2) = (40.6\ \text{kg/min})(0.0093 - 0.00568) = \mathbf{0.147\ kg/min}$$

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Cooling with Dehumidification

14-80C To drop its relative humidity to more desirable levels.

14-81E Air is cooled and dehumidified at constant pressure. The amount of water removed from the air and the cooling requirement are to be determined.

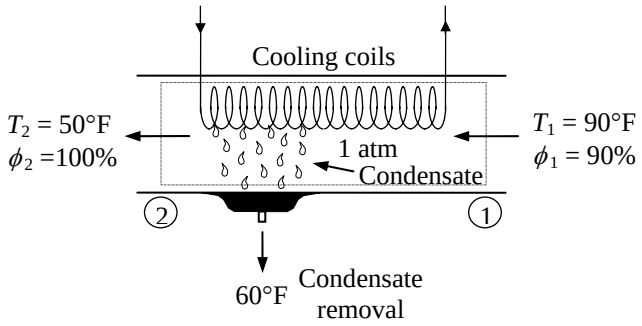
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31E) to be

$$h_1 = 52.2 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0278 \text{ kg H}_2\text{O/kg dry air}$$

and

$$\phi_2 = 1.0$$
$$h_2 = 20.3 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0076 \text{ kg H}_2\text{O/kg dry air}$$



Also,

$$h_w \cong h_{f\,@\,60°\text{F}} = 28.08 \text{ Btu/lbm} \quad \text{(Table A-4)}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\Delta\omega = \omega_1 - \omega_2 = 0.0278 - 0.0076 = \mathbf{0.0202 \text{ lbm H}_2\text{O/lbm dry air}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$

$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$q_{\text{out}} = h_1 - h_2 - (\omega_1 - \omega_2)h_w$$

$$= (52.2 - 20.3)\text{Btu/lbm} - (0.0202)(28.08)$$

$$= \mathbf{31.3 \text{ Btu/lbm dry air}}$$

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14-82 Air is cooled and dehumidified at constant pressure. The amount of water removed from the air and the rate of cooling are to be determined.

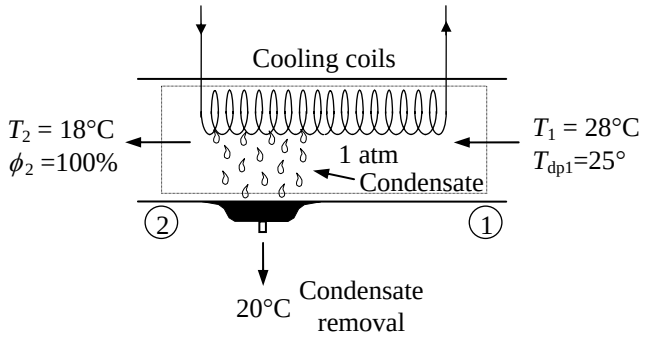
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$h_1 = 79.6$ kJ/kg dry air
$\omega_1 = 0.0202$ kg H_2O/kg dry air
$\nu_1 = 0.881\,m^3$/kg dry air

and

$\phi_2 = 1.0$
$h_2 = 51.0$ kJ/kg dry air
$\omega_2 = 0.0130$ kg H_2O/kg dry air



Also,

$h_w \cong h_{f\,@\,20°C} = 83.915$ kJ/kg (Table A-4)

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass flow rate of air is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{\nu_1} = \frac{(10{,}000/3600)\,m^3/s}{0.881\,m^3/kg\ dry\ air} = 3.153\,kg/s$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (3.153\,kg/s)(0.0202 - 0.0130) = \mathbf{0.0227\ kg/s}$$

Energy Balance:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\ (steady)} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$

$$\dot{Q}_{out} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{out} = (3.153\,kg/s)(79.6 - 51.0)kJ/kg - (0.02227\,kg/s)(83.915\,kJ/kg)$$

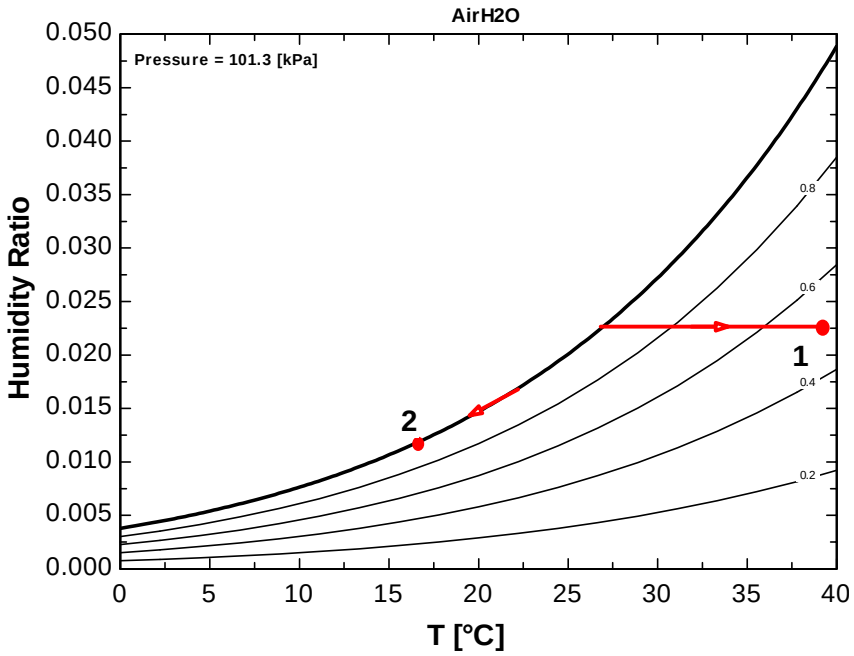
$$= \mathbf{88.3\ kW}$$

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14-83 Air is cooled and dehumidified at constant pressure. The syatem hardware and the psychrometric diagram are to be sketched and the inlet volume flow rate is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties (a) The schematic of the cooling and dehumidifaction process and the the process on the psychrometric chart are given below. The psychrometric chart is obtained from the Property Plot feature of EES.

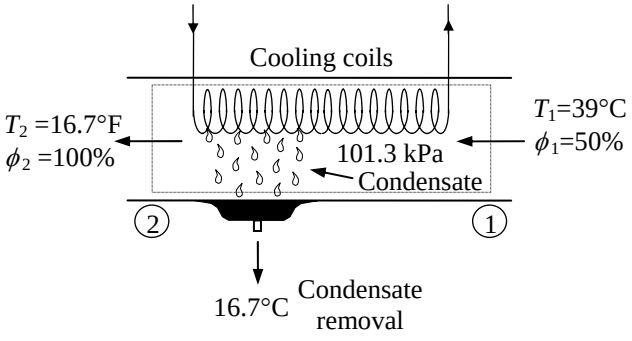


(b) The inlet and the exit states of the air are completely specified, and the total pressure is 101.3 kPa (1 atm). The properties of the air at the inlet and exit states are determined from the psychrometric chart (Figure A-31 or EES) to be

$$T_{dp} = 26.7°C$$
$$h_1 = 96.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0222 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.916 \text{ m}^3\text{/kg dry air}$$

and

$$T_2 = T_2 - 10 = 26.7 - 10 = 16.7°C$$
$$\phi_2 = 1.0$$
$$h_2 = 46.9 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0119 \text{ kg H}_2\text{O/kg dry air}$$



Also,

$$h_w \cong h_{f\,@16.7°C} = 70.10 \text{ kJ/kg} \quad \text{(Table A-4)}$$

Analysis Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$$

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Energy Balance:

$$\begin{aligned}
\dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \\
\dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\
\sum \dot{m}_i h_i &= \dot{Q}_{out} + \sum \dot{m}_e h_e \\
\dot{Q}_{\text{out}} &= \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w) \\
&= \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w \\
&= \dot{m}_a (h_1 - h_2) - \dot{m}_a (\omega_1 - \omega_2) h_w
\end{aligned}$$

Solving for the mass flow rate of dry air,

$$\dot{m}_a = \frac{\dot{Q}_{\text{out}}}{(h_1 - h_2) - (\omega_1 - \omega_2) h_w} = \frac{1340 \text{ kW}}{(96.5 - 46.9) \text{ kJ/kg} - (0.0222 - 0.0119)(70.1 \text{ kJ/kg})} = 27.4 \text{ kg/s}$$

The volume flow rate is then determined from

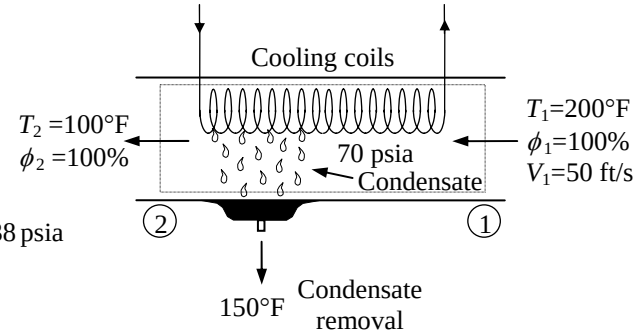
$$\dot{V}_1 = \dot{m}_a v_1 = (27.4 \text{ kg/s})(0.916 \text{ m}^3/\text{kg}) = \mathbf{25.1 m^3/s}$$

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14-84E Saturated humid air at a specified state is cooled at constant pressure to a specified temperature. The rate at which liquid water is formed and the rate of cooling are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 70 psia. The properties of the air at the inlet and exit states are determined to be



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 200°F}} = (1.0)(11.538 \text{ psia}) = 11.538 \text{ psia}$$

$$h_{g1} = h_{g\,@\,200°\text{F}} = 1145.7 \text{ Btu/lbm}$$

$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(11.538 \text{ psia})}{(70 - 11.538) \text{ psia}} = 0.1228 \text{ lbm H}_2\text{O/lbm dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (0.240 \text{ Btu/lbm} \cdot °\text{F})(200°\text{F}) + (0.1228)(1145.7 \text{ Btu/lbm}) = 188.7 \text{ Btu/lbm dry air}$$

$$\nu_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(660 \text{ K})}{(70 - 11.538) \text{ psia}} = 4.182 \text{ ft}^3/\text{lbm dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat @ 100°F}} = (1.0)(0.95052 \text{ psia}) = 0.95052 \text{ psia}$$

$$h_{g2} = h_{g\,@\,100°\text{F}} = 1104.7 \text{ Btu/lbm}$$

$$\omega_2 = \frac{0.622\, P_{v2}}{P_2 - P_{v2}} = \frac{0.622(0.95052 \text{ psia})}{(70 - 0.95052) \text{ psia}} = 0.00856 \text{ lbm H}_2\text{O/lbm dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (0.240 \text{ Btu/lbm} \cdot °\text{F})(100°\text{F}) + (0.00856)(1104.7 \text{ Btu/lbm}) = 33.46 \text{ Btu/lbm dry air}$$

We assume that the condensate leaves this system at the average temperature of the air inlet and exit:

$$h_w \cong h_{f\,@\,150°\text{F}} = 117.99 \text{ Btu/lbm} \quad \text{(Table A-4E)}$$

The mass flow rate of the dry air is

$$\dot{V}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (50 \text{ ft/s})\left(\frac{\pi (3/12 \text{ ft})^2}{4}\right) = 2.454 \text{ ft}^3/\text{s}$$

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{2.454 \text{ ft}^3/\text{s}}{4.182 \text{ ft}^3/\text{lbm dry air}} = 0.5868 \text{ lbm/s}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (0.5868 \text{ lbm/s})(0.1228 - 0.00856) = \mathbf{0.0670 \text{ lbm/s}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$

$$\dot{Q}_{\text{out}} = \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w)$$

$$= \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w$$

$$= (0.5868 \text{ lbm})(188.7 - 33.46) \text{Btu/lbm} - (0.06704 \text{ lbm})(117.99 \text{ Btu/lbm})$$

$$= \mathbf{83.2 \text{ Btu/s}}$$

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14-85 Air is first cooled, then dehumidified, and finally heated. The temperature of air before it enters the heating section, the amount of heat removed in the cooling section, and the amount of heat supplied in the heating section are to be determined.

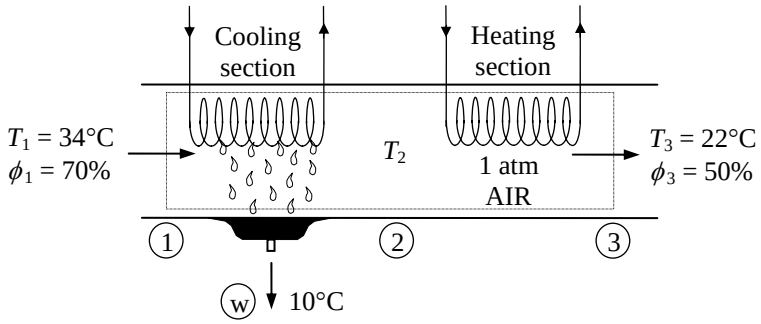
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (a) The amount of moisture in the air decreases due to dehumidification ($\omega_3 < \omega_1$), and remains constant during heating ($\omega_3 = \omega_2$). The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The intermediate state (state 2) is also known since $\phi_2 = 100\%$ and $\omega_2 = \omega_3$. Therefore, we can determine the properties of the air at all three states from the psychrometric chart (Fig. A-31) to be

$$h_1 = 95.2 \text{ kJ / kg dry air}$$
$$\omega_1 = 0.0238 \text{ kg } H_2O \text{ / kg dry air}$$

and

$$h_3 = 43.1 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0082 \text{ kg } H_2O \text{ / kg dry air } (= \omega_2)$$



Also,

$$h_w \cong h_{f\,@10°C} = 42.02 \text{ kJ/kg} \quad \text{(Table A - 4)}$$
$$h_2 = 31.8 \text{ kJ/kg dry air}$$
$$T_2 = \mathbf{11.1°C}$$

(b) The amount of heat removed in the cooling section is determined from the energy balance equation applied to the cooling section,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out,cooling}}$$
$$\dot{Q}_{\text{out,cooling}} = \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w) = \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w$$

or, per unit mass of dry air,

$$q_{\text{out,cooling}} = (h_1 - h_2) - (\omega_1 - \omega_2) h_w$$
$$= (95.2 - 31.8) - (0.0238 - 0.0082)42.02$$
$$= \mathbf{62.7 \text{ kJ/kg dry air}}$$

(c) The amount of heat supplied in the heating section per unit mass of dry air is

$$q_{\text{in,heating}} = h_3 - h_2 = 43.1 - 31.8 = \mathbf{11.3 \text{ kJ / kg dry air}}$$

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14-86 Air is cooled by passing it over a cooling coil through which chilled water flows. The rate of heat transfer, the mass flow rate of water, and the exit velocity of airstream are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The saturation pressure of water at 32ºC is 4.76 kPa (Table A-4). Then the dew point temperature of the incoming air stream at 32°C becomes

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,0.7\times 4.76\,\text{kPa}} = 25.8°\text{C} \quad \text{(Table A-5)}$$

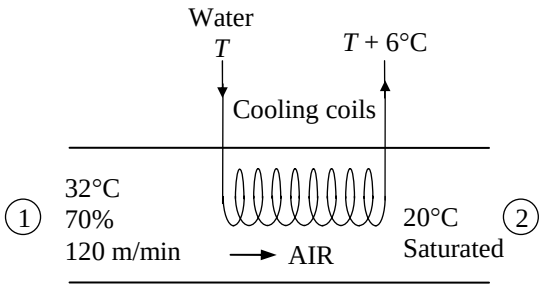
since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense. The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Then the properties of the air at both states are determined from the psychrometric chart (Fig. A-31 or EES) to be

$$h_1 = 86.35\,\text{kJ/kg dry air}$$
$$\omega_1 = 0.02114\,\text{kg H}_2\text{O/kg dry air}$$
$$\upsilon_1 = 0.8939\,\text{m}^3\text{/kg dry air}$$

and

$$h_2 = 57.43\,\text{kJ/kg dry air}$$
$$\omega_2 = 0.0147\,\text{kg H}_2\text{O/kg dry air}$$
$$\upsilon_2 = 0.8501\,\text{m}^3\text{/kg dry air}$$

Also, $h_w \cong h_{f\,@\,20°\text{C}} = 83.91\,\text{kJ/kg}$ (Table A-4)



Then,

$$\dot{\boldsymbol{V}}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (120\,\text{m/min})\left(\frac{\pi(0.4\,\text{m})^2}{4}\right) = 15.08\,\text{m}^3/\text{min}$$

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\upsilon_1} = \frac{15.08\,\text{m}^3/\text{min}}{0.8939\,\text{m}^3/\text{kg dry air}} = 16.87\,\text{kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the water),

Water Mass Balance: $\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (16.87\,\text{kg/min})(0.02114 - 0.0147) = 0.1086\ \text{kg/min}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out}} \longrightarrow Q_{out} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (16.87\,\text{kg/min})(86.35 - 57.43)\text{kJ/kg} - (0.1086\,\text{kg/min})(83.91\,\text{kJ/kg}) = \mathbf{478.7\ kJ/min}$$

(*b*) Noting that the heat lost by the air is gained by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{cooling water}} = \dot{m}_{\text{cooling water}}\Delta h = \dot{m}_{\text{cooling water}} c_p \Delta T$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}_w}{c_p \Delta T} = \frac{478.7\,\text{kJ/min}}{(4.18\,\text{kJ/kg}\cdot°\text{C})(6°\text{C})} = \mathbf{19.09\ kg/min}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{\boldsymbol{V}}_1}{\upsilon_1} = \frac{\dot{\boldsymbol{V}}_2}{\upsilon_2} \longrightarrow \frac{V_1 A}{\upsilon_1} = \frac{V_2 A}{\upsilon_2}$$

$$V_2 = \frac{\upsilon_2}{\upsilon_1} V_1 = \frac{0.8501}{0.8939}(120\ \text{m/min}) = \mathbf{114\ m/min}$$

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14-87 Problem 14-86 is reconsidered. A general solution of the problem in which the input variables may be supplied and parametric studies performed is to be developed and the process is to be shown in the psychrometric chart for each set of input variables.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data from the Diagram Window"
{D=0.4
P[1] =101.32 [kPa]
T[1] = 32 [C]
RH[1] = 70/100   "%, relative humidity"
Vel[1] = 120/60 "[m/s]"
DELTAT_cw =6 [C]
P[2] = 101.32 [kPa]
T[2] = 20 [C]}
RH[2] = 100/100 "%"

"Dry air flow rate, m_dot_a, is constant"
Vol_dot[1]= (pi * D^2)/4*Vel[1]
v[1]=VOLUME(AirH2O,T=T[1],P=P[1],R=RH[1])
m_dot_a = Vol_dot[1]/v[1]

"Exit vleocity"
Vol_dot[2]= (pi * D^2)/4*Vel[2]
v[2]=VOLUME(AirH2O,T=T[2],P=P[2],R=RH[2])
m_dot_a = Vol_dot[2]/v[2]

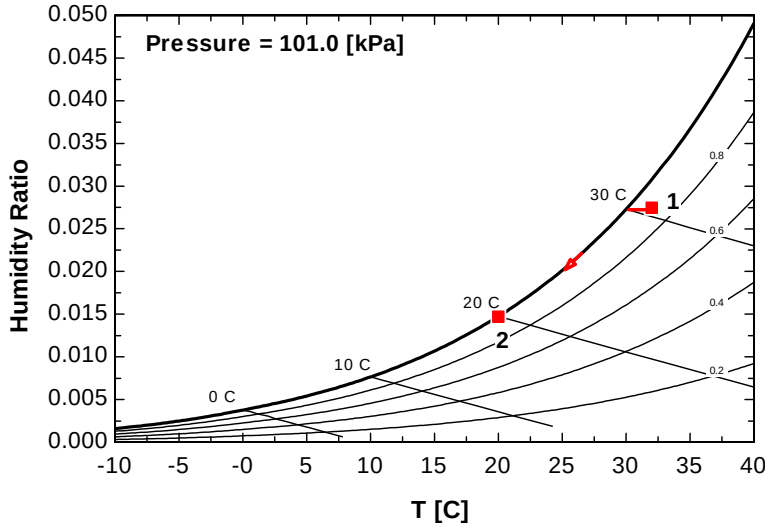
"Mass flow rate of the condensed water"
m_dot_v[1]=m_dot_v[2]+m_dot_w
w[1]=HUMRAT(AirH2O,T=T[1],P=P[1],R=RH[1])
m_dot_v[1] = m_dot_a*w[1]
w[2]=HUMRAT(AirH2O,T=T[2],P=P[2],R=RH[2])
m_dot_v[2] = m_dot_a*w[2]

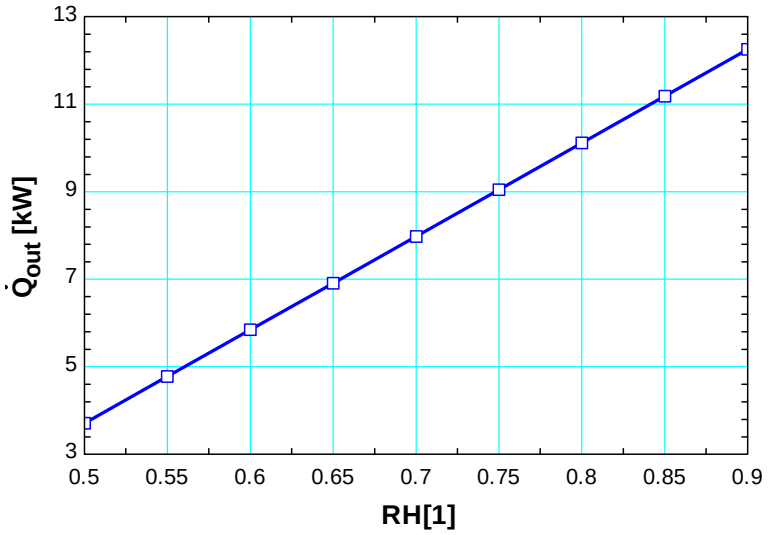
"SSSF conservation of energy for the air"
m_dot_a *(h[1] + (1+w[1])*Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)) + Q_dot = m_dot_a*(h[2]
+(1+w[2])*Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)) +m_dot_w*h_liq_2
h[1]=ENTHALPY(AirH2O,T=T[1],P=P[1],w=w[1])
h[2]=ENTHALPY(AirH2O,T=T[2],P=P[2],w=w[2])
h_liq_2=ENTHALPY(Water,T=T[2],P=P[2])

"SSSF conservation of energy for the cooling water"
-Q_dot =m_dot_cw*Cp_cw*DELTAT_cw  "Note: Q_netwater=-Q_netair"
Cp_cw = SpecHeat(water,T=10,P=P[2])"kJ/kg-K"
```

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RH_1	m_{cw} [kg/s]	Q_{out} [kW]	Vel_2 [m/s]
0.5	0.1475	3.706	1.921
0.55	0.19	4.774	1.916
0.6	0.2325	5.842	1.911
0.65	0.275	6.91	1.907
0.7	0.3175	7.978	1.902
0.75	0.36	9.046	1.897
0.8	0.4025	10.11	1.893
0.85	0.445	11.18	1.888
0.9	0.4875	12.25	1.884





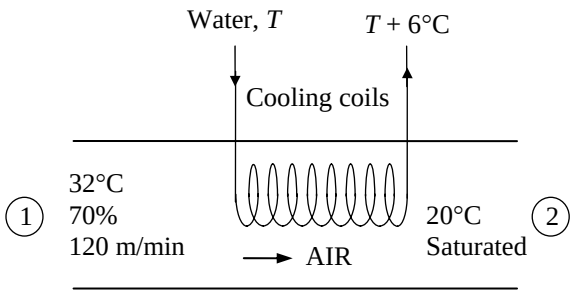
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14-88 Air is cooled by passing it over a cooling coil. The rate of heat transfer, the mass flow rate of water, and the exit velocity of airstream are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The dew point temperature of the incoming air stream at 35°C is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 32°C}}$$
$$= (0.7)(4.76\ \text{kPa}) = 3.332\ \text{kPa}$$
$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 3.332 kPa}} = 25.8°\text{C}$$



Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.

The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. Then the properties of the air at both states are determined to be

$$P_{a1} = P_1 - P_{v1} = 95 - 3.332 = 91.67\ \text{kPa}$$

$$\nu_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\ \text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(305\ \text{K})}{91.67\ \text{kPa}} = 0.9549\ \text{m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(3.332\ \text{kPa})}{(95 - 3.332)\ \text{kPa}} = 0.02261\ \text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\ \text{kJ/kg} \cdot °\text{C})(32°\text{C}) + (0.02261)(2559.2\ \text{kJ/kg})$$
$$= 90.01\ \text{kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = (1.00) P_{\text{sat @ 20°C}} = 2.339\ \text{kPa}$$

$$\nu_2 = \frac{R_a T_2}{P_{a2}} = \frac{(0.287\ \text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293\ \text{K})}{(95 - 2.339)\ \text{kPa}} = 0.9075\ \text{m}^3/\text{kg dry air}$$

$$\omega_2 = \frac{0.622\, P_{v2}}{P_2 - P_{v2}} = \frac{0.622(2.339\ \text{kPa})}{(95 - 2.339)\ \text{kPa}} = 0.0157\ \text{kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\ \text{kJ/kg} \cdot °\text{C})(20°\text{C}) + (0.0157)(2537.4\ \text{kJ/kg})$$
$$= 59.94\ \text{kJ/kg dry air}$$

Also,

$$h_w \cong h_{f\,@\,20°\text{C}} = 83.915\ \text{kJ/kg} \qquad \text{(Table A-4)}$$

Then,

$$\dot{V}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (120\ \text{m/min})\left(\frac{\pi(0.4\ \text{m})^2}{4}\right) = 15.08\ \text{m}^3/\text{min}$$

$$\dot{m}_{a1} = \frac{\dot{V}_1}{\nu_1} = \frac{15.08\ \text{m}^3/\text{min}}{0.9549\ \text{m}^3/\text{kg dry air}} = 15.79\ \text{kg/min}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section (excluding the water),

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Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \quad \longrightarrow \quad \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (15.79\ \text{kg/min})(0.02261 - 0.0157) = 0.1090\ \text{kg/min}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out}} \quad \rightarrow \quad \dot{Q}_{out} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (15.79\ \text{kg/min})(90.01 - 59.94)\text{kJ/kg} - (0.1090\ \text{kg/min})(83.915\ \text{kJ/kg}) = \mathbf{465.7\ kJ/min}$$

(*b*) Noting that the heat lost by the air is gained by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{cooling water}} = \dot{m}_{\text{cooling water}} \Delta h = \dot{m}_{\text{cooling water}} c_p \Delta T$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}_w}{c_p \Delta T} = \frac{465.7\ \text{kJ/min}}{(4.18\ \text{kJ/kg} \cdot °\text{C})(6°\text{C})} = \mathbf{18.57\ kg/min}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{\nu_1} = \frac{\dot{V}_2}{\nu_2} \longrightarrow \frac{V_1 A}{\nu_1} = \frac{V_2 A}{\nu_2}$$

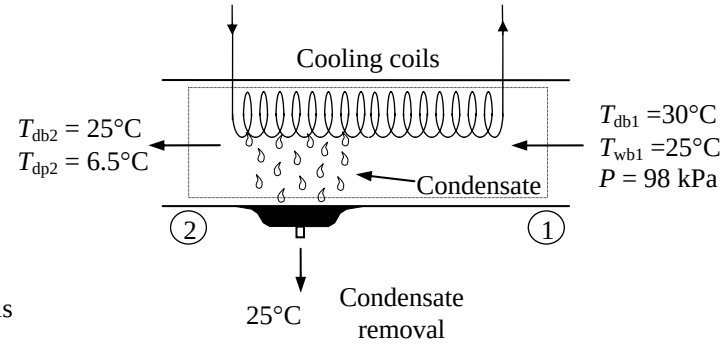
$$V_2 = \frac{\nu_2}{\nu_1} V_1 = \frac{0.9075}{0.9549}(120\ \text{m/min}) = \mathbf{114\ m/min}$$

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14-89 Air flows through an air conditioner unit. The inlet and exit states are specified. The rate of heat transfer and the mass flow rate of condensate water are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The inlet state of the air is completely specified, and the total pressure is 98 kPa. The properties of the air at the inlet state may be determined from (Fig. A-31) or using EES psychrometric functions to be (we used EES)



$$h_1 = 77.88 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01866 \text{ kg H}_2\text{O/kg dry air}$$
$$\phi_1 = 0.6721$$

The partial pressure of water vapor at the exit state is

$$P_{v2} = P_{\text{sat@ 6.5°C}} = 0.9682 \text{ kPa} \quad \text{(Table A - 4)}$$

The saturation pressure at the exit state is

$$P_{g2} = P_{\text{sat@ 25°C}} = 3.17 \text{ kPa} \quad \text{(Table A - 4)}$$

Then, the relative humidity at the exit state becomes

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{0.9682}{3.17} = 0.3054$$

Now, the exit state is also fixed. The properties are obtained from EES to be

$$h_2 = 40.97 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.006206 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_2 = 0.8820 \text{ m}^3\text{/kg}$$

The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_2}{\nu_2} = \frac{1000 \text{ m}^3\text{/min}}{0.8820 \text{ m}^3\text{/kg}} = 1133.8 \text{ kg/min}$$

The mass flow rate of condensate water is

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (1133.8 \text{ kg/min})(0.01866 - 0.006206) = 14.12 \text{ kg/min} = \mathbf{847.2 \text{ kg/h}}$$

The enthalpy of condensate water is

$$h_{w2} = h_{f@ 25°C} = 104.83 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(1133.8 \text{ kg/min})(77.88 \text{ kJ/kg}) = \dot{Q}_{\text{out}} + (1133.8 \text{ kg/min})(40.97 \text{ kJ/kg}) + (14.12 \text{ kg/min})(104.83 \text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 40{,}377 \text{ kJ/min} = \mathbf{672.9 \text{ kW}}$$

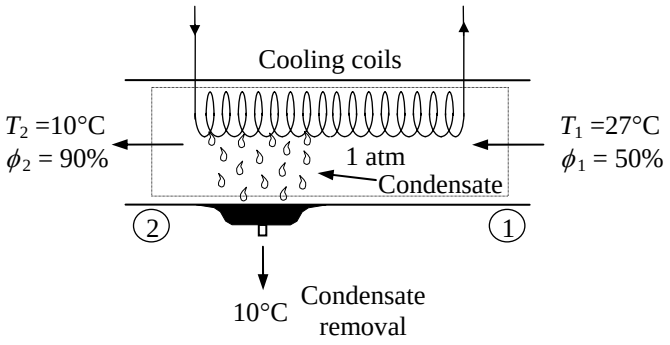
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14-90 Atmospheric air enters the evaporator of an automobile air conditioner at a specified pressure, temperature, and relative humidity. The dew point and wet bulb temperatures at the inlet to the evaporator section, the required heat transfer rate from the atmospheric air to the evaporator fluid, and the rate of condensation of water vapor in the evaporator section are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The inlet and exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at the inlet and exit states may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$T_{dp1} = \mathbf{15.7°C}$
$T_{wb1} = \mathbf{19.5°C}$
$h_1 = 55.60$ kJ/kg dry air
$\omega_1 = 0.01115$ kg H_2O/kg dry air
$v_1 = 0.8655\ m^3$ / kg dry air
$h_2 = 27.35$ kJ/kg dry air
$\omega_2 = 0.00686$ kg H_2O/kg dry air



The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{V_{car}\,\text{ACH}}{v_1} = \frac{(2\ \text{m}^3/\text{change})(5\,\text{changes/min})}{0.8655\ \text{m}^3} = 11.55\ \text{kg/min}$$

The mass flow rates of vapor at the inlet and exit are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01115)(11.55\ \text{kg/min}) = 0.1288\ \text{kg/min}$$

$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.00686)(11.55\ \text{kg/min}) = 0.07926\ \text{kg/min}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{out} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$

where the the enthalpy of condensate water is

$$h_{w2} = h_{f@10°C} = 42.02\ \text{kJ/kg} \quad \text{(Table A - 4)}$$

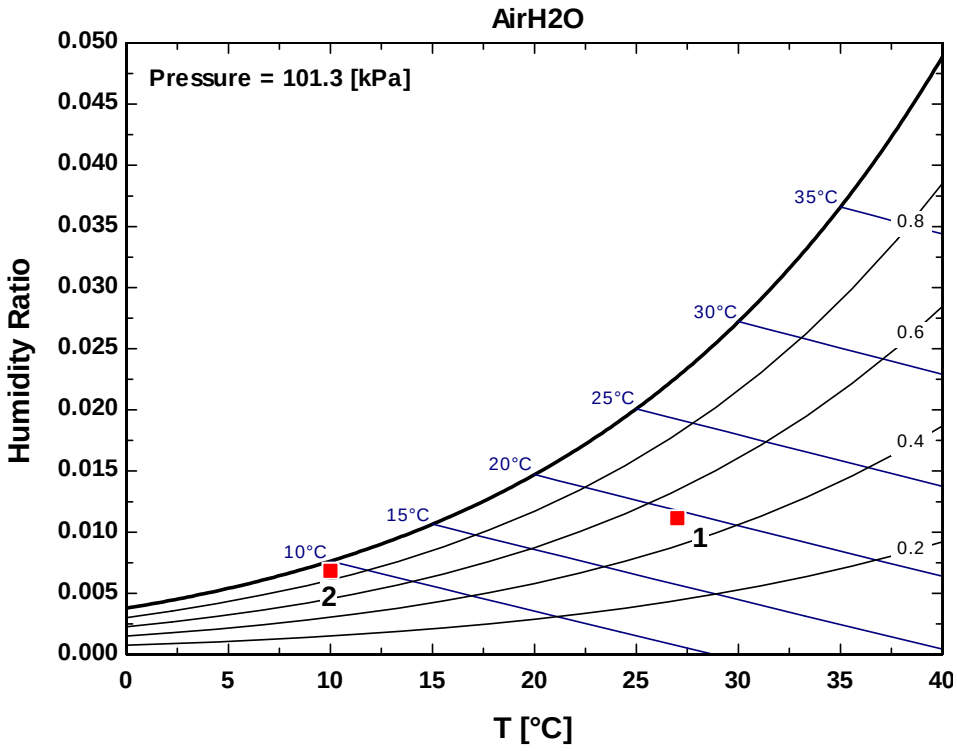
and the rate of condensation of water vapor is

$$\dot{m}_w = \dot{m}_{v1} - \dot{m}_{v2} = 0.1288 - 0.07926 = \mathbf{0.0495\ kg/min}$$

Substituting,

$$\dot{m}_a h_1 = \dot{Q}_{out} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(11.55\ \text{kg/min})(55.60\ \text{kJ/kg}) = \dot{Q}_{out} + (11.55\ \text{kg/min})(27.35\ \text{kJ/kg}) + (0.0495\ \text{kg/min})(42.02\ \text{kJ/kg})$$
$$\dot{Q}_{out} = 324.4\ \text{kJ/min} = \mathbf{5.41\,kW}$$

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Discussion We we could not show the process line between the states 1 and 2 because we do not know the process path.

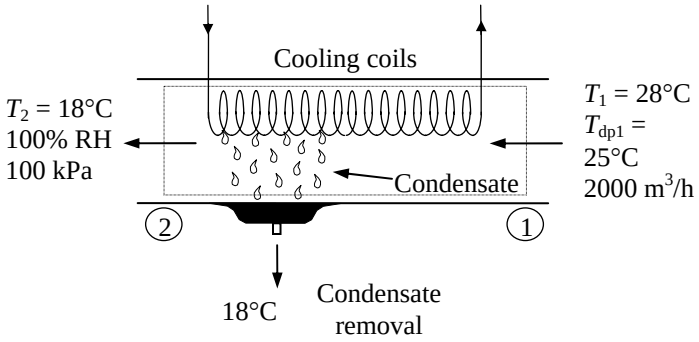
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14-91 Atmospheric air flows into an air conditioner that uses chilled water as the cooling fluid. The mass flow rate of the condensate water and the volume flow rate of chilled water supplied to the air conditioner are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis We may assume that the exit relative humidity is 100 percent since the exit temperature (18°C) is below the dew-point temperature of the inlet air (25°C). The properties of the air at the exit state may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$h_2 = 51.34 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.01311 \text{ kg H}_2\text{O/kg dry air}$$



The partial pressure of water vapor at the inlet state is (Table A-4)

$$P_{v1} = P_{\text{sat@ 25°C}} = 3.17 \text{ kPa}$$

The saturation pressure at the inlet state is

$$P_{g1} = P_{\text{sat@ 28°C}} = 3.783 \text{ kPa} \quad \text{(Table A - 4)}$$

Then, the relative humidity at the inlet state becomes

$$\phi_1 = \frac{P_{v1}}{P_{g1}} = \frac{3.17}{3.783} = 0.8379$$

Now, the inlet state is also fixed. The properties are obtained from EES to be

$$h_1 = 80.14 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.02036 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.8927 \text{ m}^3\text{/kg}$$

The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{(2000/60) \text{ m}^3\text{/h}}{0.8927 \text{ m}^3\text{/kg}} = 37.34 \text{ kg/min}$$

The mass flow rate of condensate water is

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$$
$$= (37.34 \text{ kg/min})(0.02036 - 0.01311)$$
$$= 0.2707 \text{ kg/min}$$
$$= \mathbf{16.24 \text{ kg/h}}$$

The enthalpy of condensate water is

$$h_{w2} = h_{f@18°C} = 75.54 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(37.34 \text{ kg/min})(80.14 \text{ kJ/kg}) = \dot{Q}_{\text{out}} + (37.34 \text{ kg/min})(51.34 \text{ kJ/kg}) + (0.2707 \text{ kg/min})(75.54 \text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 1055 \text{ kJ/min} = 17.59 \text{ kW}$$

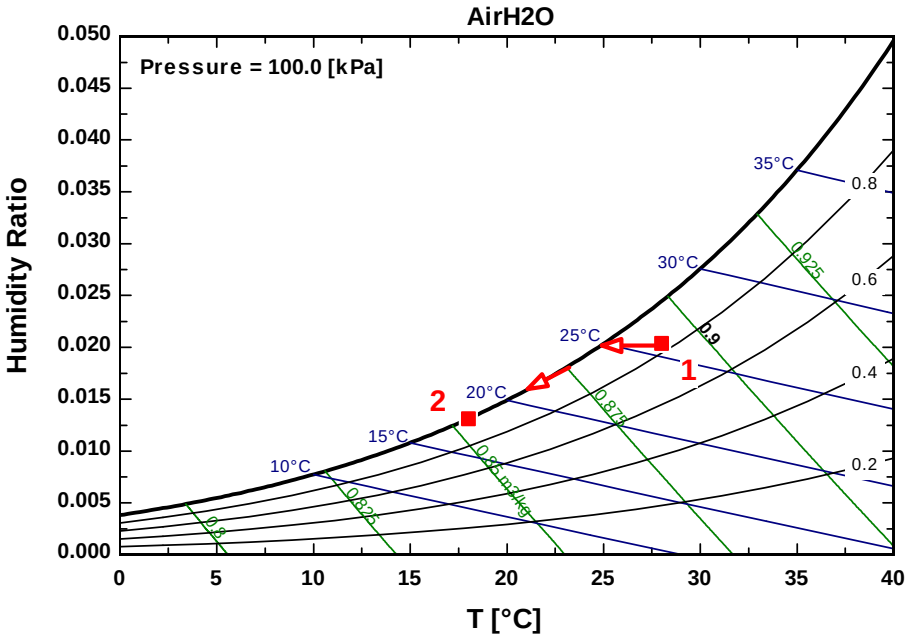
Noting that the rate of heat lost from the air is received by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{in}} = \dot{m}_{cw} c_p \Delta T_{cw} \longrightarrow \dot{m}_{cw} = \frac{\dot{Q}_{\text{in}}}{c_p \Delta T_{cw}} = \frac{1055 \text{ kJ/min}}{(4.18 \text{ kJ/kg.°C})(10°\text{C})} = 25.24 \text{ kg/min}$$

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where we used the specific heat of water value at room temperature. Assuming a density of 1000 kg/m^3 for water, the volume flow rate is determined to be

$$\dot{V}_{cw} = \frac{\dot{m}_{cw}}{\rho_{cw}} = \frac{25.24\ \text{kg/min}}{1000\ \text{kg/m}^3} = \mathbf{0.0252\ m^3/min}$$



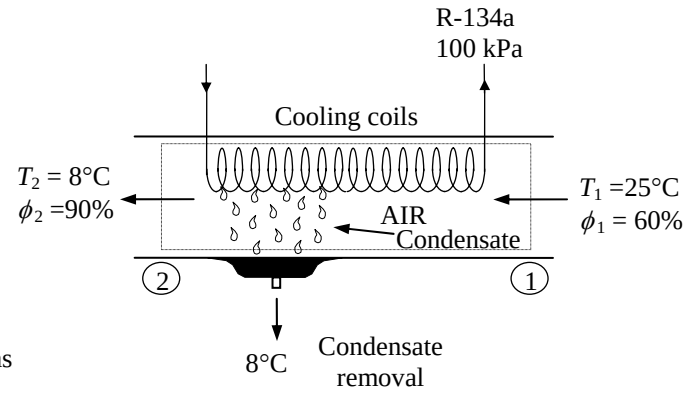
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14-92 An automobile air conditioner using refrigerant 134a as the cooling fluid is considered. The inlet and exit states of moist air in the evaporator are specified. The volume flow rate of the air entering the evaporator of the air conditioner is to be determined.

Assumptions **1** All processes are steady flow and the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis We assume that the total pressure of moist air is 100 kPa. Then, the inlet and exit states of the moist air for the evaporator are completely specified. The properties may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 55.88 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01206 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.8724 \text{ m}^3\text{/kg dry air}$$
$$h_2 = 23.31 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.006064 \text{ kg H}_2\text{O/kg dry air}$$



The mass flow rate of dry air is given by

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{\dot{V}_1}{0.8724 \text{ m}^3\text{/kg}}$$

The mass flow rate of condensate water is expressed as

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = \frac{\dot{V}_1}{0.8724}(0.01206 - 0.006064) = 0.006873\dot{V}_1$$

The enthalpy of condensate water is

$$h_{w2} = h_{f@8°C} = 33.63 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$

$$\frac{\dot{V}_1}{0.8724}(55.88) = \dot{Q}_{\text{out}} + \frac{\dot{V}_1}{0.8724}(23.11) + 0.006873\dot{V}_1(33.63) \qquad (1)$$

The properties of the R-134a at the inlet of the compressor and the enthalpy at the exit for the isentropic process are (R-134a tables)

$$\left.\begin{matrix} P_{R1} = 200 \text{ kPa} \\ x_{R1} = 1 \end{matrix}\right\} \begin{matrix} h_{R1} = 244.46 \text{ kJ/kg} \\ s_{R1} = 0.9377 \text{ kJ/kg.K} \end{matrix}$$

$$\left.\begin{matrix} P_{R2} = 1600 \text{ kPa} \\ s_{R2} = s_{R1} \end{matrix}\right\} h_{R2,s} = 287.87 \text{ kJ/kg}$$

The enthalpies of R-134a at the condenser exit and the throttle exit are

$$h_{R3} = h_{f@1600 \text{ kPa}} = 135.93 \text{ kJ/kg}$$
$$h_{R4} = h_{R3} = 135.93 \text{ kJ/kg}$$

The mass flow rate of the refrigerant can be determined from the expression for the compressor power:

$$\dot{W}_C = \dot{m}_R \frac{h_{R2,s} - h_{R1}}{\eta_C}$$

$$6 \text{ kW} = \dot{m}_R \frac{(287.87 - 244.46) \text{ kJ/kg}}{0.85} \longrightarrow \dot{m}_R = 0.1175 \text{ kg/s} = 7.049 \text{ kg/min}$$

The rate of heat absorbed by the R-134a in the evaporator is

$$\dot{Q}_{R,\text{in}} = \dot{m}_R(h_{R1} - h_{R4}) = (7.049 \text{ kg/min})(244.46 - 135.93) \text{ kJ/kg} = 765.0 \text{ kJ/min}$$

The rate of heat lost from the air in the evaporator is absorbed by the refrigerant-134a. That is, $\dot{Q}_{R,\text{in}} = \dot{Q}_{\text{out}}$. Then, the volume flow rate of the air at the inlet of the evaporator can be determined from Eq. (1) to be

$$\frac{\dot{V}_1}{0.8724}(55.88) = 765.0 + \frac{\dot{V}_1}{0.8724}(23.11) + 0.006873\nu_1(33.63) \longrightarrow \dot{V}_1 = \mathbf{20.62 \text{ m}^3\text{/min}}$$

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14-93 Air is cooled and dehumidified at constant pressure. The cooling required is provided by a simple ideal vapor-compression refrigeration system using refrigerant-134a as the working fluid. The exergy destruction in the total system per 1000 m^3 of dry air is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

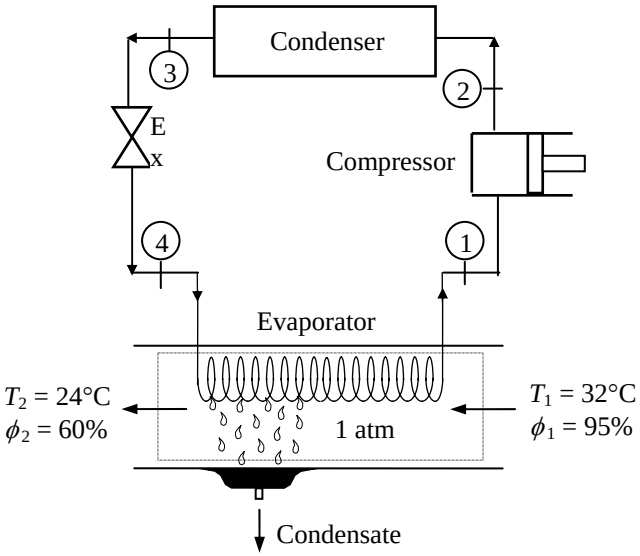
$$h_1 = 106.8 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0292 \text{ kg } H_2O\text{/kg dry air}$$
$$\nu_1 = 0.905 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 52.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0112 \text{ kg } H_2O\text{/kg dry air}$$

We assume that the condensate leaves this system at the average temperature of the air inlet and exit. Then, from Table A-4,

$$h_w \cong h_{f\,@\,28°C} = 117.4 \text{ kJ/kg}$$



Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass of air is

$$m_a = \frac{V_1}{\nu_1} = \frac{1000 \text{ m}^3}{0.905 \text{ m}^3 \text{ / kg dry air}} = 1105 \text{ kg}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$
$$m_w = m_a(\omega_1 - \omega_2) = (1105 \text{ kg})(0.0292 - 0.0112) = 19.89 \text{ kg}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$
$$Q_{\text{out}} = m_a(h_1 - h_2) - m_w h_w$$
$$Q_{\text{out}} = (1105 \text{ kg})(106.8 - 52.7)\text{kJ/kg} - (19.89 \text{ kg})(117.4 \text{ kJ/kg}) = 57{,}450 \text{ kJ}$$

We obtain the properties for the vapor-compression refrigeration cycle as follows (Tables A-11,through A-13):

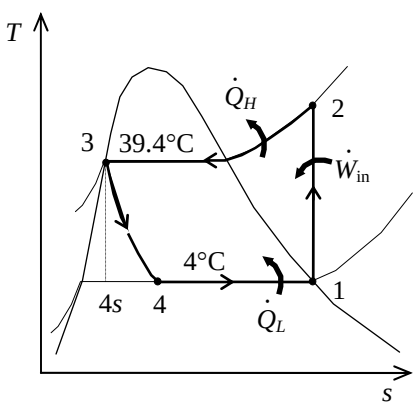
$$\left.\begin{array}{l} T_1 = 4°C \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,4°C} = 252.77 \text{ kJ/kg} \\ s_1 = s_{g\,@\,4°C} = 0.92927 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = P_{\text{sat}\,@\,39.4°C} = 1 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 275.29 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,1\text{MPa}} = 107.32 \text{ kJ/kg} \\ s_3 = s_{f\,@\,1\text{MPa}} = 0.39189 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$h_4 \cong h_3 = 107.32 \text{ kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_4 = 4°C \\ h_4 = 107.32 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_4 = 0.2561 \\ s_4 = 0.4045 \text{ kJ/kg} \cdot \text{K} \end{array}$$



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The mass flow rate of refrigerant-134a is

$$m_R = \frac{Q_L}{h_1 - h_4} = \frac{57{,}450\ \text{kJ}}{(252.77 - 107.32)\text{kJ/kg}} = 395.0\ \text{kg}$$

The amount of heat rejected from the condenser is

$$Q_H = m_R(h_2 - h_3) = (395.0\ \text{kg})(275.29 - 107.32)\ \text{kJ/kg} = 66{,}350\ \text{kg}$$

Next, we calculate the exergy destruction in the components of the refrigeration cycle:

$$X_{\text{destroyed},12} = m_R T_0 (s_2 - s_1) = 0 \quad \text{(since the process is isentropic)}$$

$$X_{\text{destroyed},23} = T_0\left(m_R(s_3 - s_2) + \frac{Q_H}{T_H}\right)$$

$$= (305\ \text{K})\left((395\ \text{kg})(0.39189 - 0.92927)\ \text{kJ/kg}\cdot\text{K} + \frac{66{,}350\ \text{kJ}}{305\ \text{K}}\right) = 1609\ \text{kJ}$$

$$X_{\text{destroyed},34} = m_R T_0 (s_4 - s_3) = (395\ \text{kg})(305\ \text{K})(0.4045 - 0.39189)\ \text{kJ/kg}\cdot\text{K} = 1519\ \text{kJ}$$

The entropies of water vapor in the air stream are

$$s_{g1} = s_{g@32°\text{C}} = 8.4114\ \text{kJ/kg}\cdot\text{K}$$

$$s_{g2} = s_{g@24°\text{C}} = 8.5782\ \text{kJ/kg}\cdot\text{K}$$

The entropy change of water vapor in the air stream is

$$\Delta S_{\text{vapor}} = m_a(\omega_2 s_{g2} - \omega_1 s_{g1}) = (1105)(0.0112 \times 8.5782 - 0.0292 \times 8.4114) = -165.2\ \text{kJ/K}$$

The entropy of water leaving the cooling section is

$$S_w = m_w s_{f@28°\text{C}} = (19.89\ \text{kg})(0.4091\ \text{kJ/kg}\cdot\text{K}) = 8.14\ \text{kJ/K}$$

The partial pressures of water vapor and dry air for air streams are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat}@32°\text{C}} = (0.95)(4.760\ \text{kPa}) = 4.522\ \text{kPa}$$

$$P_{a1} = P_1 - P_{v1} = 101.325 - 4.522 = 96.80\ \text{kPa}$$

$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat}@24°\text{C}} = (0.60)(2.986\ \text{kPa}) = 1.792\ \text{kPa}$$

$$P_{a2} = P_2 - P_{v2} = 101.325 - 1.792 = 99.53\ \text{kPa}$$

The entropy change of dry air is

$$\Delta S_a = m_a(s_2 - s_1) = m_a\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_{a2}}{P_{a1}}\right)$$

$$= (1105)\left[(1.005)\ln\frac{297}{305} - (0.287)\ln\frac{99.53}{96.80}\right] = -38.34\ \text{kJ/kg dry air}$$

The entropy change of R-134a in the evaporator is

$$\Delta S_{R,\,41} = m_R(s_1 - s_4) = (395\ \text{kg})(0.92927 - 0.4045) = 207.3\ \text{kJ/K}$$

An entropy balance on the evaporator gives

$$S_{\text{gen,evaporator}} = \Delta S_{R,41} + \Delta S_{\text{vapor}} + \Delta S_a + S_w = 207.3 + (-165.2) + (-38.34) + 8.14 = 11.90\ \text{kJ/K}$$

Then, the exergy destruction in the evaporator is

$$X_{\text{dest}} = T_0 S_{\text{gen, evaporator}} = (305\ \text{K})(11.90\ \text{kJ/K}) = 3630\ \text{kJ}$$

Finally the total exergy destruction is

$$X_{\text{dest, total}} = X_{\text{dest, compressor}} + X_{\text{dest, condenser}} + X_{\text{dest, throttle}} + X_{\text{dest, evaporator}}$$

$$= 0 + 1609 + 1519 + 3630$$

$$= \mathbf{6758\ kJ}$$

The greatest exergy destruction occurs in the evaporator. Note that heat is absorbed from humid air and rejected to the ambient air at 32°C (305 K), which is also taken as the dead state temperature.

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Evaporative Cooling

14-94C Evaporative cooling is the cooling achieved when water evaporates in dry air. It will not work on humid climates.

14-95C During evaporation from a water body to air, the latent heat of vaporization will be equal to *convection* heat transfer from the air when *conduction* from the lower parts of the water body to the surface is negligible, and temperature of the surrounding surfaces is at about the temperature of the water surface so that the *radiation* heat transfer is negligible.

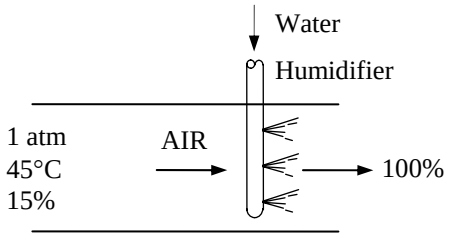
14-96C In steady operation, the mass transfer process does not have to involve heat transfer. However, a mass transfer process that involves phase change (evaporation, sublimation, condensation, melting etc.) must involve heat transfer. For example, the evaporation of water from a lake into air (mass transfer) requires the transfer of latent heat of water at a specified temperature to the liquid water at the surface (heat transfer).

14-97 Desert dwellers often wrap their heads with a water-soaked porous cloth. The temperature of this cloth on a desert with specified temperature and relative humidity is to be determined.

Analysis Since the cloth behaves as the wick on a wet bulb thermometer, the temperature of the cloth will become the wet-bulb temperature. According to the pshchrometric chart, this temperature is

$$T_2 = T_{wb1} = \mathbf{23.3°C}$$

This process can be represented by an evaporative cooling process as shown in the figure.

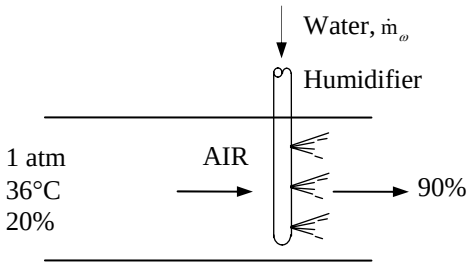


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14-98 Air is cooled by an evaporative cooler. The exit temperature of the air and the required rate of water supply are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31) at 36°C and 20% relative humidity we read

$$T_{wb1} = 19.5°C$$
$$\omega_1 = 0.0074 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.887 \text{ m}^3\text{/kg dry air}$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is,

$$T_{wb2} \cong T_{wb1} = 19.5°C$$

At this wet-bulb temperature and 90% relative humidity we read

$$T_2 = \mathbf{20.5°C}$$
$$\omega_2 = 0.0137 \text{ kg H}_2\text{O / kg dry air}$$

Thus air will be cooled to 20.5°C in this evaporative cooler.

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{4 \text{ m}^3/\text{min}}{0.887 \text{ m}^3/\text{kg dry air}} = 4.51 \text{ kg/min}$$

Then the required rate of water supply to the evaporative cooler is determined from

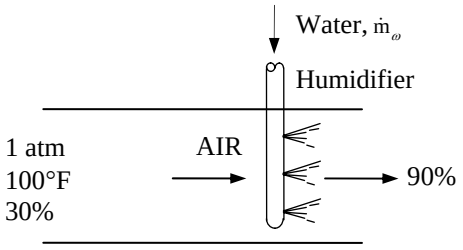
$$\dot{m}_{supply} = \dot{m}_{w2} - \dot{m}_{w1} = \dot{m}_a(\omega_2 - \omega_1)$$
$$= (4.51 \text{ kg/min})(0.0137 - 0.0074)$$
$$= \mathbf{0.028 \text{ kg/min}}$$

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14-99E Air is cooled by an evaporative cooler. The exit temperature of the air and the required rate of water supply are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31E or EES) at 100°F and 30% relative humidity we read

$$T_{wb1} = 74.3°\text{F}$$
$$\omega_1 = 0.0123\ \text{lbm H}_2\text{O/lbm dry air}$$
$$\nu_1 = 14.4\ \text{ft}^3\text{/lbm dry air}$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is,

$$T_{wb2} \cong T_{wb1} = 74.3°\text{F}$$

At this wet-bulb temperature and 90% relative humidity we read

$$T_2 = \mathbf{76.6°F}$$
$$\omega_2 = 0.0178\ \text{lbm H}_2\text{O/lbm dry air}$$

Thus air will be cooled to 76.6°F in this evaporative cooler.

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{200\ \text{ft}^3/\text{min}}{14.4\ \text{ft}^3/\text{lbm dry air}} = 13.9\ \text{lbm/min}$$

Then the required rate of water supply to the evaporative cooler is determined from

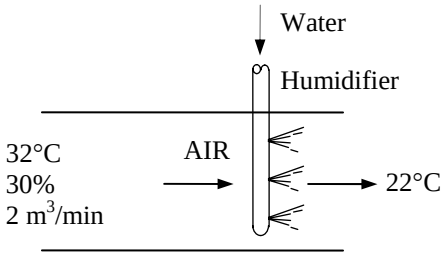
$$\dot{m}_{\text{supply}} = \dot{m}_{w2} - \dot{m}_{w1} = \dot{m}_a(\omega_2 - \omega_1) = (13.9\ \text{lbm/min})(0.0178 - 0.0123) = \mathbf{0.076\ lbm/min}$$

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14-100 Air is cooled by an evaporative cooler. The final relative humidity and the amount of water added are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31) at 32°C and 30% relative humidity we read

$$T_{wb1} = 19.4°C$$
$$\omega_1 = 0.0089 \text{ kg } H_2O/\text{kg dry air}$$
$$\upsilon_1 = 0.877 \text{ m}^3/\text{kg dry air}$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is,

$$T_{wb2} \cong T_{wb1} = 19.4°C$$

At this wet-bulb temperature and 22°C temperature we read

$$\phi_2 = \mathbf{79\%}$$
$$\omega_2 = 0.0130 \text{ kg } H_2O/\text{kg dry air}$$

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{\upsilon_1} = \frac{5 \text{ m}^3/\text{min}}{0.877 \text{ m}^3/\text{kg dry air}} = 5.70 \text{ kg/min}$$

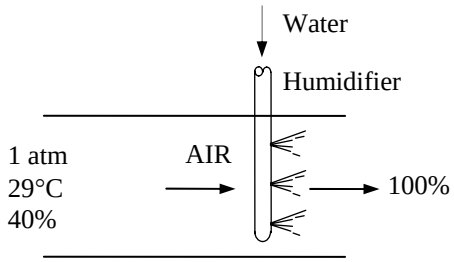
Then the required rate of water supply to the evaporative cooler is determined from

$$\dot{m}_{supply} = \dot{m}_{w2} - \dot{m}_{w1} = \dot{m}_a(\omega_2 - \omega_1) = (5.70 \text{ kg/min})(0.0130 - 0.0089) = \mathbf{0.0234 \text{ kg/min}}$$

14-101 Air enters an evaporative cooler at a specified state and relative humidity. The lowest temperature that air can attain is to be determined.

Analysis From the psychrometric chart (Fig. A-31) at 29°C and 40% relative humidity we read

$$T_{wb1} = 19.3°C$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature, which is the lowest temperature that can be obtained in an evaporative cooler. That is,

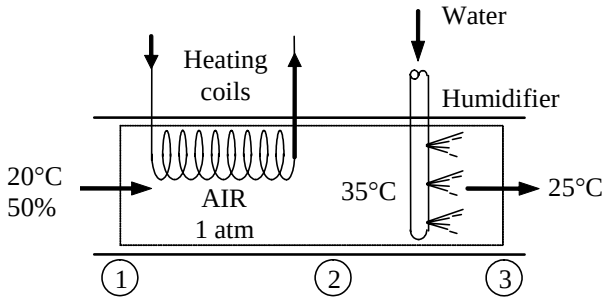
$$T_{min} = T_{wb1} = \mathbf{19.3°C}$$

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14-102 Air is first heated in a heating section and then passed through an evaporative cooler. The exit relative humidity and the amount of water added are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31 or EES) at 20°C and 50% relative humidity we read

$$\omega_1 = 0.00726 \text{ kg H}_2\text{O/kg dry air}$$



The specific humidity ω remains constant during the heating process. Therefore, $\omega_2 = \omega_1 = 0.00726$ kg H_2O / kg dry air. At this ω value and 35°C we read $T_{wb2} = 19.1$°C.

Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is, $T_{wb3} \cong T_{wb2} =$ 19.1°C. At this T_{wb} value and 25°C we read

$$\phi_3 = \mathbf{57.5\%}$$
$$\omega_3 = 0.0114 \text{ kg H}_2\text{O/kg dry air}$$

(*b*) The amount of water added to the air per unit mass of air is

$$\Delta\omega_{23} = \omega_3 - \omega_2 = 0.0114 - 0.00726 = \mathbf{0.00413 \text{ kg H}_2\text{O/kg dry air}}$$

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Adiabatic Mixing of Airstreams

14-103C This will occur when the straight line connecting the states of the two streams on the psychrometric chart crosses the saturation line.

14-104C Yes.

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14-105 Two airstreams are mixed steadily. The specific humidity, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture are to be determined.

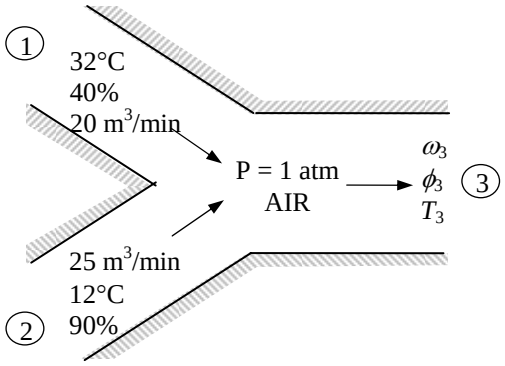
Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 62.7 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0119 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.882 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 31.9 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0079 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_2 = 0.819 \text{ m}^3\text{/kg dry air}$$



Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{\nu_1} = \frac{20 \text{ m}^3/\text{min}}{0.882 \text{ m}^3/\text{kg dry air}} = 22.7 \text{ kg/min}$$
$$\dot{m}_{a2} = \frac{\dot{V}_2}{\nu_2} = \frac{25 \text{ m}^3/\text{min}}{0.819 \text{ m}^3/\text{kg dry air}} = 30.5 \text{ kg/min}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (22.7 + 30.5) \text{ kg/min} = 53.2 \text{ kg/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$
$$\frac{22.7}{30.5} = \frac{0.0079 - \omega_3}{\omega_3 - 0.0119} = \frac{31.9 - h_3}{h_3 - 62.7}$$

which yields,

$$\omega_3 = \mathbf{0.0096 \text{ kg H}_2\text{O / kg dry air}}$$
$$h_3 = 45.0 \text{ kJ / kg dry air}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

$$T_3 = \mathbf{20.6°C}$$
$$\phi_3 = \mathbf{63.4\%}$$
$$\nu_3 = 0.845 \text{ m}^3\text{/kg dry air}$$

Finally, the volume flow rate of the mixture is determined from

$$\dot{V}_3 = \dot{m}_{a3}\nu_3 = (53.2 \text{ kg/min})(0.845 \text{ m}^3/\text{kg}) = \mathbf{45.0 \text{ m}^3\text{/min}}$$

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14-106 Two airstreams are mixed steadily. The specific humidity, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture are to be determined.

Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Analysis The properties of each inlet stream are determined to be

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 32°C}} = (0.40)(4.760\ \text{kPa}) = 1.90\ \text{kPa}$$

$$P_{a1} = P_1 - P_{v1} = 90 - 1.90 = 88.10\ \text{kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(305\ \text{K})}{88.10\ \text{kPa}}$$

$$= 0.994\ \text{m}^3/\text{kg dry air}$$

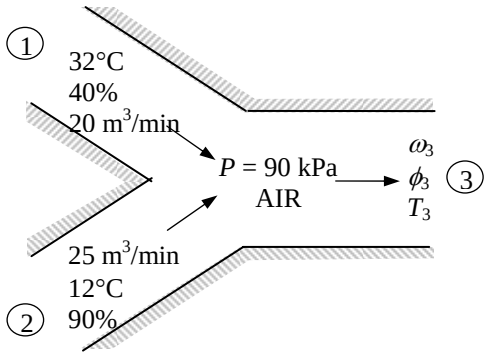
$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.90\ \text{kPa})}{(90-1.90)\ \text{kPa}}$$

$$= 0.0134\ \text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1}$$

$$= (1.005\ \text{kJ/kg}\cdot°\text{C})(32°\text{C}) + (0.0134)(2559.2\ \text{kJ/kg})$$

$$= 66.45\ \text{kJ/kg dry air}$$



and

$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat@12°C}} = (0.90)(1.403\ \text{kPa}) = 1.26\ \text{kPa}$$

$$P_{a2} = P_2 - P_{v2} = 90 - 1.26 = 88.74\ \text{kPa}$$

$$v_2 = \frac{R_a T_2}{P_{a2}} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(285\ \text{K})}{88.74\ \text{kPa}} = 0.922\ \text{m}^3/\text{kg dry air}$$

$$\omega_2 = \frac{0.622\, P_{v2}}{P_2 - P_{v2}} = \frac{0.622(1.26\ \text{kPa})}{(90-1.26)\ \text{kPa}} = 0.00883\ \text{kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\ \text{kJ/kg}\cdot°\text{C})(12°\text{C}) + (0.00883)(2522.9\ \text{kJ/kg}) = 34.34\ \text{kJ/kg dry air}$$

Then the mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{20\ \text{m}^3/\text{min}}{0.994\ \text{m}^3/\text{kg dry air}} = 20.12\ \text{kg/min}$$

$$\dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{25\ \text{m}^3/\text{min}}{0.922\ \text{m}^3/\text{kg dry air}} = 27.11\ \text{kg/min}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (20.12 + 27.11)\ \text{kg/min} = 47.23\ \text{kg/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1} \longrightarrow \frac{20.12}{27.11} = \frac{0.00883 - \omega_3}{\omega_3 - 0.0134} = \frac{34.34 - h_3}{h_3 - 66.45}$$

which yields

$$\omega_3 = \mathbf{0.0108\ kg\ H_2O/kg\ dry\ air}$$

$$h_3 = 48.02\ \text{kJ/kg dry air}$$

These two properties fix the state of the mixture. Other properties are determined from

$$h_3 = c_p T_3 + \omega_3 h_{g3} \cong c_p T_3 + \omega_3 (2501.3 + 1.82 T_3)$$

$$48.02\ \text{kJ/kg} = (1.005\ \text{kJ/kg}\cdot°\text{C})T_3 + (0.0108)(2500.9 + 1.82T_3)\ \text{kJ/kg} \longrightarrow T_3 = \mathbf{20.5°C}$$

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$$\omega_3 = \frac{0.622\,P_{v3}}{P_3 - P_{v3}}$$

$$0.0108 = \frac{0.622\,P_{v3}}{90 - P_{v3}} \longrightarrow P_{v3} = 1.54\,\text{kPa}$$

$$\phi_3 = \frac{P_{v3}}{P_{g3}} = \frac{P_{v3}}{P_{\text{sat}\,@\,T_3}} = \frac{1.54\,\text{kPa}}{2.41\,\text{kPa}} = 0.639 \quad \text{or} \quad \mathbf{63.9\%}$$

Finally,

$$P_{a3} = P_3 - P_{v3} = 90 - 1.54 = 88.46\,\text{kPa}$$

$$\nu_3 = \frac{R_a T_3}{P_{a3}} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293.5\,\text{K})}{88.46\,\text{kPa}} = 0.952\,\text{m}^3/\text{kg dry air}$$

$$\dot{V}_3 = \dot{m}_{a3}\nu_3 = (47.23\,\text{kg/min})(0.952\,\text{m}^3/\text{kg}) = \mathbf{45.0\,m^3/min}$$

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14-107 A stream of warm air is mixed with a stream of saturated cool air. The temperature, the specific humidity, and the relative humidity of the mixture are to be determined.

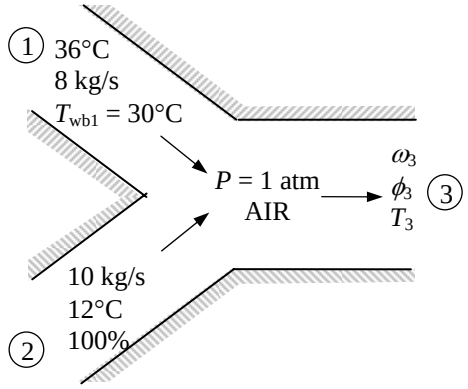
Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties The properties of each inlet stream are determined from the psychrometric chart (Fig. A-31 or EES) to be

$$h_1 = 99.4 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0246 \text{ kg H}_2\text{O/kg dry air}$$

and

$$h_2 = 34.1 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.00873 \text{ kg H}_2\text{O/kg dry air}$$



Analysis The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$
$$\frac{8}{10} = \frac{0.00873 - \omega_3}{\omega_3 - 0.0246} = \frac{34.1 - h_3}{h_3 - 99.4}$$

which yields,

(*b*) $\omega_3 = \mathbf{0.0158 \text{ kg H}_2\text{O/kg dry air}}$

$h_3 = 63.1 \text{ kJ/kg dry air}$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

(*a*) $T_3 = \mathbf{22.8°C}$

(*c*) $\phi_3 = \mathbf{90.1\%}$

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14-108 Problem 14-107 is reconsidered. The effect of the mass flow rate of saturated cool air stream on the mixture temperature, specific humidity, and relative humidity is to be investigated.

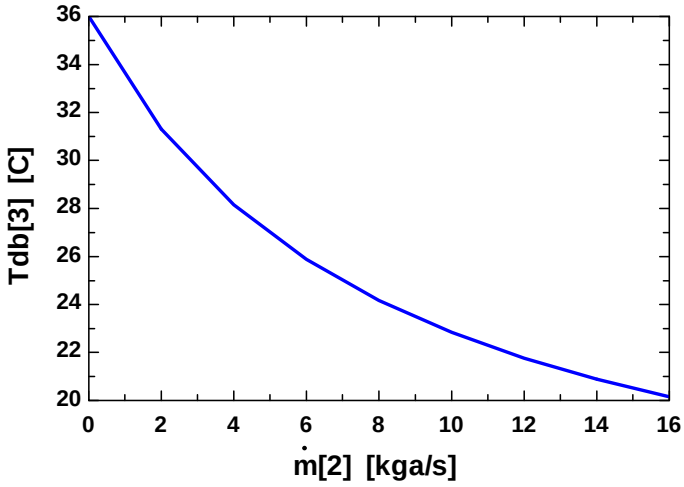
Analysis The problem is solved using EES, and the solution is given below.

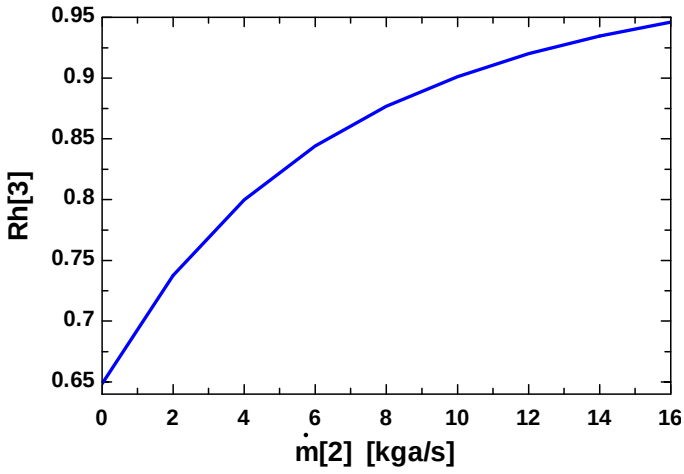
```
P=101.325 [kPa]
Tdb[1] =36 [C]
Twb[1] =30 [C]
m_dot[1] = 8 [kg/s]
Tdb[2] =12 [C]
Rh[2] = 1.0
m_dot[2] = 10 [kg/s]
P[1]=P
P[2]=P[1]
P[3]=P[1]

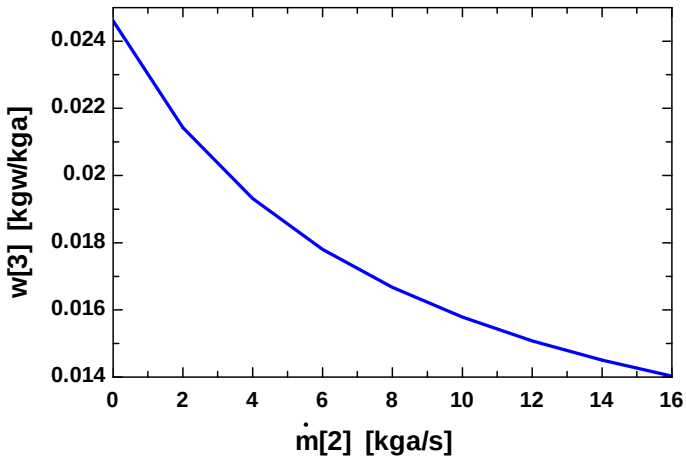
"Energy balance for the steady-flow mixing process:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams, we also neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kW]
E_dot_in = m_dot[1]*h[1]+m_dot[2]*h[2]
E_dot_out = m_dot[3]*h[3]
"Conservation of mass of dry air during mixing:"
m_dot[1]+m_dot[2] = m_dot[3]
"Conservation of mass of water vapor during mixing:"
m_dot[1]*w[1]+m_dot[2]*w[2] = m_dot[3]*w[3]
m_dot[1]=V_dot[1]/v[1]*convert(1/min,1/s)
m_dot[2]=V_dot[2]/v[2]*convert(1/min,1/s)
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],B=Twb[1])
Rh[1]=RELHUM(AirH2O,T=Tdb[1],P=P[1],B=Twb[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
Tdb[3]=TEMPERATURE(AirH2O,h=h[3],P=P[3],w=w[3])
Rh[3]=RELHUM(AirH2O,T=Tdb[3],P=P[3],w=w[3])
v[3]=VOLUME(AirH2O,T=Tdb[3],P=P[3],w=w[3])
Twb[2]=WETBULB(AirH2O,T=Tdb[2],P=P[2],R=RH[2])
Twb[3]=WETBULB(AirH2O,T=Tdb[3],P=P[3],R=RH[3])
m_dot[3]=V_dot[3]/v[3]*convert(1/min,1/s)
```

m_2 [kga/s]	Tdb_3 [C]	Rh_3	w_3 [kgw/kga]
0	36	0.6484	0.02461
2	31.31	0.7376	0.02143
4	28.15	0.7997	0.01931
6	25.88	0.8442	0.0178
8	24.17	0.8768	0.01667
10	22.84	0.9013	0.01579
12	21.77	0.92	0.01508
14	20.89	0.9346	0.0145
16	20.15	0.9461	0.01402

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14-109E Two airstreams are mixed steadily. The mass flow ratio of the two streams for a specified mixture relative humidity and the relative humidity of the mixture are to be determined.

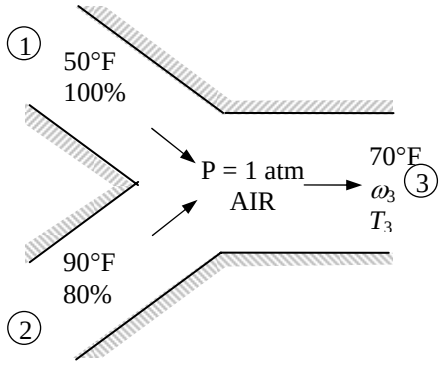
Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31E or from EES) to be

$$h_1 = 20.3 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0076 \text{ lbm H}_2\text{O/lbm dry air}$$

and

$$h_2 = 48.7 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0246 \text{ lbm H}_2\text{O/lbm dry air}$$



Analysis An application of Eq. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams gives

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$
$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{0.0246 - \omega_3}{\omega_3 - 0.0076} = \frac{48.7 - h_3}{h_3 - 20.3}$$

This equation cannot be solved directly. An iterative solution is needed. A mixture relative humidity ϕ_3 is selected. At this relative humidity and the given temperature (70°F), specific humidity and enthalpy are read from the psychrometric chart. These values are substituted into the above equation. If the equation is not satisfied, a new value of ϕ_3 is selected. This procedure is repeated until the equation is satisfied. Alternatively, EES software can be used. We used the following EES program to get these results:

$$\phi_3 = \mathbf{100\%}$$
$$\omega_3 = 0.0158 \text{ lbm H}_2\text{O/lbm dry air}$$
$$h_3 = 34.0 \text{ Btu/lbm dry air}$$
$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \mathbf{1.07}$$

```
"Given"
P=14.696 [psia]
T_1=50 [F]
phi_1=1.0
T_2=90 [F]
phi_2=0.80
T_3=70 [F]

"Analysis"
Fluid$='AirH2O'
"1st stream properties"
h_1=enthalpy(Fluid$, T=T_1, P=P, R=phi_1)
w_1=humrat(Fluid$, T=T_1, P=P, R=phi_1)
"2nd stream properties"
h_2=enthalpy(Fluid$, T=T_2, P=P, R=phi_2)
w_2=humrat(Fluid$, T=T_2, P=P, R=phi_2)
(w_2-w_3)/(w_3-w_1)=(h_2-h_3)/(h_3-h_1)
Ratio=(w_2-w_3)/(w_3-w_1)
"mixture properties"
phi_3=relhum(Fluid$, h=h_3, P=P, T=T_3)
h_3=enthalpy(Fluid$, R=phi_3, P=P, T=T_3)
```

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14-110 Two airstreams are mixed steadily. The temperature and the relative humidity of the mixture are to be determined.

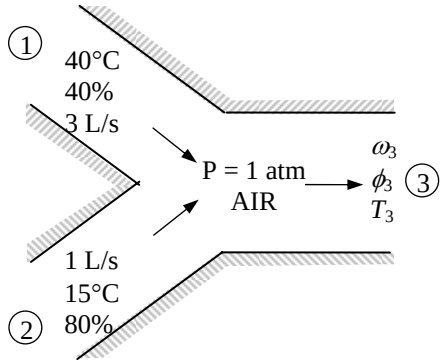
Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31 or from EES) to be

$$h_1 = 88.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0187 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.914 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 36.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0085 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_2 = 0.828 \text{ m}^3\text{/kg dry air}$$



Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{\nu_1} = \frac{0.003 \text{ m}^3/\text{s}}{0.914 \text{ m}^3/\text{kg dry air}} = 0.003282 \text{ kg/s}$$
$$\dot{m}_{a2} = \frac{\dot{V}_2}{\nu_2} = \frac{0.001 \text{ m}^3/\text{s}}{0.828 \text{ m}^3/\text{kg dry air}} = 0.001208 \text{ kg/s}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (0.003282 + 0.001208) \text{ kg/s} = 0.00449 \text{ kg/s}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$
$$\frac{0.003282}{0.001208} = \frac{0.0085 - \omega_3}{\omega_3 - 0.0187} = \frac{36.7 - h_3}{h_3 - 88.5}$$

which yields

$$\omega_3 = 0.0160 \text{ kg H}_2\text{O/kg dry air}$$
$$h_3 = 74.6 \text{ kJ/kg dry air}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

$$T_3 = \mathbf{33.4°C}$$
$$\phi_3 = 0.493 = \mathbf{49.3\%}$$

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14-111 Two airstreams are mixed steadily. The rate of exergy destruction is to be determined.

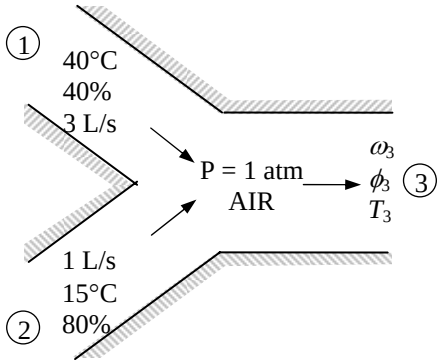
Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31 or from EES) to be

$$h_1 = 88.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0187 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.914 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 36.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0085 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_2 = 0.828 \text{ m}^3\text{/kg dry air}$$



The entropies of water vapor in the air streams are

$$s_{g1} = s_{g\,@\,40°\text{C}} = 8.2556 \text{ kJ/kg} \cdot \text{K}$$
$$s_{g2} = s_{g\,@\,15°\text{C}} = 8.7803 \text{ kJ/kg} \cdot \text{K}$$

Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{\nu_1} = \frac{0.003 \text{ m}^3/\text{s}}{0.914 \text{ m}^3/\text{kg dry air}} = 0.003282 \text{ kg/s}$$
$$\dot{m}_{a2} = \frac{\dot{V}_2}{\nu_2} = \frac{0.001 \text{ m}^3/\text{s}}{0.828 \text{ m}^3/\text{kg dry air}} = 0.001208 \text{ kg/s}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (0.003282 + 0.001208) \text{ kg/s} = 0.00449 \text{ kg/s}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$
$$\frac{0.003282}{0.001208} = \frac{0.0085 - \omega_3}{\omega_3 - 0.0187} = \frac{36.7 - h_3}{h_3 - 88.5}$$

which yields

$$\omega_3 = 0.0160 \text{ kg H}_2\text{O/kg dry air}$$
$$h_3 = 74.6 \text{ kJ/kg dry air}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

$$T_3 = 33.4°\text{C}$$
$$\phi_3 = 0.493$$

The entropy of water vapor in the mixture is

$$s_{g3} = s_{g\,@\,33.4°\text{C}} = 8.3833 \text{ kJ/kg} \cdot \text{K}$$

An entropy balance on the mixing chamber for the water gives

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$$\begin{aligned}\Delta\dot{S}_w &= \dot{m}_{a3}\omega_3 s_3 - \dot{m}_{a1}\omega_1 s_1 - \dot{m}_{a2}\omega_2 s_2 \\ &= 0.00449\times 0.0160\times 8.3833 - 0.003282\times 0.0187\times 8.2556 - 0.001208\times 0.0085\times 8.7803 \\ &= 5.426\times 10^{-6}\ \text{kW/K}\end{aligned}$$

The partial pressures of water vapor and dry air for all three air streams are

$$\begin{aligned}P_{v1} &= \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 40°C}} = (0.40)(7.3851\,\text{kPa}) = 2.954\,\text{kPa}\\ P_{a1} &= P_1 - P_{v1} = 101.325 - 2.954 = 98.37\,\text{kPa}\\ P_{v2} &= \phi_2 P_{g2} = \phi_2 P_{\text{sat @ 15°C}} = (0.80)(1.7057\,\text{kPa}) = 1.365\,\text{kPa}\\ P_{a2} &= P_2 - P_{v2} = 101.325 - 1.365 = 99.96\,\text{kPa}\\ P_{v3} &= \phi_3 P_{g3} = \phi_3 P_{\text{sat @ 33.4°C}} = (0.493)(5.150\,\text{kPa}) = 2.539\,\text{kPa}\\ P_{a3} &= P_3 - P_{v3} = 101.325 - 2.539 = 98.79\,\text{kPa}\end{aligned}$$

An entropy balance on the mixing chamber for the dry air gives

$$\begin{aligned}\Delta\dot{S}_a &= \dot{m}_{a1}(s_3 - s_1) + \dot{m}_{a2}(s_3 - s_2)\\ &= \dot{m}_{a1}\left(c_p \ln\frac{T_3}{T_1} - R\ln\frac{P_{a3}}{P_{a1}}\right) + \dot{m}_{a2}\left(c_p \ln\frac{T_3}{T_2} - R\ln\frac{P_{a3}}{P_{a2}}\right)\\ &= 0.003282\left[(1.005)\ln\frac{306.4}{313} - (0.287)\ln\frac{98.79}{98.37}\right] + 0.001208\left[(1.005)\ln\frac{306.4}{288} - (0.287)\ln\frac{98.79}{99.96}\right]\\ &= (0.003282)(-0.02264) + (0.001208)(0.06562)\\ &= 4.964\times 10^{-6}\ \text{kW/K}\end{aligned}$$

The rate of entropy generation is

$$\dot{S}_{\text{gen}} = \Delta\dot{S}_a + \Delta\dot{S}_w = 4.964\times 10^{-6} + 5.426\times 10^{-6} = 10.39\times 10^{-6}\ \text{kW/K}$$

Finally, the rate of exergy destruction is

$$\dot{X}_{\text{dest}} = T_0\dot{S}_{\text{gen}} = (298\,\text{K})(10.39\times 10^{-6}\ \text{kW/K}) = \mathbf{0.0031\,kW}$$

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Wet Cooling Towers

14-112C The working principle of a natural draft cooling tower is based on buoyancy. The air in the tower has a high moisture content, and thus is lighter than the outside air. This light moist air rises under the influence of buoyancy, inducing flow through the tower.

14-113C A spray pond cools the warm water by spraying it into the open atmosphere. They require 25 to 50 times the area of a wet cooling tower for the same cooling load.

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14-114 Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

Assumptions **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

Analysis (*a*) The mass flow rate of dry air through the tower remains constant ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$), but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

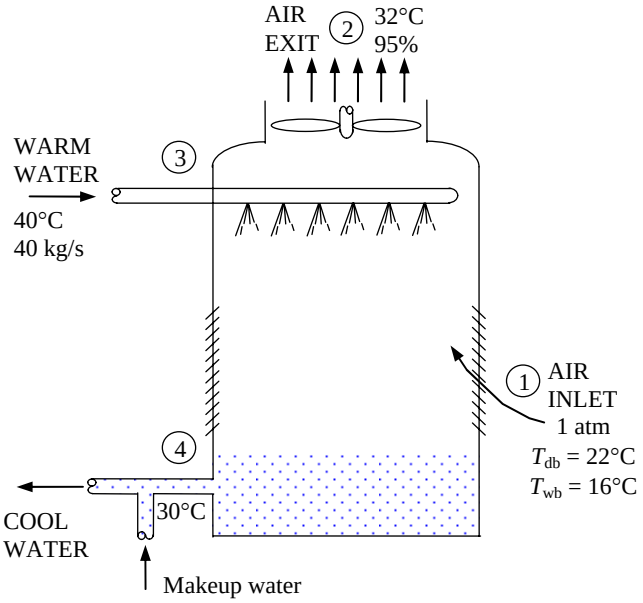
Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \quad \longrightarrow \quad \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$

Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$

From the psychrometric chart (Fig. A-31 or EES),

$$h_1 = 44.7 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.008875 \text{ kg H}_2\text{O/kg dry air}$$

$$\nu_1 = 0.848 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 106.6 \text{ kJ/kg dry air}$$

$$\omega_2 = 0.02905 \text{ kg H}_2\text{O/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f@40°\text{C}} = 167.53 \text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f@30°\text{C}} = 125.74 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(40 \text{ kg/s})(167.53 - 125.74)\text{kJ/kg}}{(106.6 - 44.7) \text{ kJ/kg} - (0.02905 - 0.008875)(125.74) \text{ kJ/kg}} = 28.17 \text{ kg/s}$$

Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a \nu_1 = (28.17 \text{ kg/s})(0.848 \text{ m}^3/\text{kg}) = \mathbf{23.9 \ m^3/s}$$

(*b*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (28.17 \text{ kg/s})(0.02905 - 0.008875) = \mathbf{0.568 \ kg/s}$$

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14-115 Water is cooled by air in a cooling tower. The relative humidity of the air at the exit and the water's exit temperature are to be determined.

Assumptions **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

Analysis The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

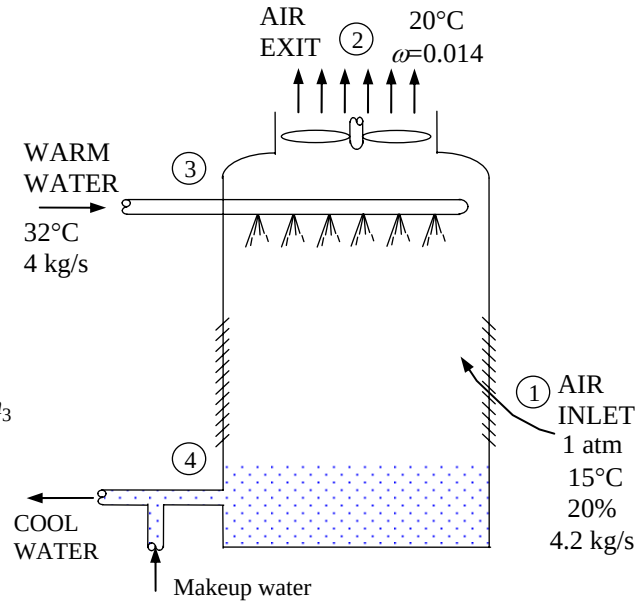
Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 - \dot{m}_3 h_3$$

Solving for h_4,

$$h_4 = \frac{\dot{m}_3 h_3 - \dot{m}_a (h_2 - h_1)}{\dot{m}_3 - \dot{m}_{\text{makeup}}}$$

From the psychrometric chart (Fig. A-31),

$$h_1 = 20.4 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.00211 \text{ kg H}_2\text{O/kg dry air}$$

$$\nu_1 = 0.819 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 55.7 \text{ kJ/kg dry air}$$

$$\phi_2 = 1 = \mathbf{100\%}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,32°\text{C}} = 134.1 \text{ kJ/kg H}_2\text{O}$$

Also, $\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (4.2 \text{ kg/s})(0.014 - 0.00211) = 0.050 \text{ kg/s}$

Substituting,

$$h_4 = \frac{\dot{m}_3 h_3 - \dot{m}_a (h_2 - h_1)}{\dot{m}_3 - \dot{m}_{\text{makeup}}} = \frac{(4)(134.1) - (0.12)(55.7 - 20.4)}{4 - 0.050} = 98.31 \text{ kJ/kg H}_2\text{O}$$

The exit temperature of the water is then (Table A-4)

$$T_4 = T_{\text{sat}\,@\,h_f = 98.31 \text{ kJ/kg}} = \mathbf{23.4°C}$$

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14-116 Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

Assumptions **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

Analysis (*a*) The mass flow rate of dry air through the tower remains constant ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$), but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e}$$

$$\dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

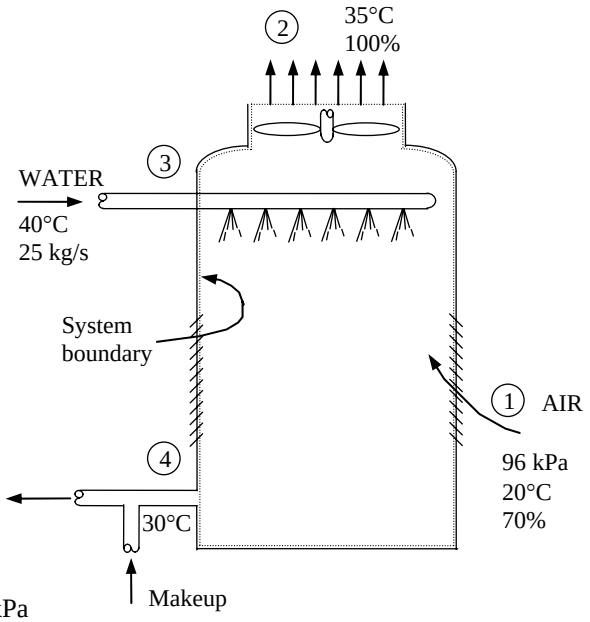
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$



The properties of air at the inlet and the exit are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 20°C}} = (0.70)(2.3392 \text{ kPa}) = 1.637 \text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 96 - 1.637 = 94.363 \text{ kPa}$$

$$\nu_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})}{94.363 \text{ kPa}} = 0.891 \text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.637 \text{ kPa})}{(96 - 1.637) \text{ kPa}} = 0.0108 \text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005 \text{ kJ/kg} \cdot \text{°C})(20\text{°C}) + (0.0108)(2537.4 \text{ kJ/kg}) = 47.5 \text{ kJ/kg dry air}$$

and $$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat @ 35°C}} = (1.00)(5.6291 \text{ kPa}) = 5.6291 \text{ kPa}$$

$$\omega_2 = \frac{0.622\, P_{v2}}{P_2 - P_{v2}} = \frac{0.622(5.6291 \text{ kPa})}{(96 - 5.6291) \text{ kPa}} = 0.0387 \text{ kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005 \text{ kJ/kg} \cdot \text{°C})(35\text{°C}) + (0.0387)(2564.6 \text{ kJ/kg}) = 134.4 \text{ kJ/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,40\text{°C}} = 167.53 \text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f\,@\,30\text{°C}} = 125.74 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(25 \text{ kg/s})(167.53 - 125.74)\text{kJ/kg}}{(134.4 - 47.5) \text{ kJ/kg} - (0.0387 - 0.0108)(125.74) \text{ kJ/kg}} = 12.53 \text{ kg/s}$$

Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a \nu_1 = (12.53 \text{ kg/s})(0.891 \text{ m}^3/\text{kg}) = \mathbf{11.2\ m^3/s}$$

(*b*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (12.53 \text{ kg/s})(0.0387 - 0.0108) = \mathbf{0.35\ kg/s}$$

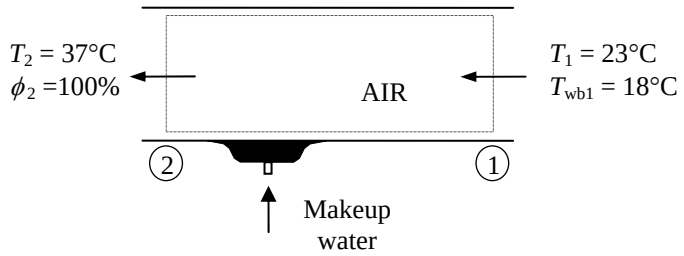
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14-117 A natural-draft cooling tower is used to remove waste heat from the cooling water flowing through the condenser of a steam power plant. The mass flow rate of the cooling water, the volume flow rate of air into the cooling tower, and the mass flow rate of the required makeup water are to be determined.

Assumptions **1** All processes are steady-flow and the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The inlet and exit states of the moist air for the tower are completely specified. The properties may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 50.74 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01085 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.8536 \text{ m}^3\text{/kg dry air}$$
$$h_2 = 142.83 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.04112 \text{ kg H}_2\text{O/kg dry air}$$



The enthalpies of cooling water at the inlet and exit of the condenser are (Table A-4)

$$h_{w3} = h_{f@\,40°C} = 167.53 \text{ kJ/kg}$$
$$h_{w4} = h_{f@\,26°C} = 109.01 \text{ kJ/kg}$$

The steam properties for the condenser are (Steam tables)

$$\left.\begin{matrix} P_{s1} = 200 \text{ kPa} \\ x_{s1} = 0 \end{matrix}\right\} h_{s1} = 504.71 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_{s2} = 10 \text{ kPa} \\ s_{s2} = 7.962 \text{ kJ/kg.K} \end{matrix}\right\} h_{s2} = 2524.3 \text{ kJ/kg}$$

$$\left.\begin{matrix} P_{s3} = 10 \text{ kPa} \\ x_{s1} = 0 \end{matrix}\right\} h_{s3} = 191.81 \text{ kJ/kg}$$

The mass flow rate of dry air is given by

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{\dot{V}_1}{0.8536 \text{ m}^3\text{/kg}}$$

The mass flow rates of vapor at the inlet and exit of the cooling tower are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01085)\frac{\dot{V}_1}{0.8536} = 0.01271\dot{V}_1$$

$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.04112)\frac{\dot{V}_1}{0.8536} = 0.04817\dot{V}_1$$

Mass and energy balances on the cooling tower give

$$\dot{m}_{v1} + \dot{m}_{cw3} = \dot{m}_{v2} + \dot{m}_{cw4}$$

$$\dot{m}_a h_1 + \dot{m}_{cw3} h_{w3} = \dot{m}_a h_2 + \dot{m}_{cw4} h_{w4}$$

The mass flow rate of the makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_{v2} - \dot{m}_{v1} = \dot{m}_{cw3} - \dot{m}_{cw4}$$

An energy balance on the condenser gives

$$0.18\dot{m}_s h_{s1} + 0.82\dot{m}_s h_{s2} + \dot{m}_{cw4} h_{w4} + \dot{m}_{\text{makeup}} h_{w4} = \dot{m}_s h_{s3} + \dot{m}_{cw3} h_{w3}$$

Solving all the above equations simultaneously with known and determined values using EES, we obtain

$$\dot{m}_{cw3} = \mathbf{1413 \text{ kg/s}}$$
$$\dot{V}_1 = \mathbf{47{,}700 \text{ m}^3\text{/min}}$$
$$\dot{m}_{\text{makeup}} = \mathbf{28.19 \text{ kg/s}}$$

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14-118E Water is cooled by air in a cooling tower. The mass flow rate of dry air is to be determined.

Assumptions **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

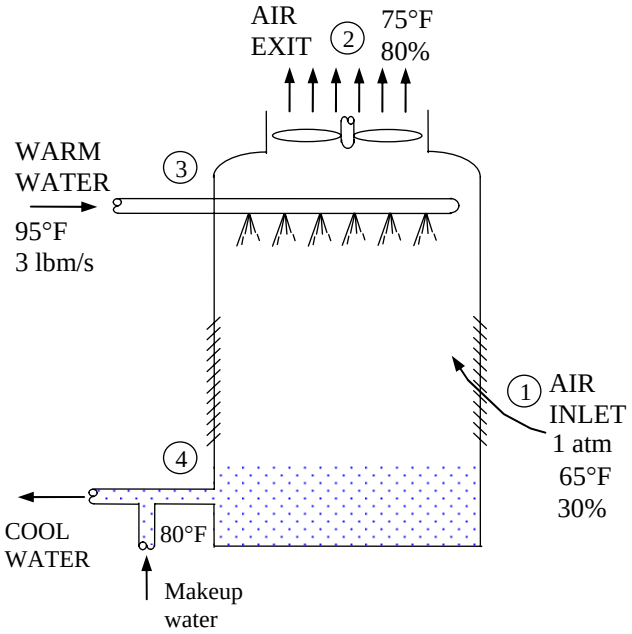
Analysis The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$
$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$
$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$
$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$
$$0 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 - \dot{m}_3 h_3$$

Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3 (h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1) h_4}$$

From the psychrometric chart (Fig. A-31E),

$$h_1 = 19.9 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.00391 \text{ lbm H}_2\text{O/lbm dry air}$$
$$\nu_1 = 13.31 \text{ ft}^3\text{/lbm dry air}$$

and

$$h_2 = 34.3 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0149 \text{ lbm H}_2\text{O/lbm dry air}$$

From Table A-4E,

$$h_3 \cong h_{f\,@\,95°\text{C}} = 63.04 \text{ Btu/lbm H}_2\text{O}$$
$$h_4 \cong h_{f\,@\,80°\text{C}} = 48.07 \text{ Btu/lbm H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(3 \text{ lbm/s})(63.04 - 48.07)\text{Btu/lbm}}{(34.3 - 19.9) \text{ Btu/lbm} - (0.0149 - 0.00391)(48.07) \text{ Btu/lbm}} = \mathbf{3.22 \text{ lbm/s}}$$

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14-119E Water is cooled by air in a cooling tower. The exergy lost in the cooling tower is to be determined.

Assumptions **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

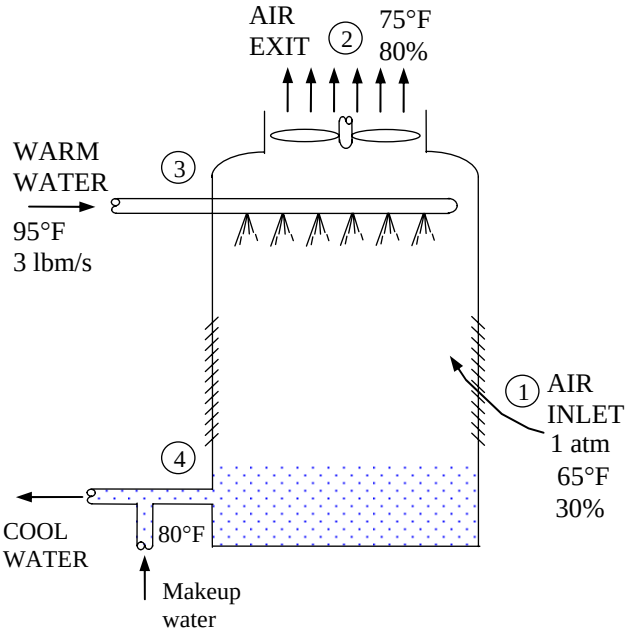
Analysis The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$
$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$
$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$
$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$
$$0 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 - \dot{m}_3 h_3$$

Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3 (h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1) h_4}$$

From the psychrometric chart (Fig. A-31E),

$$h_1 = 19.9 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.00391 \text{ lbm H}_2\text{O/lbm dry air}$$
$$\nu_1 = 13.31 \text{ ft}^3\text{/lbm dry air}$$

and

$$h_2 = 34.3 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0149 \text{ lbm H}_2\text{O/lbm dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,95°\text{F}} = 63.04 \text{ Btu/lbm H}_2\text{O}$$
$$h_4 \cong h_{f\,@\,80°\text{F}} = 48.07 \text{ Btu/lbm H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(3 \text{ lbm/s})(63.04 - 48.07)\text{Btu/lbm}}{(34.3 - 19.9) \text{ Btu/lbm} - (0.0149 - 0.00391)(48.07) \text{ Btu/lbm}} = 3.22 \text{ lbm/s}$$

The mass of water stream at state 3 per unit mass of dry air is

$$m_3 = \frac{\dot{m}_3}{\dot{m}_a} = \frac{3 \text{ lbm water/s}}{3.22 \text{ lbm dry air/s}} = 0.9317 \text{ lbm water/lbm dry air}$$

The mass flow rate of water stream at state 4 per unit mass of dry air is

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$$m_4 = m_3 - (\omega_2 - \omega_1) = 0.9317 - (0.0149 - 0.00391) = 0.9207 \text{ lbm water/lbm dry air}$$

The entropies of water streams are

$$s_3 = s_{f\,@\,95°\text{F}} = 0.12065 \text{ Btu/lbm} \cdot \text{R}$$
$$s_4 = s_{f\,@\,80°\text{F}} = 0.09328 \text{ Btu/lbm} \cdot \text{R}$$

The entropy change of water stream is

$$\Delta s_{\text{water}} = m_4 s_4 - m_3 s_3 = 0.9207 \times 0.09328 - 0.9317 \times 0.12065 = -0.02653 \text{ Btu/R} \cdot \text{lbm dry air}$$

The entropies of water vapor in the air stream are

$$s_{g1} = s_{g\,@\,65°\text{F}} = 2.0788 \text{ Btu/lbm} \cdot \text{R}$$
$$s_{g2} = s_{g\,@\,80°\text{F}} = 2.0352 \text{ Btu/lbm} \cdot \text{R}$$

The entropy change of water vapor in the air stream is

$$\Delta s_{\text{vapor}} = \omega_2 s_{g2} - \omega_1 s_{g1} = 0.0149 \times 2.0352 - 0.00391 \times 2.0788 = 0.02220 \text{ Btu/R} \cdot \text{lbm dry air}$$

The partial pressures of water vapor and dry air for air streams are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat}\,@\,65°\text{F}} = (0.30)(0.30578 \text{ psia}) = 0.0917 \text{ psia}$$
$$P_{a1} = P_1 - P_{v1} = 14.696 - 0.0917 = 14.60 \text{ psia}$$
$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat}\,@\,75°\text{F}} = (0.80)(0.43016 \text{ psia}) = 0.3441 \text{ psia}$$
$$P_{a2} = P_2 - P_{v2} = 14.696 - 0.3441 = 14.35 \text{ psia}$$

The entropy change of dry air is

$$\Delta s_a = s_2 - s_1 = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_{a2}}{P_{a1}}$$
$$= (0.240) \ln \frac{535}{525} - (0.06855) \ln \frac{14.35}{14.60} = 0.005712 \text{ Btu/lbm dry air}$$

The entropy generation in the cooling tower is the total entropy change:

$$s_{\text{gen}} = \Delta s_{\text{water}} + \Delta s_{\text{vapor}} + \Delta s_a = -0.02653 + 0.02220 + 0.005712 = 0.001382 \text{ Btu/R} \cdot \text{lbm dry air}$$

Finally, the exergy destruction per unit mass of dry air is

$$x_{\text{dest}} = T_0 s_{\text{gen}} = (525 \text{ R})(0.001382 \text{ Btu/R} \cdot \text{lbm dry air}) = \mathbf{0.726 \text{ Btu/lbm dry air}}$$

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Review Problems

14-120 Air is compressed by a compressor and then cooled to the ambient temperature at high pressure. It is to be determined if there will be any condensation in the compressed air lines.

Assumptions The air and the water vapor are ideal gases.

Properties The saturation pressure of water at 20°C is 2.3392 kPa (Table A-4)..

Analysis The vapor pressure of air before compression is

$$P_{v1} = \phi_1 P_g = \phi_1 P_{\text{sat @ 25°C}} = (0.50)(2.3392\text{ kPa}) = 1.17\text{ kPa}$$

The pressure ratio during the compression process is (800 kPa)/(92 kPa) = 8.70. That is, the pressure of air and any of its components increases by 8.70 times. Then the vapor pressure of air after compression becomes

$$P_{v2} = P_{v1} \times (\text{Pressure ratio}) = (1.17\text{ kPa})(8.70) = 10.2\text{ kPa}$$

The dew-point temperature of the air at this vapor pressure is

$$T_{\text{dp}} = T_{\text{sat @ } P_{v2}} = T_{\text{sat @ 10.2 kPa}} = 46.1°\text{C}$$

which is greater than 20°C. Therefore, part of the moisture in the compressed air will **condense** when air is cooled to 20°C.

14-121E The mole fraction of the water vapor at the surface of a lake and the mole fraction of water in the lake are to be determined and compared.

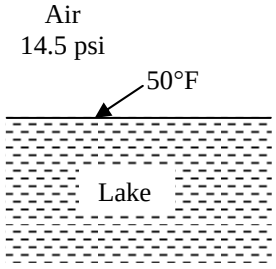
Assumptions **1** Both the air and water vapor are ideal gases. **2** Air is weakly soluble in water and thus Henry's law is applicable.

Properties The saturation pressure of water at 60°F is 0.2564 psia (Table A-4E). Henry's constant for air dissolved in water at 290 K (60ºF) is given in Table 16-2 to be H = 62,000 bar. Since we do not have the value at 50ºF, we may use this value.

Analysis The air at the water surface will be saturated. Therefore, the partial pressure of water vapor in the air at the lake surface will simply be the saturation pressure of water at 60°F,

$$P_{\text{vapor}} = P_{\text{sat @ 50°F}} = 0.1781\text{ psia}$$

Assuming both the air and vapor to be ideal gases, the mole fraction of water vapor in the air at the surface of the lake is determined to be



$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{0.1781\text{ psia}}{14.5\text{ psia}} = \mathbf{0.0123\ \ (or\ 1.23\ \ percent)}$$

The partial pressure of dry air just above the lake surface is

$$P_{\text{dry air}} = P - P_{\text{vapor}} = 14.5 - 0.1781 = 14.32\text{ psia}$$

Then the mole fraction of air in the water becomes

$$y_{\text{dry air, liquid side}} = \frac{P_{\text{dry air, gas side}}}{\text{H}} = \frac{14.32\text{ psia}(1\text{atm}/14.696\text{ psia})}{62{,}000\text{ bar}(1\text{ atm}/1.01325\text{ bar})} = 1.593 \times 10^{-5}$$

which is very small, as expected. Therefore, the mole fraction of water in the lake near the surface is

$$y_{\text{water, liquid side}} = 1 - y_{\text{dry air, liquid side}} = 1 - 1.593 \times 10^{-5} \cong \mathbf{1.0}$$

Discussion The concentration of air in water just below the air-water interface is 1.59 moles per 100,000 moles. The amount of air dissolved in water will decrease with increasing depth.

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14-122E A room is cooled adequately by a 7500 Btu/h air-conditioning unit. If the room is to be cooled by an evaporative cooler, the amount of water that needs to be supplied to the cooler is to be determined.

Assumptions **1** The evaporative cooler removes heat at the same rate as the air conditioning unit. **2** Water evaporates at an average temperature of 70°F.

Properties The enthalpy of vaporization of water at 70°F is 1053.7 Btu/lbm (Table A-4E).

Analysis Noting that 1 lbm of water removes 1053.7 Btu of heat as it evaporates, the amount of water that needs to evaporate to remove heat at a rate of 7500 Btu/h is determined from $\dot{Q} = \dot{m}_{\text{water}} h_{fg}$ to be

$$\dot{m}_{\text{water}} = \frac{\dot{Q}}{h_{fg}} = \frac{7500\ \text{Btu/h}}{1053.7\ \text{Btu/lbm}} = \mathbf{7.12\ lbm/h}$$

14-123E The required size of an evaporative cooler in cfm (ft^3/min) for an 8-ft high house is determined by multiplying the floor area of the house by 4. An equivalent rule is to be obtained in SI units.

Analysis Noting that 1 ft = 0.3048 m and thus 1 ft^2 = 0.0929 m^2 and 1 ft^3 = 0.0283 m^3, and noting that a flow rate of 4 ft^3/min is required per ft^2 of floor area, the required flow rate in SI units per m^2 of floor area is determined to

$$1\,\text{ft}^2 \leftrightarrow 4\,\text{ft}^3/\text{min}$$
$$0.0929\,\text{m}^2 \leftrightarrow 4 \times 0.0283\,\text{m}^3/\text{min}$$
$$1\,\text{m}^2 \leftrightarrow 1.22\,\text{m}^3/\text{min}$$

Therefore, a flow rate of **1.22 m^3/min** is required per m^2 of floor area.

14-124 A cooling tower with a cooling capacity of 220 kW is claimed to evaporate 9500 kg of water per day. It is to be determined if this is a reasonable claim.

Assumptions **1** Water evaporates at an average temperature of 30°C. **2** The coefficient of performance of the air-conditioning unit is COP = 3.

Properties The enthalpy of vaporization of water at 30°C is 2429.8 kJ/kg (Table A-4).

Analysis Using the definition of COP, the electric power consumed by the air conditioning unit when running is

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_{\text{cooling}}}{\text{COP}} = \frac{220\ \text{kW}}{3} = 73.33\ \text{kW}$$

Then the rate of heat rejected at the cooling tower becomes

$$\dot{Q}_{\text{rejected}} = \dot{Q}_{\text{cooling}} + \dot{W}_{in} = 220 + 73.33 = 293.3\ \text{kW}$$

Noting that 1 kg of water removes 2429.8 kJ of heat as it evaporates, the amount of water that needs to evaporate to remove heat at a rate of 293.3 kW is determined from $\dot{Q}_{\text{rejected}} = \dot{m}_{\text{water}} h_{fg}$ to be

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{rejected}}}{h_{fg}} = \frac{293.3\ \text{kJ/s}}{2429.8\ \text{kJ/kg}} = 0.1207\ \text{kg/s} = 434.6\ \text{kg/h} = 10{,}430\ \text{kg/day}$$

In practice, the air-conditioner will run intermittently rather than continuously at the rated power, and thus the water use will be less. Therefore, the claim amount of 9500 kg per day is **reasonable**.

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14-125E It is estimated that 190,000 barrels of oil would be saved per day if the thermostat setting in residences in summer were raised by 6°F (3.3°C). The amount of money that would be saved per year is to be determined.

Assumptions The average cooling season is given to be 120 days, and the cost of oil to be $20/barrel.

Analysis The amount of money that would be saved per year is determined directly from

(190,000 barrel/day)(120 days/year)($70/barrel) = **$1,596,000,000**

Therefore, the proposed measure will save more than one and half billion dollars a year.

14-126 Shading the condenser can reduce the air-conditioning costs by up to 10 percent. The amount of money shading can save a homeowner per year during its lifetime is to be determined.

Assumptions It is given that the annual air-conditioning cost is $500 a year, and the life of the air-conditioning system is 20 years.

Analysis The amount of money that would be saved per year is determined directly from

($500 / year)(20 years)(0.10) = **$1000**

Therefore, the proposed measure will save about $1000 during the lifetime of the system.

14-127 Air at a specified state is heated to to a specified temperature. The relative humidity after the heating is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The properties of the air at the ambient state are determined from the psychrometric chart (Figure A-31) to be

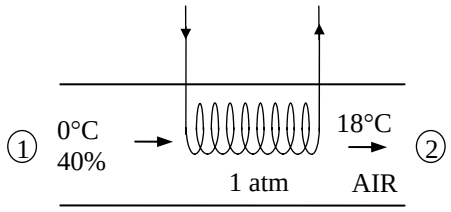
$$\omega_1 = 0.0015 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$

As the outside air infiltrates into the dacha, it does not gain or lose any water. Therefore the humidity ratio inside the dacha is the same as that outside,

$$\omega_2 = \omega_1 = 0.0015 \text{ kg H}_2\text{O/kg dry air}$$

Entering the psychrometry chart at this humidity ratio and the temperature inside the dacha gives

$$\phi_2 = 0.118 = \mathbf{11.8\%}$$



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14-128 Air is humidified by evaporating water into this air. The amount of heating per m^3 of air is to be determined.

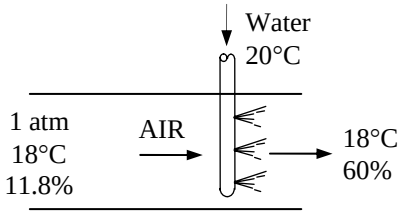
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 21.9 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.00151 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.8268 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 37.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.00773 \text{ kg H}_2\text{O/kg dry air}$$



Also,

$$h_w \cong h_{f\,@\,20°C} = 83.92 \text{ kJ/kg} \quad \text{(Table A-4)}$$

Analysis The amount of moisture in the air increases due to humidification ($\omega_2 > \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and humidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i + \dot{Q}_{\text{in}} = \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{in}} = \dot{m}_{a2}h_2 - \dot{m}_{a1}h_1 - \dot{m}_w h_w = \dot{m}_a(h_2 - h_1) - \dot{m}_w h_w$$
$$q_{\text{in}} = h_2 - h_1 - (\omega_2 - \omega_1)h_w$$
$$= (37.7 - 21.9)\text{kJ/kg} - (0.00773 - 0.00151)(83.92)$$
$$= 15.28 \text{ kJ/kg dry air}$$

The heat transfer per unit volume is

$$Q_{\text{in}} = \frac{q_{\text{in}}}{\nu_1} = \frac{15.28 \text{ kJ/kg dry air}}{0.827 \text{ m}^3\text{/kg dry air}} = \mathbf{18.5 \text{ kJ/m}^3}$$

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14-129E Air is cooled by evaporating water into this air. The amount of water required and the cooling produced are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31E) to be

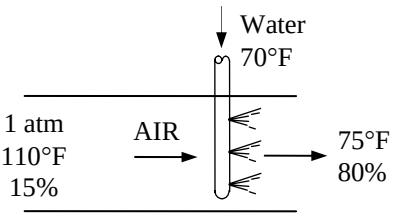
$h_1 = 35.6$ Btu/lbm dry air

$\omega_1 = 0.0082$ lbm H_2O/lbm dry air

and

$h_2 = 34.3$ Btu/lbm dry air

$\omega_2 = 0.0149$ lbm H_2O/lbm dry air



Also,

$h_w \cong h_{f\,@\,70°\mathrm{F}} = 38.08$ Btu/lbm (Table A-4E)

Analysis The amount of moisture in the air increases due to humidification ($\omega_2 > \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and humidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\Delta\omega = \omega_2 - \omega_1 = 0.0149 - 0.0082 = \mathbf{0.0067\ lbm\ H_2O/klbm\ dry\ air}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$

$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 + \dot{m}_w h_w - \dot{m}_{a2}h_2 = \dot{m}_a(h_1 - h_2) + \dot{m}_w h_w$$

$$q_{\text{out}} = h_1 - h_2 + (\omega_2 - \omega_1)h_w$$

$$= (335.6 - 34.3)\text{Btu/lbm} + (0.0067)(38.08)$$

$$= \mathbf{1.47\ Btu/lbm\ dry\ air}$$

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14-130E Air is humidified adiabatically by evaporating water into this air. The temperature of the air at the exit is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31E) to be

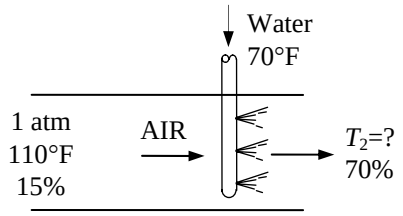
$$h_1 = 35.6 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0082 \text{ lbm H}_2\text{O/lbm dry air}$$

and $h_w \cong h_{f\,@\,70°\text{F}} = 38.08 \text{ Btu/lbm}$ (Table A-4E)

Analysis The amount of moisture in the air increases due to humidification ($\omega_2 > \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and humidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$
$$\dot{m}_w = \dot{m}_a(\omega_2 - \omega_1)$$



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_{a1}h_1 + \dot{m}_w h_w = \dot{m}_{a2}h_2$$
$$\dot{m}_w h_w = \dot{m}_a(h_2 - h_1)$$
$$(\omega_2 - \omega_1)h_w = h_2 - h_1$$

Substituting,

$$(\omega_2 - 0.0092)(83.92) = h_2 - 64.0$$

The solution of this equation requires a trial-error method. An air exit temperature is assumed. At this temperature and the given relative humidity, the enthalpy and specific humidity values are obtained from psychrometric chart and substituted into this equation. If the equation is not satisfied, a new value of exit temperature is assumed and this continues until the equation is satisfied. Alternatively, an equation solver such as EES may be used for the direct results. We used the following EES program to get these results:

$$T_2 = \mathbf{79.6°F}$$
$$h_2 = 35.8 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0152 \text{ lbm H}_2\text{O/lbm dry air}$$

```
"Given"
P=14.696 [psia]
T_1=110 [F]
phi_1=0.15
phi_2=0.70
T_w=70 [F]
"Analysis"
Fluid1$='AirH2O'
Fluid2$='steam_iapws'
h_1=enthalpy(Fluid1$, T=T_1, R=phi_1, P=P)
w_1=humrat(Fluid1$, T=T_1, R=phi_1, P=P)
h_2=enthalpy(Fluid1$, T=T_2, R=phi_2, P=P)
w_2=humrat(Fluid1$, T=T_2, R=phi_2, P=P)
h_w=enthalpy(Fluid2$, T=T_w, x=0)
q=0
q=h_1-h_2+(w_2-w_1)*h_w
```

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14-131E Air is cooled and dehumidified at constant pressure. The rate of cooling and the minimum humid air temperature required to meet this cooling requirement are to be determined.

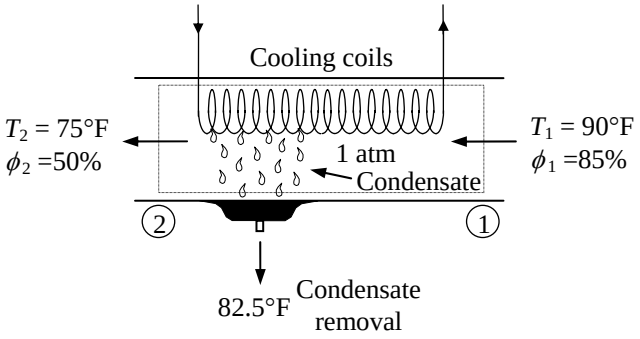
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 50.6 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0263 \text{ lbm H}_2\text{O/lbm dry air}$$
$$\nu_1 = 14.44 \text{ ft}^3\text{/lbm dry air}$$

and

$$h_2 = 28.2 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$



We assume that the condensate leaves this system at the average temperature of the air inlet and exit. Then,

$$h_w \cong h_{f\,@82.5°\text{F}} = 50.56 \text{ Btu/lbm} \quad \text{(Table A-4)}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass of air is

$$m_a = \frac{V_1}{\nu_1} = \frac{1000 \text{ ft}^3}{14.44 \text{ ft}^3 \text{/ lbm dry air}} = 69.25 \text{ lbm}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$m_w = m_a(\omega_1 - \omega_2) = (69.25 \text{ kg})(0.0263 - 0.0093) = 1.177 \text{ lbm}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$
$$Q_{\text{out}} = m_a(h_1 - h_2) - m_w h_w$$
$$Q_{\text{out}} = (69.25 \text{ kg})(50.6 - 28.2)\text{Btu/lbm} - (1.177 \text{ lbm})(50.56 \text{ Btu/lbm})$$
$$= \mathbf{1492 \text{ Btu}}$$

For the desired dehumidification, the air at the exit should be saturated with a specific humidity of 0.0093 lbm water/lbm dry air. That is,

$$\phi_2 = 1.0$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$

The temperature of the air at this state is the minimum air temperature required during this process:

$$T_2 = \mathbf{55.2°F}$$

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14-132E Air is cooled and dehumidified at constant pressure by a simple ideal vapor-compression refrigeration system. The system's COP is to be determined.

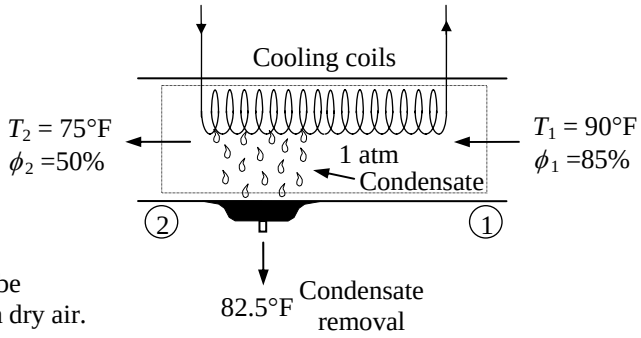
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 50.6 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0263 \text{ lbm H}_2\text{O/lbm dry air}$$
$$\nu_1 = 14.44 \text{ ft}^3\text{/lbm dry air}$$

and

$$h_2 = 28.2 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$



For the desired dehumidification, the air at the exit should be saturated with a specific humidity of 0.0093 lbm water/lbm dry air. That is,

$$\phi_2 = 1.0$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$

The temperature of the air at this state is the minimum air temperature required during this process:

$$T_{2,\min} = 55.2°\text{F}$$

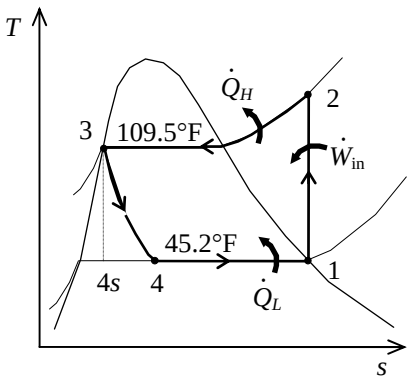
From the problem statement, the properties of R-134a at various states are (Tables A-11E through A-13E or from EES):

$$\left.\begin{array}{l} T_1 = 55.2 - 10 = 45.2°\text{F} \\ P_1 = P_{\text{sat @ 45.2°F}} = 55 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 55 \text{ psia}} = 109.49 \text{ Btu/lbm} \\ s_1 = s_{g\ @\ 55 \text{ psia}} = 0.22156 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} T_{\text{sat}} = 90 + 19.5 = 109.5°\text{F} \\ P_2 = P_{\text{sat @ 109.5°F}} = 160 \text{ psia} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 119.01 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 160 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \quad h_3 = h_{f\ @\ 160 \text{ psia}} = 48.52 \text{ Btu/lbm}$$

$$h_4 \cong h_3 = 48.52 \text{ Btu/lbm} \quad \text{(throttling)}$$



The COP of this system is then

$$\text{COP} = \frac{q_L}{w_{\text{in}}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{109.49 - 48.52}{119.01 - 109.49} = \mathbf{6.40}$$

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14-133 A tank contains saturated air at a specified state. The mass of the dry air, the specific humidity, and the enthalpy of the air are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The air is saturated, thus the partial pressure of water vapor is equal to the saturation pressure at the given temperature,

$$P_v = P_g = P_{\text{sat @ 20°C}} = 2.339\ \text{kPa}$$
$$P_a = P - P_v = 90 - 2.339 = 87.66\ \text{kPa}$$

1.8 m^3
20°C
90 kPa

Treating air as an ideal gas,

$$m_a = \frac{P_a V}{R_a T} = \frac{(87.66\ \text{kPa})(1.8\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = \mathbf{1.88\ kg}$$

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(2.339\ \text{kPa})}{(90 - 2.339)\ \text{kPa}} = \mathbf{0.0166\ kg\ H_2O/kg\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$
$$= (1.005\ \text{kJ/kg}\cdot\text{°C})(20\text{°C}) + (0.0166)(2537.4\ \text{kJ/kg})$$
$$= \mathbf{62.2\ kJ/kg\ dry\ air}$$

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14-134 Problem 14-133 is reconsidered. The properties of the air at the initial state are to be determined and the effects of heating the air at constant volume until the pressure is 110 kPa is to be studied.

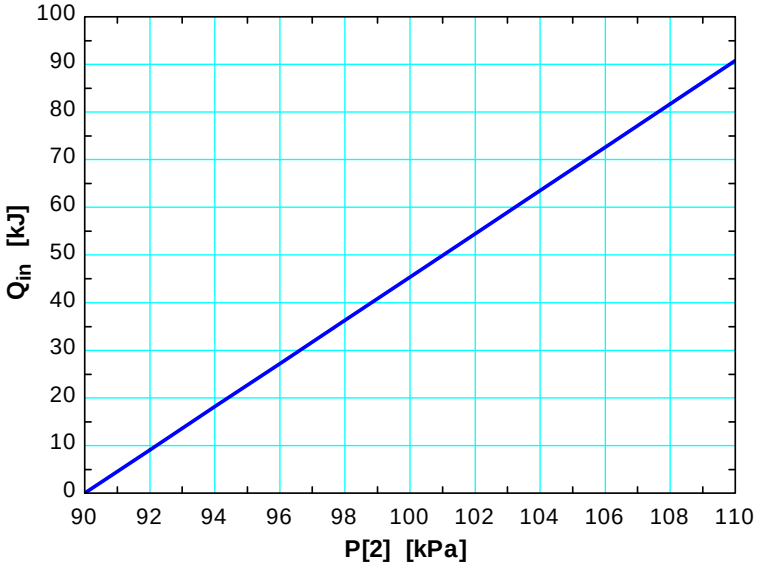
Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data:"
Tdb[1] = 20 [C]
P[1]=90 [kPa]
Rh[1]=1.0
P[2]=110 [kPa]
Vol = 1.8 [m^3]

w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
m_a=Vol/v[1]
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],w=w[1])
"Energy Balance for the constant volume tank:"
E_in - E_out = DELTAE_tank
DELTAE_tank=m_a*(u[2] -u[1])
E_in = Q_in
E_out = 0 [kJ]
u[1]=INTENERGY(AirH2O,T=Tdb[1],P=P[1],w=w[1])
u[2]=INTENERGY(AirH2O,T=Tdb[2],P=P[2],w=w[2])
"The ideal gas mixture assumption applied to the constant volume process yields:"
P[1]/(Tdb[1]+273)=P[2]/(Tdb[2]+273)
"The mass of the water vapor and dry air are constant, thus:"
w[2]=w[1]
Rh[2]=RELHUM(AirH2O,T=Tdb[2],P=P[2],w=w[2])
h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],w=w[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])

PROPERTIES AT THE INITIAL STATE
h[1]=62.25 [kJ/kga]
m_a=1.875 [kga]
v[1]=0.9599 [m^3/kga]
w[1]=0.01659 [kgw/kga]
```

P_2 [kPa]	Q_{in} [kJ]
90	0
92	9.071
94	18.14
96	27.22
98	36.29
100	45.37
102	54.44
104	63.52
106	72.61
108	81.69
110	90.78



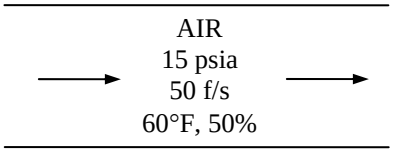
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14-135E Air at a specified state and relative humidity flows through a circular duct. The dew-point temperature, the volume flow rate of air, and the mass flow rate of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The vapor pressure of air is

$$P_v = \phi P_g = \phi P_{\text{sat @ 60°F}} = (0.50)(0.2564\ \text{psia}) = 0.128\ \text{psia}$$



Thus the dew-point temperature of the air is

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 0.128 psia}} = \mathbf{41.3°F} \ \text{(from EES)}$$

(*b*) The volume flow rate is determined from

$$\dot{\boldsymbol{V}} = VA = V\frac{\pi D^2}{4} = (50\ \text{ft/s})\left(\frac{\pi \times (8/12\ \text{ft})^2}{4}\right) = \mathbf{17.45\ ft^3/s}$$

(*c*) To determine the mass flow rate of dry air, we first need to calculate its specific volume,

$$P_a = P - P_v = 15 - 0.128 = 14.872\ \text{psia}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520\ \text{R})}{14.872\ \text{psia}} = 12.95\ \text{ft}^3/\text{lbm dry air}$$

Thus,

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{17.45\ \text{ft}^3/\text{s}}{12.95\ \text{ft}^3/\text{lbm dry air}} = \mathbf{1.35\ lbm/s}$$

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14-136 Air enters a cooling section at a specified pressure, temperature, and relative humidity. The temperature of the air at the exit and the rate of heat transfer are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

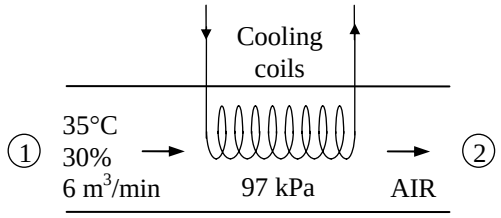
Analysis (*a*) The amount of moisture in the air also remains constant $(\omega_1 = \omega_2)$ as it flows through the cooling section since the process involves no humidification or dehumidification. The total pressure is 97 kPa. The properties of the air at the inlet state are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 35°C}} = (0.3)(5.629\ \text{kPa}) = 1.69\ \text{kPa}$$

$$P_{a1} = P_1 - P_{v1} = 97 - 1.69 = 95.31\ \text{kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\ \text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(308\ \text{K})}{95.31\ \text{kPa}}$$

$$= 0.927\ \text{m}^3/\text{kg dry air}$$



$$\omega_1 = \frac{0.622\ P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.69\ \text{kPa})}{(97 - 1.69)\ \text{kPa}} = 0.0110\ \text{kg H}_2\text{O/kg dry air}\ (= \omega_2)$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\ \text{kJ/kg°C})(35°\text{C}) + (0.0110)(2564.6\ \text{kJ/kg}) = 63.44\ \text{kJ/kg dry air}$$

The air at the final state is saturated and the vapor pressure during this process remains constant. Therefore, the exit temperature of the air must be the dew-point temperature,

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 1.69 kPa}} = \mathbf{14.8°C}$$

(*b*) The enthalpy of the air at the exit is

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\ \text{kJ/kg} \cdot °\text{C})(14.8°\text{C}) + (0.0110)(2528.1\ \text{kJ/kg}) = 42.78\ \text{kJ/kg dry air}$$

Also

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{6\ \text{m}^3/\text{s}}{0.927\ \text{m}^3/\text{kg dry air}} = 6.47\ \text{kg/min}$$

Then the rate of heat transfer from the air in the cooling section becomes

$$\dot{Q}_{\text{out}} = \dot{m}_a (h_1 - h_2) = (6.47\ \text{kg/min})(63.44 - 42.78)\text{kJ/kg} = \mathbf{134\ kJ/min}$$

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14-137 The outdoor air is first heated and then humidified by hot steam in an air-conditioning system. The rate of heat supply in the heating section and the mass flow rate of the steam required in the humidifying section are to be determined.

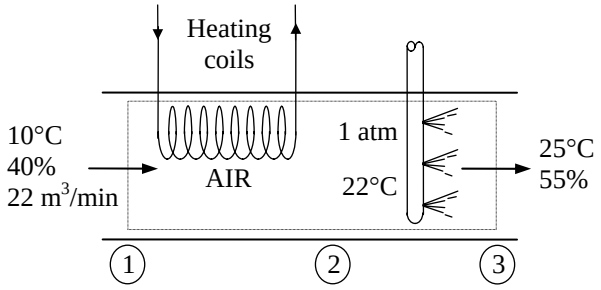
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The amount of moisture in the air also remains constants it flows through the heating section $(\omega_1 = \omega_2)$, but increases in the humidifying section $(\omega_3 > \omega_2)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 17.7 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0030 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$\nu_1 = 0.807 \text{ m}^3\text{/kg dry air}$$

$$h_2 = 29.8 \text{ kJ / kg dry air}$$
$$\omega_2 = \omega_1 = 0.0030 \text{ kg H}_2\text{O / kg dry air}$$

$$h_3 = 52.9 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0109 \text{ kg H}_2\text{O / kg dry air}$$



Analysis (*a*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{22 \text{ m}^3 / \text{min}}{0.807 \text{ m}^3 / \text{kg}} = 27.3 \text{ kg/min}$$

Then the rate of heat transfer to the air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (27.3 \text{ kg / min})(29.8 - 17.7)\text{kJ / kg} = \mathbf{330.3 \text{ kJ / min}}$$

(*b*) The conservation of mass equation for water in the humidifying section can be expressed as

$$\dot{m}_{a2}\omega_2 + \dot{m}_w = \dot{m}_{a3}\omega_3 \quad \text{or} \quad \dot{m}_w = \dot{m}_a(\omega_3 - \omega_2)$$

Thus,

$$\dot{m}_w = (27.3 \text{ kg / min})(0.0109 - 0.0030) = \mathbf{0.216 \text{ kg / min}}$$

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14-138 Air is cooled and dehumidified in an air-conditioning system with refrigerant-134a as the working fluid. The rate of dehumidification, the rate of heat transfer, and the mass flow rate of the refrigerant are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The saturation pressure of water at 30ºC is 4.2469 kPa. Then the dew point temperature of the incoming air stream at 30°C becomes

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ } 0.7\times 4.2469\,\text{kPa}} = 24°\text{C}$$

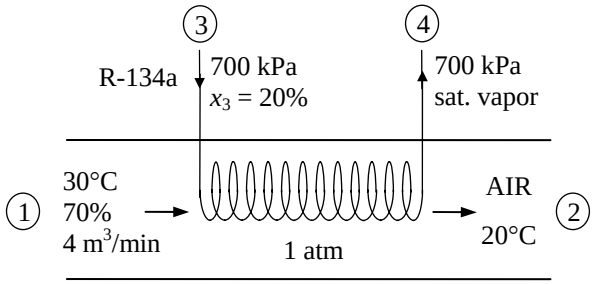
Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.

The mass flow rate of dry air remains constant during the entire process, but the amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Then the properties of the air at both states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 78.3\,\text{kJ/kg dry air}$$
$$\omega_1 = 0.0188\,\text{kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.885\,\text{m}^3\text{/kg dry air}$$

and

$$h_2 = 57.5\ \text{kJ/kg dry air}$$
$$\omega_2 = 0.0147\ \text{kg H}_2\text{O/kg dry air}$$



Also, $h_w \cong h_{f\,@\,20°\text{C}} = 83.915\,\text{kJ/kg}$ (Table A-4)

Then, $$\dot{m}_{a1} = \frac{\dot{\mathcal{V}}_1}{\nu_1} = \frac{4\,\text{m}^3/\text{min}}{0.885\,\text{m}^3/\text{kg dry air}} = 4.52\,\text{kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the refrigerant),

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \quad \longrightarrow \quad \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (4.52\ \text{kg/min})(0.0188 - 0.0147) = \mathbf{0.0185\ kg/min}$$

(*b*) *Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}\,^{\nearrow 0\ \text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{\text{out}} + \sum \dot{m}_e h_e \quad \longrightarrow \quad \dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (4.52\,\text{kg/min})(78.3 - 57.5)\text{kJ/kg} - (0.0185\,\text{kg/min})(83.915\,\text{kJ/kg}) = \mathbf{92.5\,kJ/min}$$

(*c*) The inlet and exit enthalpies of the refrigerant are

$$h_3 = h_g + x_3 h_{fg} = 88.82 + 0.2\times 176.21 = 124.06\,\text{kJ/kg}$$
$$h_4 = h_{g\,@\,700\,\text{kPa}} = 265.03\,\text{kJ/kg}$$

Noting that the heat lost by the air is gained by the refrigerant, the mass flow rate of the refrigerant becomes

$$\dot{Q}_R = \dot{m}_R(h_4 - h_3) \ \rightarrow \ \dot{m}_R = \frac{\dot{Q}_R}{h_4 - h_3} = \frac{92.5\,\text{kJ/min}}{(265.03 - 124.06)\,\text{kJ/kg}} = \mathbf{0.66\ kg/min}$$

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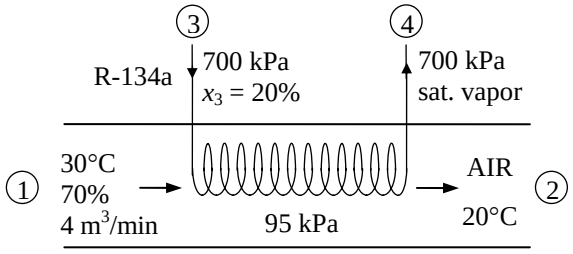
14-139 Air is cooled and dehumidified in an air-conditioning system with refrigerant-134a as the working fluid. The rate of dehumidification, the rate of heat transfer, and the mass flow rate of the refrigerant are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The dew point temperature of the incoming air stream at 30°C is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 30°C}} = (0.7)(4.247\ \text{kPa}) = 2.973\ \text{kPa}$$

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 2.973 kPa}} = 24°\text{C}$$



Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.

The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. The properties of the air at both states are determined to be

$$P_{a1} = P_1 - P_{v1} = 95 - 2.97 = 92.03\ \text{kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303\ \text{K})}{92.03\ \text{kPa}} = 0.945\ \text{m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(2.97\ \text{kPa})}{(95 - 2.97)\ \text{kPa}} = 0.0201\ \text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\ \text{kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.0201)(2555.6\ \text{kJ/kg}) = 81.50\ \text{kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = (1.00) P_{\text{sat @ 20°C}} = 2.3392\ \text{kPa}$$

$$\omega_2 = \frac{0.622\, P_{v2}}{P_2 - P_{v2}} = \frac{0.622(2.3392\ \text{kPa})}{(95 - 2.3392)\ \text{kPa}} = 0.0157\ \text{kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\ \text{kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0157)(2537.4\ \text{kJ/kg}) = 59.94\ \text{kJ/kg dry air}$$

Also, $h_w \cong h_{f\,@\,20°\text{C}} = 83.915\ \text{kJ/kg}$ (Table A-4)

Then, $$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{4\ \text{m}^3/\text{min}}{0.945\ \text{m}^3/\text{kg dry air}} = 4.23\ \text{kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the refrigerant),

Water Mass Balance: $$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (4.23\ \text{kg/min})(0.0201 - 0.0157) = \mathbf{0.0186\ kg/min}$$

(*b*) *Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \dot{Q}_{\text{out}} + \sum \dot{m}_e h_e \longrightarrow \dot{Q}_{\text{out}} = \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (4.23\ \text{kg/min})(81.50 - 59.94)\text{kJ/kg} - (0.0186\ \text{kg/min})(83.915\ \text{kJ/kg}) = \mathbf{89.7\ kJ/min}$$

(*c*) The inlet and exit enthalpies of the refrigerant are

$$h_3 = h_g + x_3 h_{fg} = 88.82 + 0.2 \times 176.21 = 124.06\ \text{kJ/kg}$$

$$h_4 = h_{g\,@\,700\ \text{kPa}} = 265.03\ \text{kJ/kg}$$

Noting that the heat lost by the air is gained by the refrigerant, the mass flow rate of the refrigerant is determined from

$$\dot{Q}_R = \dot{m}_R(h_4 - h_3)$$

$$\dot{m}_R = \frac{\dot{Q}_R}{h_4 - h_3} = \frac{89.7\ \text{kJ/min}}{(265.03 - 124.06)\ \text{kJ/kg}} = \mathbf{0.636\ kg/min}$$

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14-140 Air is heated and dehumidified in an air-conditioning system consisting of a heating section and an evaporative cooler. The temperature and relative humidity of the air when it leaves the heating section, the rate of heat transfer in the heating section, and the rate of water added to the air in the evaporative cooler are to be determined.

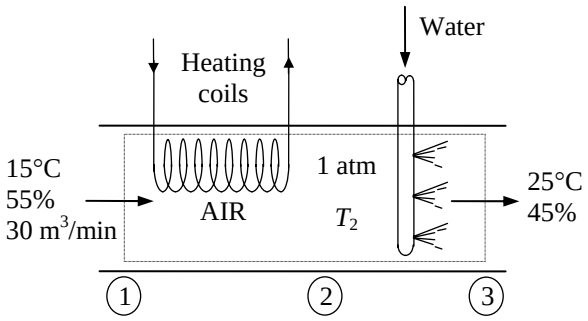
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) Assuming the wet-bulb temperature of the air remains constant during the evaporative cooling process, the properties of air at various states are determined from the psychrometric chart (Fig. A-31 or EES) to be

$$\left.\begin{matrix} T_1 = 15°\text{C} \\ \phi_1 = 55\% \end{matrix}\right\} \begin{matrix} h_1 = 29.8\ \text{kJ/kg dry air} \\ \omega_1 = 0.00581\ \text{kg/ H}_2\text{O/kg dry air} \\ \nu_1 = 0.824\ \text{m}^3/\text{kg} \end{matrix}$$

$$\left.\begin{matrix} \omega_2 = \omega_1 \\ T_{\text{wb2}} = T_{\text{wb3}} \end{matrix}\right\} \begin{matrix} T_2 = \mathbf{32.5°C} \\ \phi_2 = \mathbf{19.2\%} \\ h_2 \cong h_3 = 47.8\ \text{kJ/kg dry air} \end{matrix}$$

$$\left.\begin{matrix} T_3 = 25°\text{C} \\ \phi_3 = 45\% \end{matrix}\right\} \begin{matrix} h_3 = 47.8\ \text{kJ/kg dry air} \\ \omega_3 = 0.00888\ \text{kg/ H}_2\text{O/kg dry air} \\ T_{wb3} = 17.1°\text{C} \end{matrix}$$



(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{\nu_1} = \frac{30\ \text{m}^3/\text{min}}{0.824\ \text{m}^3/\text{kg dry air}} = 36.4\ \text{kg/min}$$

Then the rate of heat transfer to air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (36.4\ \text{kg/min})(47.8 - 29.8)\text{kJ/kg} = \mathbf{655\ kJ/min}$$

(*c*) The rate of water added to the air in evaporative cooler is

$$\dot{m}_{w,\text{added}} = \dot{m}_{w3} - \dot{m}_{w2} = \dot{m}_a (\omega_3 - \omega_2) = (36.4\ \text{kg/min})(0.00888 - 0.00581) = \mathbf{0.112\ kg/min}$$

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14-141 Problem 14-140 is reconsidered. The effect of total pressure in the range 94 to 100 kPa on the results required in the problem is to be studied.

Analysis The problem is solved using EES, and the solution is given below.

```
P=101.325 [kPa]
Tdb[1] =15 [C]
Rh[1] = 0.55
Vol_dot[1]= 30 [m^3/min]
Tdb[3] = 25 [C]
Rh[3] = 0.45
P[1]=P
P[2]=P[1]
P[3]=P[1]

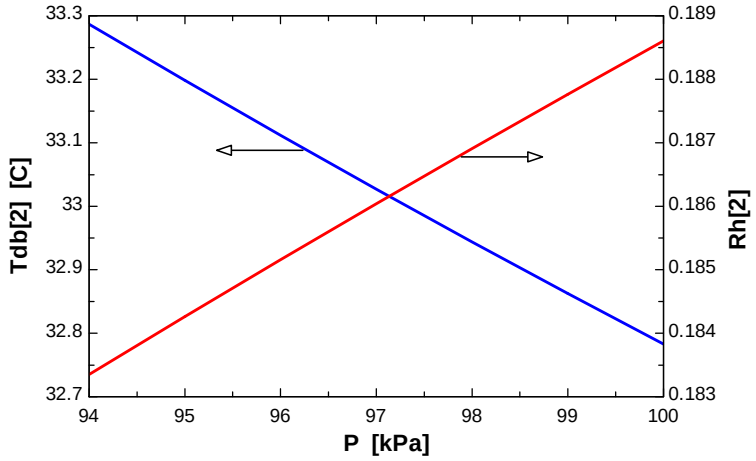
"Energy balance for the steady-flow heating process 1 to 2:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams, we also neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kJ/min]
E_dot_in = m_dot_a*h[1]+Q_dot_in
E_dot_out = m_dot_a*h[2]
"Conservation of mass of dry air during mixing: m_dot_a = constant"
m_dot_a = Vol_dot[1]/v[1]
"Conservation of mass of water vapor during the heating process:"
m_dot_a*w[1] = m_dot_a*w[2]
"Conservation of mass of water vapor during the evaporative cooler process:"
m_dot_a*w[2]+m_dot_w = m_dot_a*w[3]

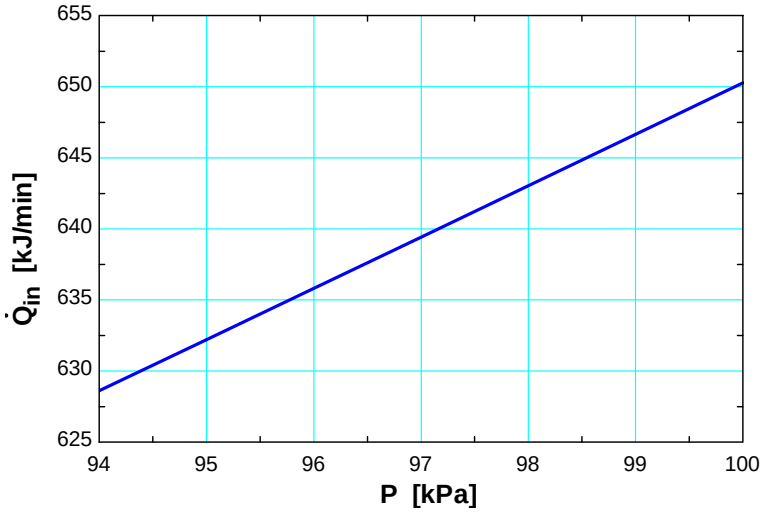
"During the evaporative cooler process:"
Twb[2] = Twb[3]
Twb[3] =WETBULB(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
{h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],B=Twb[2])}
h[2]=h[3]
Tdb[2]=TEMPERATURE(AirH2O,h=h[2],P=P[2],w=w[2])

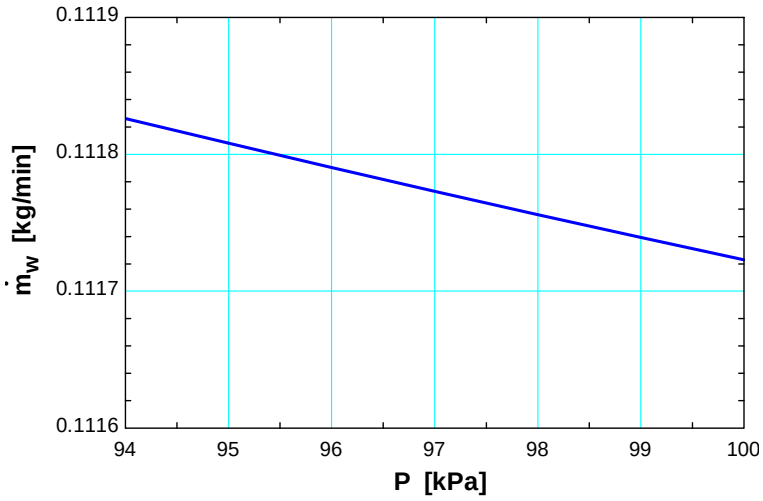
w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
h[3]=ENTHALPY(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
w[3]=HUMRAT(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
```

P [kPa]	m_w [kg/min]	Q_{in} [kJ/min]	Rh_2	Tdb_2 [C]
94	0.1118	628.6	0.1833	33.29
95	0.1118	632.2	0.1843	33.2
96	0.1118	635.8	0.1852	33.11
97	0.1118	639.4	0.186	33.03
98	0.1118	643	0.1869	32.94
99	0.1117	646.6	0.1878	32.86
100	0.1117	650.3	0.1886	32.78

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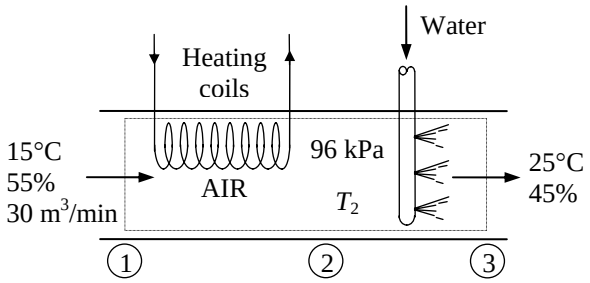


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14-142 Air is heated and dehumidified in an air-conditioning system consisting of a heating section and an evaporative cooler. The temperature and relative humidity of the air when it leaves the heating section, the rate of heat transfer in the heating section, and the rate at which water is added to the air in the evaporative cooler are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) Assuming the wet-bulb temperature of the air remains constant during the evaporative cooling process, the properties of air at various states are determined to be



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @15°C}} = (0.55)(1.7057\ \text{kPa}) = 0.938\ \text{kPa}$$

$$P_{a1} = P_1 - P_{v1} = 96 - 0.938 = 95.06\ \text{kPa}$$

$$\nu_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{95.06\ \text{kPa}}$$

$$= 0.8695\ \text{m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.8695\ \text{kPa})}{(96 - 0.8695)\ \text{kPa}} = 0.006138\ \text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\ \text{kJ/kg}\cdot\text{°C})(15\text{°C}) + (0.006138)(2528.3\ \text{kJ/kg}) = 30.59\ \text{kJ/kg dry air}$$

and

$$P_{v3} = \phi_3 P_{g3} = \phi_3 P_{\text{sat @ 25°C}} = (0.45)(3.17\ \text{kPa}) = 1.426\ \text{kPa}$$

$$P_{a3} = P_3 - P_{v3} = 96 - 1.426 = 94.57\ \text{kPa}$$

$$\omega_3 = \frac{0.622\, P_{v3}}{P_3 - P_{v3}} = \frac{0.622(1.426\ \text{kPa})}{(96 - 1.426)\ \text{kPa}} = 0.009381\ \text{kg H}_2\text{O/kg dry air}$$

$$h_3 = c_p T_3 + \omega_3 h_{g3} = (1.005\ \text{kJ/kg}\cdot\text{°C})(25\text{°C}) + (0.009381)(2546.5\ \text{kJ/kg})$$

$$= 49.01\ \text{kJ/kg dry air}$$

Also,

$$h_2 \cong h_3 = 49.01\ \text{kJ/kg}$$

$$\omega_2 = \omega_1 = 0.006138\ \text{kg H}_2\text{O/kg dry air}$$

Thus,

$$h_2 = c_p T_2 + \omega_2 h_{g2} \cong c_p T_2 + \omega_2 (2500.9 + 1.82 T_2) = (1.005\ \text{kJ/kg}\cdot\text{°C}) T_2 + (0.006138)(2500.9 + 1.82 T_2)$$

Solving for T_2,

$$T_2 = \mathbf{33.1°C} \longrightarrow P_{g2} = P_{\text{sat@33.1°C}} = 5.072\ \text{kPa}$$

Thus, $$\phi_2 = \frac{\omega_2 P_2}{(0.622 + \omega_2) P_{g2}} = \frac{(0.006138)(96)}{(0.622 + 0.006138)(5.072)} = 0.185 \text{ or } \mathbf{18.5\%}$$

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{\nu}_1}{\nu_1} = \frac{30\ \text{m}^3/\text{min}}{0.8695\ \text{m}^3/\text{kg dry air}} = 34.5\ \text{kg/min}$$

Then the rate of heat transfer to air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (34.5\ \text{kg/min})(49.01 - 30.59)\text{kJ/kg} = \mathbf{636\ kJ/min}$$

(*c*) The rate of water addition to the air in evaporative cooler is

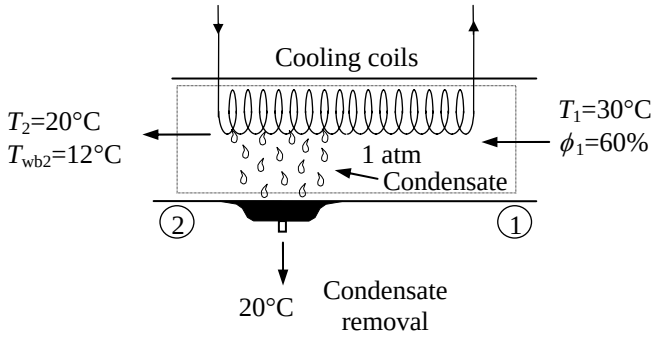
$$\dot{m}_{w,\text{added}} = \dot{m}_{w3} - \dot{m}_{w2} = \dot{m}_a (\omega_3 - \omega_2) = (34.5\ \text{kg/min})(0.009381 - 0.006138) = \mathbf{0.112\ kg/min}$$

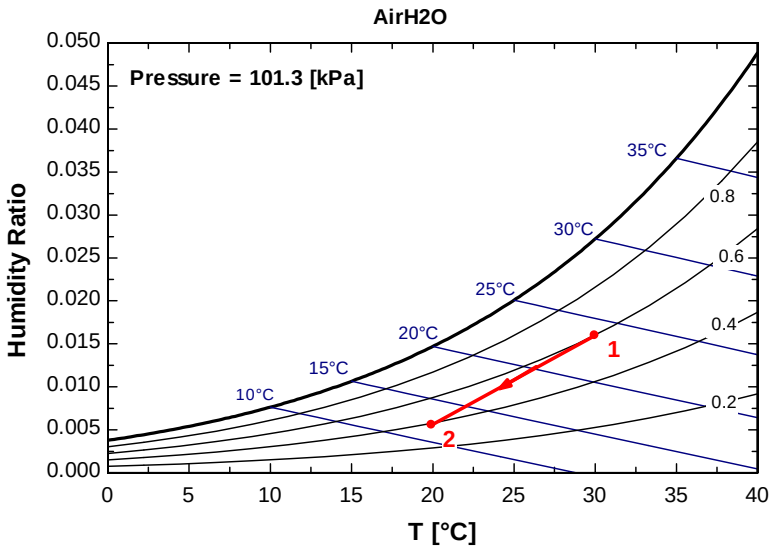
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14-143 Air is cooled, dehumidified and heated at constant pressure. The system hardware and the psychrometric diagram are to be sketched and the heat transfer is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (a) The schematic of the cooling and dehumidifaction process and the the process on the psychrometric chart are given below. The psychrometric chart is obtained from the Property Plot feature of EES.





(b) The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at the inlet and exit states are determined from the psychrometric chart (Figure A-31 or EES) to be

$$h_1 = 71.2 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0160 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.881 \text{ m}^3\text{/kg dry air}$$

and

$$\phi_2 = 0.376$$
$$h_2 = 33.9 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.00544 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_2 = 0.838 \text{ m}^3\text{/kg dry air}$$

Also,

$$h_w \cong h_{f\,@\,20°C} = 83.9 \text{ kJ/kg} \quad \text{(Table A-4)}$$

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The mass flow rate of air in the room to be replaced is

$$\dot{m}_a = \frac{\dot{V}_2}{v_2} = \frac{(700/60\ \text{m}^3/\text{s})}{0.838\ \text{m}^3/\text{kg}} = 13.93\ \text{kg/s}$$

The mass flow rate of outside air is

$$\dot{m}_{out} = \dot{m}_{out}(1+\omega_1) = (13.93\ \text{kg/s})(1+0.0160) = 14.15\ \text{kg/s} = \mathbf{50{,}940\ kg/h}$$

(c) Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$$

Substituting,

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (13.93\ \text{kg/s})(0.0160 - 0.00544) = 0.147\ \text{kg/s} = \mathbf{8.83\ kg/min}$$

(d) *Energy Balance*:

$$\begin{aligned}
\dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})} = 0 \\
\dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\
\sum \dot{m}_i h_i &= \dot{Q}_{out} + \sum \dot{m}_e h_e \\
\dot{Q}_{\text{out}} &= \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) \\
&= \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w
\end{aligned}$$

Noting that the cooling water absorbs this heat, the mass flow rate of cooling water is determined from

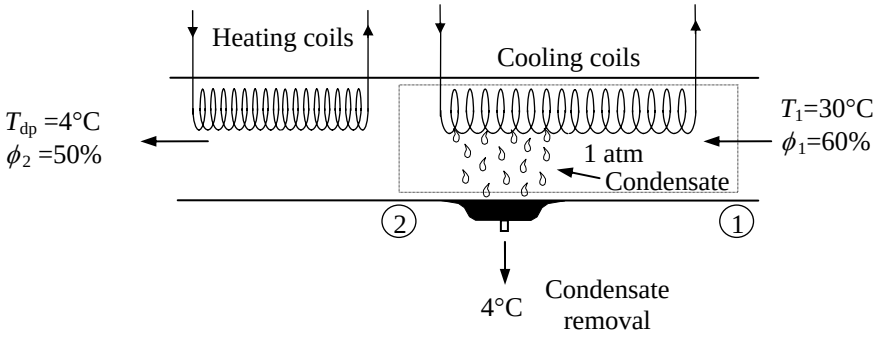
$$\begin{aligned}
\dot{m}_a(h_1 - h_2) - \dot{m}_w h_w &= \dot{m}_{cw} c_{pw} \Delta T_{cw} \\
\dot{m}_{cw} &= \frac{\dot{m}_a(h_1 - h_2) - \dot{m}_w h_w}{c_{pw}\Delta T_{cw}} \\
&= \frac{(13.93\ \text{kg/s})(71.2 - 33.9)\ \text{kJ/kg} - (0.147\ \text{kg/s})(83.9\ \text{kJ/kg})}{= (4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(15^\circ\text{C})} \\
&= 8.08\ \text{kg/s} \\
&= \mathbf{485\ kg/min}
\end{aligned}$$

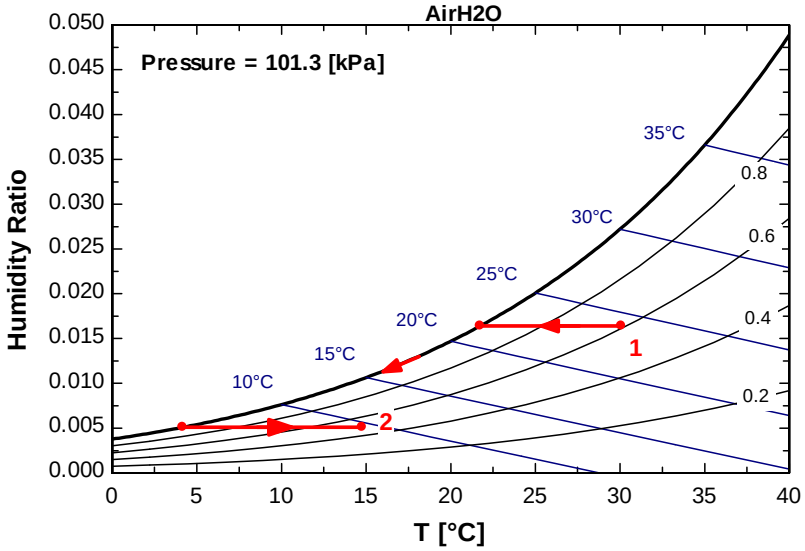
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14-144 Air is cooled, dehumidified and heated at constant pressure. The system hardware and the psychrometric diagram are to be sketched and the heat transfer is to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (a) The schematic of the cooling and dehumidifaction process and the the process on the psychrometric chart are given below. The psychrometric chart is obtained from the Property Plot feature of EES.





(b) The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at the inlet and exit states are determined from the psychrometric chart (Figure A-31 or EES) to be

$$T_{dp} = \mathbf{21.4°C}$$
$$h_1 = 71.2 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0160 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.881 \text{ m}^3\text{/kg dry air}$$

and

$$T_2 = 14.3°C$$
$$\phi_2 = 0.50$$
$$h_2 = 27.1 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.00504 \text{ kg H}_2\text{O/kg dry air}$$

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Also,

$$h_w \cong h_{f\,@\,4°C} = 16.8\,\text{kJ/kg} \quad \text{(Table A-4)}$$

The volume flow rate at the inlet is

$$\dot{V}_1 = \dot{m}_a v_1 = (100/60\,\text{kg/s})(0.881\,\text{m}^3/\text{kg}) = \mathbf{1.47\,m^3/s}$$

(c) Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$$

Energy Balance:

$$\begin{aligned}
\dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0\,\text{(steady)}} = 0 \\
\dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\
\sum \dot{m}_i h_i &= \dot{Q}_{out} + \sum \dot{m}_e h_e \\
\dot{Q}_{\text{out}} &= \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) \\
&= \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w \\
&= \dot{m}_a(h_1 - h_2) - \dot{m}_a(\omega_1 - \omega_2)h_w
\end{aligned}$$

Substituting,

$$\begin{aligned}
\dot{Q}_{\text{out}} &= \dot{m}_a\left[(h_1 - h_2) - (\omega_1 - \omega_2)h_w\right] \\
&= (100/60\,\text{kg/s})(71.2 - 27.1)\,\text{kJ/kg} - (0.0160 - 0.00504)(16.8\,\text{kJ/kg}) \\
&= \mathbf{73.2\,kW}
\end{aligned}$$

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14-145 Waste heat from the cooling water is rejected to air in a natural-draft cooling tower. The mass flow rate of the cooling water, the volume flow rate of air, and the mass flow rate of the required makeup water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

Analysis (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation is made up later in the cycle using water at 30°C. Applying the mass balance and the energy balance equations yields

Dry Air Mass Balance:

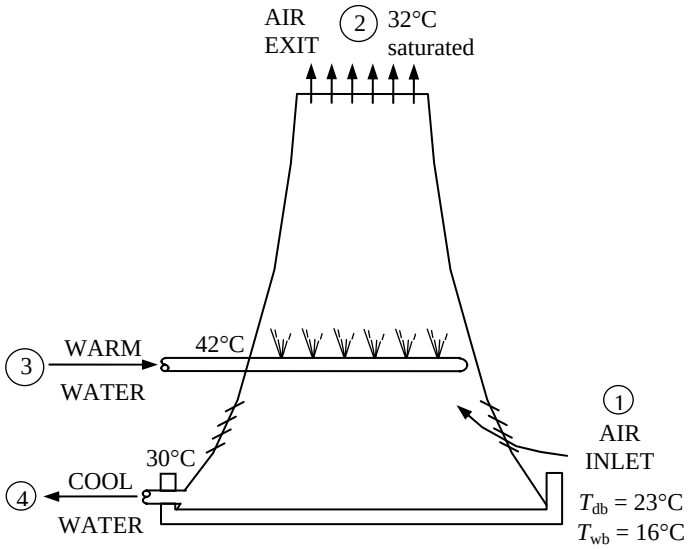
$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \quad \longrightarrow \quad \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e}$$
$$\dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$
$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \nearrow^{0\ (\text{steady})} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$
$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$
$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$
$$0 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 - \dot{m}_3 h_3$$



Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3 (h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1) h_4}$$

From the psychrometric chart (Fig. A-31 or EES),

$$h_1 = 44.67 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.008462 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.8504 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 110.69 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.03065 \text{ kg H}_2\text{O/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,42°\text{C}} = 175.90 \text{ kJ/kg H}_2\text{O}$$
$$h_4 \cong h_{f\,@\,30°\text{C}} = 125.74 \text{ kJ/kg H}_2\text{O}$$

Substituting

$$\dot{m}_a = \frac{\dot{m}_3 (175.90 - 125.74) \text{kJ/kg}}{(110.69 - 44.67) \text{ kJ/kg} - (0.03065 - 0.008462)(125.74) \text{ kJ/kg}} = 0.7933 \dot{m}_3$$

The mass flow rate of the cooling water is determined by applying the steady flow energy balance equation on the cooling water,

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$$\dot{Q}_{\text{waste}} = \dot{m}_3 h_3 - (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 = \dot{m}_3 h_3 - [\dot{m}_3 - \dot{m}_a(\omega_2 - \omega_1)]h_4$$
$$= \dot{m}_3 h_3 - \dot{m}_3[1 - 0.7933(0.03065 - 0.008462)]h_4$$
$$= \dot{m}_3(h_3 - 0.9824h_4)$$

$$70{,}000 \text{ kJ/s} = \dot{m}_3(175.90 - 0.9824 \times 125.74) \text{ kJ/kg} \longrightarrow \dot{m}_3 = \mathbf{1337\ kg/s}$$

and

$$\dot{m}_a = 0.7933\dot{m}_3 = (0.7933)(1337 \text{ kg/s}) = 1061 \text{ kg/s}$$

(*b*) Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a v_1 = (1061 \text{ kg/s})(0.8504 \text{ m}^3/\text{kg}) = \mathbf{902\ m^3/s}$$

(*c*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (1061 \text{ kg/s})(0.03065 - 0.008462) = \mathbf{23.5\ kg/s}$$

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14-146 Problem 14-145 is reconsidered. The effect of air inlet wet-bulb temperature on the required air volume flow rate and the makeup water flow rate is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
P_atm =101.325 [kPa]
T_db_1 = 23 [C]
T_wb_1 = 16 [C]
T_db_2 = 32 [C]
RH_2 = 100/100 "%. relative humidity at state 2, saturated condition"
Q_dot_waste = 70 [MW]*Convert(MW, kW)
T_cw_3 = 42 [C] "Cooling water temperature at state 3"
T_cw_4 = 30 [C] "Cooling water temperature at state 4"

"Dry air mass flow rates:"
"RH_1 is the relative humidity at state 1 on a decimal basis"

v_1=VOLUME(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
T_wb_1 = WETBULB(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
m_dot_a_1 = Vol_dot_1/v_1

"Conservaton of mass for the dry air (ma) in the SSSF mixing device:"
m_dot_a_in  - m_dot_a_out = DELTAm_dot_a_cv
m_dot_a_in = m_dot_a_1
m_dot_a_out =  m_dot_a_2
DELTAm_dot_a_cv = 0 "Steady flow requirement"

"Conservation of mass for the water vapor (mv) and cooling water for the SSSF process:"
m_dot_w_in  - m_dot_w_out = DELTAm_dot_w_cv
m_dot_w_in = m_dot_v_1 + m_dot_cw_3
m_dot_w_out =  m_dot_v_2+m_dot_cw_4
DELTAm_dot_w_cv = 0 "Steady flow requirement"
w_1=HUMRAT(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
m_dot_v_1 = m_dot_a_1*w_1
w_2=HUMRAT(AirH2O,T=T_db_2,P=P_atm,R=RH_2)
m_dot_v_2 = m_dot_a_2*w_2

"Conservation of energy for the SSSF cooling tower  process:"
"The process is adiabatic and has no work done, ngelect ke and pe"
E_dot_in_tower - E_dot_out_tower = DELTAE_dot_tower_cv
E_dot_in_tower= m_dot_a_1 *h[1] +  m_dot_cw_3*h_w[3]
E_dot_out_tower =  m_dot_a_2*h[2] + m_dot_cw_4*h_w[4]
DELTAE_dot_tower_cv = 0 "Steady flow requirement"
h[1]=ENTHALPY(AirH2O,T=T_db_1,P=P_atm,w=w_1)
h[2]=ENTHALPY(AirH2O,T=T_db_2,P=P_atm,w=w_2)
h_w[3]=ENTHALPY(steam,T=T_cw_3,x=0)
h_w[4]=ENTHALPY(steam,T=T_cw_4,x=0)

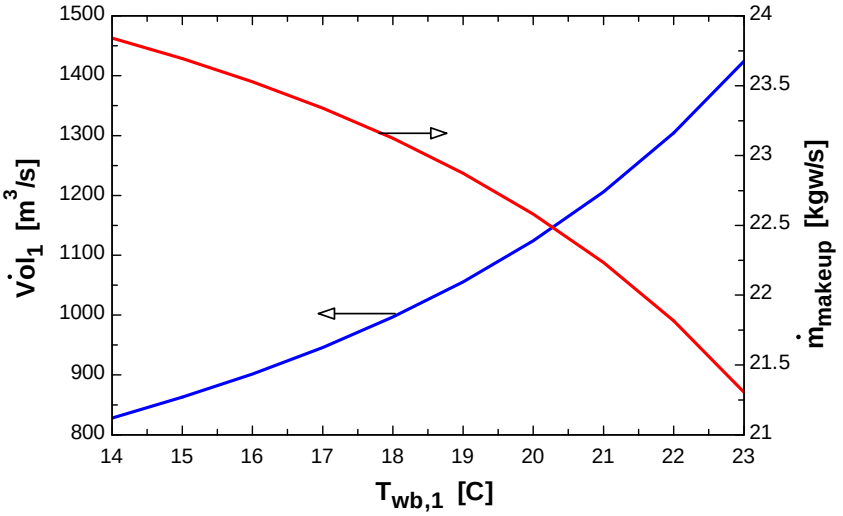

"Energy balance on the external heater determines the cooling water flow rate:"
E_dot_in_heater - E_dot_out_heater = DELTAE_dot_heater_cv
E_dot_in_heater = Q_dot_waste + m_dot_cw_4*h_w[4]
E_dot_out_heater = m_dot_cw_3 * h_w[3]
DELTAE_dot_heater_cv = 0 "Steady flow requirement"

"Conservation of mass on the external heater gives the makeup water flow rate."
"Note: The makeup water flow rate equals the amount of water vaporized in the cooling tower."
```

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m_dot_cw_in - m_dot_cw_out = DELTAm_dot_cw_cv
m_dot_cw_in = m_dot_cw_4 + m_dot_makeup
m_dot_cw_out = m_dot_cw_3
DELTAm_dot_cw_cv = 0 "Steady flow requirement"

T_{wb1} [C]	Vol_1 [m^3/s]	m_{makeup} [kgw/s]	m_{cw3} [kgw/s]	m_{a1} [kga/s]
14	828.2	23.84	1336	977.4
15	862.7	23.69	1336	1016
16	901.7	23.53	1337	1060
17	946	23.34	1337	1110
18	996.7	23.12	1338	1168
19	1055	22.87	1338	1234
20	1124	22.58	1339	1312
21	1206	22.23	1340	1404
22	1304	21.82	1341	1515
23	1424	21.3	1342	1651



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14-147 An uninsulated tank contains moist air at a specified state. Water is sprayed into the tank until the relative humidity in the tank reaches a certain value. The amount of water supplied to the tank, the final pressure in the tank, and the heat transfer during the process are to be determined.

Assumptions **1** Dry air and water vapor are ideal gases. **2** The kinetic and potential energy changes are negligible.

Analysis The initial state of the moist air is completely specified. The properties of the air at the inlet state may be determined from the psychrometric chart (Figure A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 49.16 \text{ kJ/kg dry air}, \qquad \omega_1 = 0.005433 \text{ kg H}_2\text{O/kg dry air}$$
$$\nu_1 = 0.6863 \text{ m}^3 \text{ / kg dry air}$$

The initial mass in the tank is

$$m_a = \frac{V_1}{\nu_1} = \frac{0.5 \text{ m}^3}{0.6863 \text{ m}^3} = 0.7285 \text{ kg}$$

The partial pressure of dry air in the tank is

$$P_{a2} = \frac{m_a R_a T_2}{V} = \frac{(0.7285 \text{ kg})(0.287 \text{ kJ/kg.K})(35 + 273 \text{ K})}{(0.5 \text{ m}^3)} = 128.8 \text{ kPa}$$

Then, the pressure of moist air in the tank is determined from

$$P_2 = P_{a2}\left(1 + \frac{\omega_2}{0.622}\right) = (128.8 \text{ kPa})\left(1 + \frac{\omega_2}{0.622}\right)$$

We cannot fix the final state explicitly by a hand-solution. However, using EES which has built-in functions for moist air properties, the final state properties are determined to be

$$P_2 = \mathbf{133.87 \text{ kPa}} \qquad \omega_2 = 0.02446 \text{ kg H}_2\text{O/kg dry air}$$
$$h_2 = 97.97 \text{ kJ/kg dry air} \qquad \nu_2 = 0.6867 \text{ m}^3 \text{ / kg dry air}$$

The partial pressures at the initial and final states are

$$P_{v1} = \phi_1 P_{\text{sat@35°C}} = 0.20(5.6291 \text{ kPa}) = 1.126 \text{ kPa}$$
$$P_{a1} = P_1 - P_{v1} = 130 - 1.126 = 128.87 \text{ kPa}$$
$$P_{v2} = P_2 - P_{a2} = 133.87 - 128.81 = 5.07 \text{ kPa}$$

The specific volume of water at 35ºC is

$$\nu_{w1} = \nu_{w2} = \nu_{\text{g @35°C}} = 25.205 \text{ m}^3\text{/kg}$$

The internal energies per unit mass of dry air in the tank are

$$u_1 = h_1 - P_{a1}\nu_1 - w_1 P_{v1}\nu_{w1} = 49.16 - 128.87 \times 0.6863 - 0.005433 \times 1.126 \times 25.205 = -39.44 \text{ kJ/kg}$$
$$u_2 = h_2 - P_{a2}\nu_2 - w_2 P_{v2}\nu_{w2} = 97.97 - 128.81 \times 0.6867 - 0.02446 \times 5.07 \times 25.205 = 6.396 \text{ kJ/kg}$$

The enthalpy of water entering the tank from the supply line is

$$h_{w1} = h_{\text{f @50°C}} = 209.34 \text{ kJ/kg}$$

The internal energy of water vapor at the final state is

$$u_{w2} = u_{\text{g @35°C}} = 2422.7 \text{ kJ/kg}$$

The amount of water supplied to the tank is

$$m_w = m_a(\omega_2 - \omega_1) = (0.7285 \text{ kg})(0.02446 - 0.005433) = \mathbf{0.01386 \text{ kg}}$$

An energy balance on the system gives

$$E_{\text{in}} = \Delta E_{\text{tank}}$$
$$Q_{\text{in}} + m_w h_{w1} = m_a(u_2 - u_1) + m_w u_{w2}$$
$$Q_{\text{in}} + (0.01386 \text{ kg})(209.34 \text{ kJ/kg}) = (0.7285 \text{ kg})[6.396 - (-39.44) \text{kJ/kg}] + (0.01386 \text{ kg})(2422.7 \text{ kJ/kg})$$
$$\dot{Q}_{\text{in}} = \mathbf{64.1 \text{ kJ}}$$

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Fundamentals of Engineering (FE) Exam Problems

14-148 A room is filled with saturated moist air at 25°C and a total pressure of 100 kPa. If the mass of dry air in the room is 100 kg, the mass of water vapor is

(a) 0.52 kg (b) 1.97 kg (c) 2.96 kg (d) 2.04 kg (e) 3.17 kg

Answer (d) 2.04 kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P=100 "kPa"
m_air=100 "kg"
RH=1
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v
w=0.622*P_v/(P-P_v)
w=m_v/m_air

"Some Wrong Solutions with Common Mistakes:"
W1_vmass=m_air*w1; w1=0.622*P_v/P "Using P instead of P-Pv in w relation"
W2_vmass=m_air "Taking m_vapor = m_air"
W3_vmass=P_v/P*m_air "Using wrong relation"
```

14-149 A room contains 65 kg of dry air and 0.6 kg of water vapor at 25°C and 90 kPa total pressure. The relative humidity of air in the room is

(a) 3.5% (b) 41.5% (c) 55.2% (d) 60.9% (e) 73.0%

Answer (b) 41.5%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P=90 "kPa"
m_air=65 "kg"
m_v=0.6 "kg"
w=0.622*P_v/(P-P_v)
w=m_v/m_air
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g

"Some Wrong Solutions with Common Mistakes:"
W1_RH=m_v/(m_air+m_v) "Using wrong relation"
W2_RH=P_g/P "Using wrong relation"
```

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14-150 A 40-m^3 room contains air at 30°C and a total pressure of 90 kPa with a relative humidity of 75 percent. The mass of dry air in the room is

(a) 24.7 kg (b) 29.9 kg (c) 39.9 kg (d) 41.4 kg (e) 52.3 kg

Answer (c) 39.9 kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=40 "m^3"
T1=30 "C"
P=90 "kPa"
RH=0.75
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v
R_air=0.287 "kJ/kg.K"
m_air=P_air*V/(R_air*(T1+273))

"Some Wrong Solutions with Common Mistakes:"
W1_mass=P_air*V/(R_air*T1) "Using C instead of K"
W2_mass=P*V/(R_air*(T1+273)) "Using P instead of P_air"
W3_mass=m_air*RH "Using wrong relation"
```

14-151 A room contains air at 30°C and a total pressure of 96.0 kPa with a relative humidity of 75 percent. The partial pressure of dry air is

(a) 82.0 kPa (b) 85.8 kPa (c) 92.8 kPa (d) 90.6 kPa (e) 72.0 kPa

Answer (c) 92.8 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=30 "C"
P=96 "kPa"
RH=0.75
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v

"Some Wrong Solutions with Common Mistakes:"
W1_Pair=P_v "Using Pv as P_air"
W2_Pair=P-P_g "Using wrong relation"
W3_Pair=RH*P "Using wrong relation"
```

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14-152 The air in a house is at 25°C and 65 percent relative humidity. Now the air is cooled at constant pressure. The temperature at which the moisture in the air will start condensing is

(a) 7.4°C (b) 16.3°C (c) 18.0°C (d) 11.3°C (e) 20.2°C

Answer (c) 18.0°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
RH1=0.65
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH1=P_v/P_g
T_dp=TEMPERATURE(Steam_IAPWS,x=0,P=P_v)

"Some Wrong Solutions with Common Mistakes:"
W1_Tdp=T1*RH1 "Using wrong relation"
W2_Tdp=(T1+273)*RH1-273 "Using wrong relation"
W3_Tdp=WETBULB(AirH2O,T=T1,P=P1,R=RH1); P1=100 "Using wet-bulb temperature"
```

14-153 On the psychrometric chart, a cooling and dehumidification process appears as a line that is

(a) horizontal to the left,
(b) vertical downward,
(c) diagonal upwards to the right (NE direction)
(d) diagonal upwards to the left (NW direction)
(e) diagonal downwards to the left (SW direction)

Answer (e) diagonal downwards to the left (SW direction)

14-154 On the psychrometric chart, a heating and humidification process appears as a line that is

(a) horizontal to the right,
(b) vertical upward,
(c) diagonal upwards to the right (NE direction)
(d) diagonal upwards to the left (NW direction)
(e) diagonal downwards to the right (SE direction)

Answer (c) diagonal upwards to the right (NE direction)

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14-155 An air stream at a specified temperature and relative humidity undergoes evaporative cooling by spraying water into it at about the same temperature. The lowest temperature the air stream can be cooled to is

(a) the dry bulb temperature at the given state
(b) the wet bulb temperature at the given state
(c) the dew point temperature at the given state
(d) the saturation temperature corresponding to the humidity ratio at the given state
(e) the triple point temperature of water

Answer (a) the dry bulb temperature at the given state

14-156 Air is cooled and dehumidified as it flows over the coils of a refrigeration system at 85 kPa from 35°C and a humidity ratio of 0.023 kg/kg dry air to 15°C and a humidity ratio of 0.015 kg/kg dry air. If the mass flow rate of dry air is 0.4 kg/s, the rate of heat removal from the air is

(a) 4 kJ/s (b) 8 kJ/s (c) 12 kJ/s (d) 16 kJ/s (e) 20 kJ/s

Answer (d) 16 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=85 "kPa"
T1=35 "C"
w1=0.023
T2=15 "C"
w2=0.015
m_air=0.4 "kg/s"
m_water=m_air*(w1-w2)
h1=ENTHALPY(AirH2O,T=T1,P=P,w=w1)
h2=ENTHALPY(AirH2O,T=T2,P=P,w=w2)
h_w=ENTHALPY(Steam_IAPWS,T=T2,x=0)
Q=m_air*(h1-h2)-m_water*h_w

"Some Wrong Solutions with Common Mistakes:"
W1_Q=m_air*(h1-h2) "Ignoring condensed water"
W2_Q=m_air*Cp_air*(T1-T2)-m_water*h_w; Cp_air = 1.005 "Using dry air enthalpies"
W3_Q=m_air*(h1-h2)+m_water*h_w "Using wrong sign"
```

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14-157 Air at a total pressure of 90 kPa, 15°C, and 75 percent relative humidity is heated and humidified to 25°C and 75 percent relative humidity by introducing water vapor. If the mass flow rate of dry air is 4 kg/s, the rate at which steam is added to the air is

(a) 0.032 kg/s (b) 0.013 kg/s (c) 0.019 kg/s (d) 0.0079 kg/s (e) 0 kg/s

Answer (a) 0.032 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=90 "kPa"
T1=15 "C"
RH1=0.75
T2=25 "C"
RH2=0.75
m_air=4 "kg/s"
w1=HUMRAT(AirH2O,T=T1,P=P,R=RH1)
w2=HUMRAT(AirH2O,T=T2,P=P,R=RH2)
m_water=m_air*(w2-w1)

"Some Wrong Solutions with Common Mistakes:"
W1_mv=0 "sine RH = constant"
W2_mv=w2-w1 "Ignoring mass flow rate of air"
W3_mv=RH1*m_air "Using wrong relation"
```

14-158 ··· 14-162 Design and Essay Problems



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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 15
CHEMICAL REACTIONS

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Fuels and Combustion

15-1C Nitrogen, in general, does not react with other chemical species during a combustion process but its presence affects the outcome of the process because nitrogen absorbs a large proportion of the heat released during the chemical process.

15-2C Moisture, in general, does not react chemically with any of the species present in the combustion chamber, but it absorbs some of the energy released during combustion, and it raises the dew point temperature of the combustion gases.

15-3C The number of atoms are preserved during a chemical reaction, but the total mole numbers are not.

15-4C Air-fuel ratio is the ratio of the mass of air to the mass of fuel during a combustion process. Fuel-air ratio is the inverse of the air-fuel ratio.

15-5C No. Because the molar mass of the fuel and the molar mass of the air, in general, are different.

15-6C The dew-point temperature of the product gases is the temperature at which the water vapor in the product gases starts to condense as the gases are cooled at constant pressure. It is the saturation temperature corresponding to the vapor pressure of the product gases.

15-7 Sulfur is burned with oxygen to form sulfur dioxide. The minimum mass of oxygen required and the mass of sulfur dioxide in the products are to be determined when 1 kg of sulfur is burned.

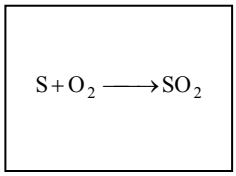
Properties The molar masses of sulfur and oxygen are 32.06 kg/kmol and 32.00 kg/kmol, respectively (Table A-1).

Analysis The chemical reaction is given by

$$S + O_2 \longrightarrow SO_2$$

Hence, 1kmol of oxygen is required to burn 1 kmol of sulfur which produces 1 kmol of sulfur dioxide whose molecular weight is

$$M_{SO2} = M_S + M_{O2} = 32.06 + 32.00 = 64.06 \text{ kg/kmol}$$



Then,

$$\frac{m_{O2}}{m_S} = \frac{N_{O2} M_{O2}}{N_S M_S} = \frac{(1 \text{ kmol})(32 \text{ kg/kmol})}{(1 \text{ kmol})(32.06 \text{ kg/kmol})} = \mathbf{0.998 \text{ kg } O_2/\text{kg S}}$$

and

$$\frac{m_{SO2}}{m_S} = \frac{N_{SO2} M_{SO2}}{N_S M_S} = \frac{(1 \text{ kmol})(64.06 \text{ kg/kmol})}{(1 \text{ kmol})(32.06 \text{ kg/kmol})} = \mathbf{1.998 \text{ kg } SO_2/\text{kg S}}$$

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15-8E Methane is burned with diatomic oxygen. The mass of water vapor in the products is to be determined when 1 lbm of methane is burned.

Properties The molar masses of CH_4, O_2, CO_2, and H_2O are 16, 32, 44, and 18 lbm/lbmol, respectively (Table A-1E).

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

Analysis The chemical reaction is given by

$$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$$

Hence, for each lbmol of methane burned, 2 lbmol of water vapor are formed. Then,

$$\frac{m_{H2O}}{m_{CH4}} = \frac{N_{H2O} M_{H2O}}{N_{CH4} M_{CH4}} = \frac{(2\ \text{lbmol})(18\ \text{lbm/lbmol})}{(1\ \text{lbmol})(16\ \text{lbm/lbmol})} = \mathbf{2.25\ lbm\ H_2O/lbm\ CH_4}$$

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Theoretical and Actual Combustion Processes

15-9C It represent the amount of air that contains the exact amount of oxygen needed for complete combustion.

15-10C No. The theoretical combustion is also complete, but the products of theoretical combustion does not contain any uncombined oxygen.

15-11C Case (b).

15-12C The causes of incomplete combustion are insufficient time, insufficient oxygen, insufficient mixing, and dissociation.

15-13C CO. Because oxygen is more strongly attracted to hydrogen than it is to carbon, and hydrogen is usually burned to completion even when there is a deficiency of oxygen.

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15-14 Propane is burned with theoretical amount of air. The mass fraction of carbon dioxide and the mole and mass fractions of the water vapor in the products are to be determined.

Properties The molar masses of C_3H_8, O_2, N_2, CO_2, and H_2O are 44, 32, 28, 44, and 18 kg/kmol, respectively (Table A-1).

Analysis (*a*) The reaction in terms of undetermined coefficients is

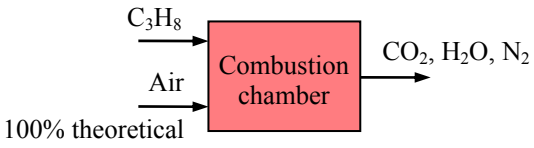
$$C_3H_8 + x(O_2 + 3.76N_2) \longrightarrow yCO_2 + zH_2O + pN_2$$

Balancing the carbon in this reaction gives

$$y = 3$$

and the hydrogen balance gives

$$2z = 8 \longrightarrow z = 4$$



The oxygen balance produces

$$2x = 2y + z \longrightarrow x = y + z/2 = 3 + 4/2 = 5$$

A balance of the nitrogen in this reaction gives

$$2 \times 3.76x = 2p \longrightarrow p = 3.76x = 3.76 \times 5 = 18.8$$

In balanced form, the reaction is

$$C_3H_8 + 5O_2 + 18.8N_2 \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$

The mass fraction of carbon dioxide is determined from

$$\text{mf}_{CO2} = \frac{m_{CO2}}{m_{\text{products}}} = \frac{N_{CO2}M_{CO2}}{N_{CO2}M_{CO2} + N_{H2O}M_{H2O} + N_{N2}M_{N2}}$$
$$= \frac{(3\text{ kmol})(44\text{ kg/kmol})}{(3\text{ kmol})(44\text{ kg/kmol}) + (4\text{ kmol})(18\text{ kg/kmol}) + (18.8\text{ kmol})(28\text{ kg/kmol})}$$
$$= \frac{132\text{ kg}}{730.4\text{ kg}} = \mathbf{0.181}$$

(*b*) The mole and mass fractions of water vapor are

$$y_{H2O} = \frac{N_{H2O}}{N_{\text{products}}} = \frac{N_{H2O}}{N_{CO2} + N_{H2O} + N_{N2}} = \frac{4\text{ kmol}}{3\text{ kmol} + 4\text{ kmol} + 18.8\text{ kmol}} = \frac{4\text{ kmol}}{25.8\text{ kmol}} = \mathbf{0.155}$$

$$\text{mf}_{H2O} = \frac{m_{H2O}}{m_{\text{products}}} = \frac{N_{H2O}M_{H2O}}{N_{CO2}M_{CO2} + N_{H2O}M_{H2O} + N_{N2}M_{N2}}$$
$$= \frac{(4\text{ kmol})(18\text{ kg/kmol})}{(3\text{ kmol})(44\text{ kg/kmol}) + (4\text{ kmol})(18\text{ kg/kmol}) + (18.8\text{ kmol})(28\text{ kg/kmol})}$$
$$= \frac{72\text{ kg}}{730.4\text{ kg}} = \mathbf{0.0986}$$

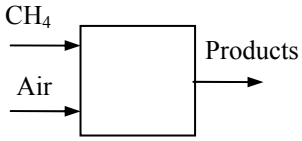
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15-15 Methane is burned with air. The mass flow rates at the two inlets are to be determined.

Properties The molar masses of CH_4, O_2, N_2, CO_2, and H_2O are 16, 32, 28, 44, and 18 kg/kmol, respectively (Table A-1).

Analysis The stoichiometric combustion equation of CH_4 is

$$CH_4 + a_{th}[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$



O_2 balance: $a_{th} = 1 + 1 \longrightarrow a_{th} = 2$

Substituting, $CH_4 + 2[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O + 7.52N_2$

The masses of the reactants are

$$m_{CH4} = N_{CH4}M_{CH4} = (1\,\text{kmol})(16\,\text{kg/kmol}) = 16\,\text{kg}$$
$$m_{O2} = N_{O2}M_{O2} = (2\,\text{kmol})(32\,\text{kg/kmol}) = 64\,\text{kg}$$
$$m_{N2} = N_{N2}M_{N2} = (2 \times 3.76\,\text{kmol})(28\,\text{kg/kmol}) = 211\,\text{kg}$$

The total mass is

$$m_{total} = m_{CH4} + m_{O2} + N_{N2} = 16 + 64 + 211 = 291\,\text{kg}$$

Then the mass fractions are

$$\text{mf}_{CH4} = \frac{m_{CH4}}{m_{total}} = \frac{16\,\text{kg}}{291\,\text{kg}} = 0.05498$$
$$\text{mf}_{O2} = \frac{m_{O2}}{m_{total}} = \frac{64\,\text{kg}}{291\,\text{kg}} = 0.2199$$
$$\text{mf}_{N2} = \frac{m_{N2}}{m_{total}} = \frac{211\,\text{kg}}{291\,\text{kg}} = 0.7251$$

For a mixture flow of 0.5 kg/s, the mass flow rates of the reactants are

$$\dot{m}_{CH4} = \text{mf}_{CH4}\dot{m} = (0.05498)(0.5\,\text{kg/s}) = \mathbf{0.02749\ kg/s}$$
$$\dot{m}_{air} = \dot{m} - \dot{m}_{CH4} = 0.5 - 0.02749 = \mathbf{0.4725\ kg/s}$$

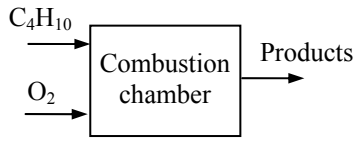
15-16 n-Butane is burned with stoichiometric amount of oxygen. The mole fractions of CO_2 water in the products and the mole number of CO_2 in the products per mole of fuel burned are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2 and H_2O. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, and O_2 are 12 kg/kmol, 2 kg/kmol, and 32 kg/kmol, respectively (Table A-1).

Analysis The combustion equation in this case is

$$C_4H_{10} + 6.5O_2 \longrightarrow 4CO_2 + 5H_2O$$



The total mole of the products are 4+5 = 9 kmol. Then the mole fractions are

$$y_{CO2} = \frac{N_{CO2}}{N_{total}} = \frac{4\,\text{kmol}}{9\,\text{kmol}} = \mathbf{0.4444}$$
$$y_{CO2} = \frac{N_{H2O}}{N_{total}} = \frac{5\,\text{kmol}}{9\,\text{kmol}} = \mathbf{0.5556}$$

Also, $N_{CO2} = \mathbf{4\ kmol\ CO_2/kmol\ C_4H_{10}}$

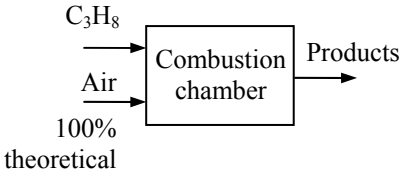
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15-17 Propane is burned with stoichiometric amount of air. The mass fraction of each product, the mass of water and air per unit mass of fuel burned are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction equation for 100% theoretical air is



$$C_3H_8 + a_{th}[O_2 + 3.76N_2] \longrightarrow B\,CO_2 + D\,H_2O + E\,N_2$$

where a_{th} is the stoichiometric coefficient for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

Carbon balance: $B = 3$

Hydrogen balance: $2D = 8 \longrightarrow D = 4$

Oxygen balance: $2a_{th} = 2B + D \longrightarrow a_{th} = 0.5(2\times 3 + 4) = 5$

Nitrogen balance: $a_{th} \times 3.76 = E \longrightarrow E = 5 \times 3.76 = 18.8$

Substituting, the balanced reaction equation is

$$C_3H_8 + 5[O_2 + 3.76N_2] \longrightarrow 3\,CO_2 + 4\,H_2O + 18.8\,N_2$$

The mass of each product and the total mass are

$$m_{CO2} = N_{CO2}M_{CO2} = (3\text{ kmol})(44\text{ kg/kmol}) = 132\text{ kg}$$
$$m_{H2O} = N_{H2O}M_{H2O} = (4\text{ kmol})(18\text{ kg/kmol}) = 72\text{ kg}$$
$$m_{N2} = N_{N2}M_{N2} = (18.8\text{ kmol})(28\text{ kg/kmol}) = 526.4\text{ kg}$$
$$m_{total} = m_{CO2} + m_{H2O} + m_{N2} = 132 + 72 + 526.4 = 730.4\text{ kg}$$

Then the mass fractions are

$$\text{mf}_{CO2} = \frac{m_{CO2}}{m_{total}} = \frac{132\text{ kg}}{730.4\text{ kg}} = \mathbf{0.1807}$$

$$\text{mf}_{H2O} = \frac{m_{H2O}}{m_{total}} = \frac{72\text{ kg}}{730.4\text{ kg}} = \mathbf{0.0986}$$

$$\text{mf}_{N2} = \frac{m_{N2}}{m_{total}} = \frac{526.4\text{ kg}}{730.4\text{ kg}} = \mathbf{0.7207}$$

The mass of water per unit mass of fuel burned is

$$\frac{m_{H2O}}{m_{C3H8}} = \frac{(4\times 18)\text{ kg}}{(1\times 44)\text{ kg}} = \mathbf{1.636\ kg\ H_2O/kg\ C_3H_8}$$

The mass of air required per unit mass of fuel burned is

$$\frac{m_{air}}{m_{C3H8}} = \frac{(5\times 4.76\times 29)\text{ kg}}{(1\times 44)\text{ kg}} = \mathbf{15.69\ kg\ air/kg\ C_3H_8}$$

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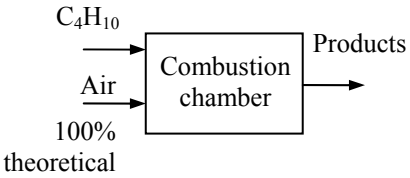
15-18 n-Octane is burned with stoichiometric amount of air. The mass fraction of each product, the mass of water in the products and the mass fraction of each reactant are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction equation for 100% theoretical air is

$$C_8H_{18} + a_{th}[O_2 + 3.76N_2] \longrightarrow B\,CO_2 + D\,H_2O + E\,N_2$$



where a_{th} is the stoichiometric coefficient for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

Carbon balance: $B = 8$

Hydrogen balance: $2D = 18 \longrightarrow D = 9$

Oxygen balance: $2a_{th} = 2B + D \longrightarrow a_{th} = 0.5(2\times 8 + 9) = 12.5$

Nitrogen balance: $a_{th} \times 3.76 = E \longrightarrow E = 12.5 \times 3.76 = 47$

Substituting, the balanced reaction equation is

$$C_8H_{18} + 12.5[O_2 + 3.76N_2] \longrightarrow 8\,CO_2 + 9\,H_2O + 47\,N_2$$

The mass of each product and the total mass are

$$m_{CO2} = N_{CO2}M_{CO2} = (8\text{ kmol})(44\text{ kg/kmol}) = 352\text{ kg}$$
$$m_{H2O} = N_{H2O}M_{H2O} = (9\text{ kmol})(18\text{ kg/kmol}) = 162\text{ kg}$$
$$m_{N2} = N_{N2}M_{N2} = (47\text{ kmol})(28\text{ kg/kmol}) = 1316\text{ kg}$$
$$m_{total} = m_{CO2} + m_{N2} + m_{H2O} = 352 + 162 + 1316 = 1830\text{ kg}$$

Then the mass fractions are

$$mf_{CO2} = \frac{m_{CO2}}{m_{total}} = \frac{352\text{ kg}}{1830\text{ kg}} = \mathbf{0.1923}$$

$$mf_{H2O} = \frac{m_{H2O}}{m_{total}} = \frac{162\text{ kg}}{1830\text{ kg}} = \mathbf{0.0885}$$

$$mf_{N2} = \frac{m_{N2}}{m_{total}} = \frac{1316\text{ kg}}{1830\text{ kg}} = \mathbf{0.7191}$$

The mass of water per unit mass of fuel burned is

$$\frac{m_{H2O}}{m_{C8H18}} = \frac{(9\times 18)\text{ kg}}{(1\times 114)\text{ kg}} = \mathbf{1.421\,kg\,H_2O/kg\,C_8H_{18}}$$

The mass of each reactant and the total mass are

$$m_{C8H18} = N_{C8H18}M_{C8H18} = (1\text{ kmol})(114\text{ kg/kmol}) = 114\text{ kg}$$
$$m_{air} = N_{air}M_{air} = (12.5\times 4.76\text{ kmol})(29\text{ kg/kmol}) = 1725.5\text{ kg}$$
$$m_{total} = m_{C8H18} + m_{air} = 114 + 1725.5 = 1839.5\text{ kg}$$

Then the mass fractions of reactants are

$$mf_{C8H18} = \frac{m_{C8H18}}{m_{total}} = \frac{114\text{ kg}}{1839.5\text{ kg}} = \mathbf{0.0620}$$

$$mf_{air} = \frac{m_{air}}{m_{total}} = \frac{1725.5\text{ kg}}{1839.5\text{ kg}} = \mathbf{0.9380}$$

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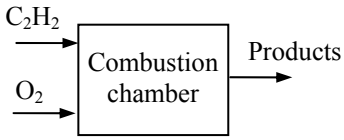
15-19 Acetylene is burned with 10 percent excess oxygen. The mass fractions of each of the products and the mass of oxygen used per unit mass of fuel burned are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and O_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, and O_2 are 12 kg/kmol, 2 kg/kmol, and 32 kg/kmol, respectively (Table A-1).

Analysis The stoichiometric combustion equation is

$$C_2H_2 + 2.5O_2 \longrightarrow 2CO_2 + H_2O$$



The combustion equation with 10% excess oxygen is

$$C_2H_2 + 2.75O_2 \longrightarrow 2CO_2 + H_2O + 0.25O_2$$

The mass of each product and the total mass are

$$m_{CO2} = N_{CO2}M_{CO2} = (2\text{ kmol})(44\text{ kg/kmol}) = 88\text{ kg}$$
$$m_{H2O} = N_{H2O}M_{H2O} = (1\text{ kmol})(18\text{ kg/kmol}) = 18\text{ kg}$$
$$m_{O2} = N_{O2}M_{O2} = (0.25\text{ kmol})(32\text{ kg/kmol}) = 8\text{ kg}$$
$$m_{total} = m_{CO2} + m_{H2O} + m_{O2} = 88 + 18 + 8 = 114\text{ kg}$$

Then the mass fractions are

$$\text{mf}_{CO2} = \frac{m_{CO2}}{m_{total}} = \frac{88\text{ kg}}{114\text{ kg}} = \mathbf{0.7719}$$

$$\text{mf}_{H2O} = \frac{m_{H2O}}{m_{total}} = \frac{18\text{ kg}}{114\text{ kg}} = \mathbf{0.1579}$$

$$\text{mf}_{O2} = \frac{m_{O2}}{m_{total}} = \frac{8\text{ kg}}{114\text{ kg}} = \mathbf{0.0702}$$

The mass of oxygen per unit mass of fuel burned is determined from

$$\frac{m_{O2}}{m_{C2H2}} = \frac{(2.75\times 32)\text{ kg}}{(1\times 26)\text{ kg}} = \mathbf{3.385\text{ kg }O_2\text{/kg }C_2H_2}$$

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15-20 n-Butane is burned with 100 percent excess air. The mole fractions of each of the products, the mass of carbon dioxide in the products per unit mass of the fuel, and the air-fuel ratio are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2, and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion equation in this case can be written as

$$C_4H_{10} + 2.0a_{th}[O_2 + 3.76N_2] \longrightarrow 4CO_2 + 5H_2O + 1.0a_{th}O_2 + (2.0 \times 3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient for air. We have automatically accounted for the 100% excess air by using the factor $2.0a_{th}$ instead of a_{th} for air. The stoichiometric amount of oxygen ($a_{th}O_2$) will be used to oxidize the fuel, and the remaining excess amount ($1.0a_{th}O_2$) will appear in the products as free oxygen. The coefficient a_{th} is determined from the O_2 balance,

O_2 balance: $\quad 2.0a_{th} = 4 + 2.5 + 1.0a_{th} \longrightarrow a_{th} = 6.5$

Substituting, $\quad C_4H_{10} + 13[O_2 + 3.76N_2] \longrightarrow 4CO_2 + 5H_2O + 6.5O_2 + 48.88N_2$

The mole fractions of the products are

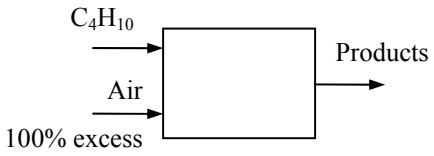
$$N_m = 4 + 5 + 6.5 + 48.88 = 64.38 \text{ kmol}$$

$$y_{CO2} = \frac{N_{CO2}}{N_m} = \frac{4 \text{ kmol}}{64.38 \text{ kmol}} = \mathbf{0.0621}$$

$$y_{H2O} = \frac{N_{H2O}}{N_m} = \frac{5 \text{ kmol}}{64.38 \text{ kmol}} = \mathbf{0.0777}$$

$$y_{O2} = \frac{N_{O2}}{N_m} = \frac{6.5 \text{ kmol}}{64.38 \text{ kmol}} = \mathbf{0.1010}$$

$$y_{N2} = \frac{N_{N2}}{N_m} = \frac{48.88 \text{ kmol}}{64.38 \text{ kmol}} = \mathbf{0.7592}$$



The mass of carbon dioxide in the products per unit mass of fuel burned is

$$\frac{m_{CO2}}{m_{C4H10}} = \frac{(4 \times 44) \text{ kg}}{(1 \times 58) \text{ kg}} = \mathbf{3.034 \text{ kg } CO_2/\text{kg } C_4H_{10}}$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$AF = \frac{m_{air}}{m_{fuel}} = \frac{(13 \times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{(1 \text{ kmol})(58 \text{ kg/kmol})} = \mathbf{30.94 \text{ kg air/kg fuel}}$$

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15-21 n-Octane is burned with 50 percent excess air. The mole fractions of each of the products, the mass of water in the products per unit mass of the fuel, and the mass fraction of each reactant are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2, and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion equation in this case can be written as

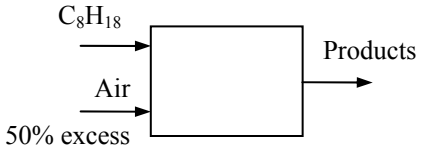
$$C_8H_{18} + 1.5a_{th}[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + 0.5a_{th}O_2 + (1.5 \times 3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient for air. We have automatically accounted for the 50% excess air by using the factor $1.5a_{th}$ instead of a_{th} for air. The stoichiometric amount of oxygen ($a_{th}O_2$) will be used to oxidize the fuel, and the remaining excess amount ($0.5a_{th}O_2$) will appear in the products as free oxygen. The coefficient a_{th} is determined from the O_2 balance,

O_2 balance: $\quad 1.5a_{th} = 8 + 4.5 + 0.5a_{th} \longrightarrow a_{th} = 12.5$

Substituting, $\quad C_8H_{18} + 18.75[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + 6.25O_2 + 70.5N_2$

The mass of each product and the total mass are



$$m_{CO2} = N_{CO2}M_{CO2} = (8\text{ kmol})(44\text{ kg/kmol}) = 352\text{ kg}$$
$$m_{H2O} = N_{H2O}M_{H2O} = (9\text{ kmol})(18\text{ kg/kmol}) = 162\text{ kg}$$
$$m_{O2} = N_{O2}M_{O2} = (6.25\text{ kmol})(32\text{ kg/kmol}) = 200\text{ kg}$$
$$m_{N2} = N_{N2}M_{N2} = (70.5\text{ kmol})(28\text{ kg/kmol}) = 1974\text{ kg}$$
$$m_{total} = m_{CO2} + m_{H2O} + m_{O2} + m_{N2} = 352 + 162 + 200 + 1974 = 2688\text{ kg}$$

Then the mass fractions are

$$\text{mf}_{CO2} = \frac{m_{CO2}}{m_{total}} = \frac{352\text{ kg}}{2688\text{ kg}} = \mathbf{0.1310}$$
$$\text{mf}_{H2O} = \frac{m_{H2O}}{m_{total}} = \frac{162\text{ kg}}{2688\text{ kg}} = \mathbf{0.0603}$$
$$\text{mf}_{O2} = \frac{m_{O2}}{m_{total}} = \frac{200\text{ kg}}{2688\text{ kg}} = \mathbf{0.0744}$$
$$\text{mf}_{N2} = \frac{m_{N2}}{m_{total}} = \frac{1974\text{ kg}}{2688\text{ kg}} = \mathbf{0.7344}$$

The mass of water per unit mass of fuel burned is

$$\frac{m_{H2O}}{m_{C8H18}} = \frac{(9 \times 18)\text{ kg}}{(1 \times 114)\text{ kg}} = \mathbf{1.421\,kg\,H_2O/kg\,C_8H_{18}}$$

The mass of each reactant and the total mass are

$$m_{C8H18} = N_{C8H18}M_{C8H18} = (1\text{ kmol})(114\text{ kg/kmol}) = 114\text{ kg}$$
$$m_{air} = N_{air}M_{air} = (17.75 \times 4.76\text{ kmol})(29\text{ kg/kmol}) = 2588\text{ kg}$$
$$m_{total} = m_{C8H18} + m_{air} = 114 + 2588 = 2702\text{ kg}$$

Then the mass fractions of reactants are

$$\text{mf}_{C8H18} = \frac{m_{C8H18}}{m_{total}} = \frac{114\text{ kg}}{2702\text{ kg}} = \mathbf{0.0422}$$
$$\text{mf}_{air} = \frac{m_{air}}{m_{total}} = \frac{2588\text{ kg}}{2702\text{ kg}} = \mathbf{0.9578}$$

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15-22 Ethyl alcohol is burned with 70% excess air. The mole fractions of the products and the reactants, the mass of water and oxygen in products per unit mass of fuel are to be determined.

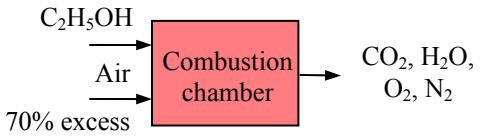
Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction with stoichiometric air is

$$C_2H_5OH + a_{th}[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3H_2O + a_{th} \times 3.76\,N_2$$

where $0.5 + a_{th} = 2 + 1.5 \longrightarrow a_{th} = 3$



Substituting,

$$C_2H_5OH + 3[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3H_2O + 3 \times 3.76\,N_2$$

The reaction with 70% excess air can be written as

$$C_2H_5OH + 1.7 \times 3[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3\,H_2O + x\,O_2 + 1.7 \times 3 \times 3.76\,N_2$$

The coefficient x is determined from O_2 balance:

$$0.5 + 1.7 \times 3 = 2 + 1.5 + x \longrightarrow x = 2.1$$

Then, $C_2H_5OH + 5.1[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3\,H_2O + 2.1\,O_2 + 19.18\,N_2$

The total moles of the products is

$$N_m = 2 + 3 + 2.1 + 19.18 = 26.28\ \text{kmol}$$

The mole fractions of the products are

$$y_{CO2} = \frac{N_{CO2}}{N_m} = \frac{2\ \text{kmol}}{26.28\ \text{kmol}} = \mathbf{0.0761}$$

$$y_{H2O} = \frac{N_{H2O}}{N_m} = \frac{3\ \text{kmol}}{26.28\ \text{kmol}} = \mathbf{0.1142}$$

$$y_{O2} = \frac{N_{O2}}{N_m} = \frac{2.1\ \text{kmol}}{26.28\ \text{kmol}} = \mathbf{0.0799}$$

$$y_{N2} = \frac{N_{N2}}{N_m} = \frac{19.18\ \text{kmol}}{26.28\ \text{kmol}} = \mathbf{0.7298}$$

The total moles of the reactants is

$$N_m = 1 + 5.1 \times 4.76 = 25.28\ \text{kmol}$$

The mole fractions of the reactants are

$$y_{C2H5OH} = \frac{N_{C2H5OH}}{N_m} = \frac{1\ \text{kmol}}{25.28\ \text{kmol}} = \mathbf{0.0396}$$

$$y_{air} = \frac{N_{air}}{N_m} = \frac{(5.1 \times 4.76)\ \text{kmol}}{25.28\ \text{kmol}} = \mathbf{0.9603}$$

The mass of water and oxygen in the products per unit mass of fuel burned is

$$\frac{m_{H2O}}{m_{C2H5OH}} = \frac{(3 \times 18)\ \text{kg}}{(1 \times 46)\ \text{kg}} = \mathbf{1.174\ kg\ H_2O/kg\ C_2H_5OH}$$

$$\frac{m_{O2}}{m_{C2H5OH}} = \frac{(2.1 \times 32)\ \text{kg}}{(1 \times 46)\ \text{kg}} = \mathbf{1.461\ kg\ O_2/kg\ C_2H_5OH}$$

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15-23 Ethyl alcohol is burned with 70% excess air. The air-fuel ratio is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

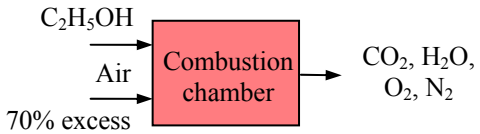
Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction with stoichiometric air is

$$C_2H_5OH + a_{th}[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3H_2O + a_{th} \times 3.76\,N_2$$

where

$$0.5 + a_{th} = 2 + 1.5 \longrightarrow a_{th} = 3$$



Substituting,

$$C_2H_5OH + 3[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3H_2O + 3 \times 3.76\,N_2$$

The reaction with 70% excess air can be written as

$$C_2H_5OH + 1.7 \times 3[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3\,H_2O + x\,O_2 + 1.7 \times 3 \times 3.76\,N_2$$

The coefficient x is determined from O_2 balance:

$$0.5 + 1.7 \times 3 = 2 + 1.5 + x \longrightarrow x = 2.1$$

Then,

$$C_2H_5OH + 5.1[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3\,H_2O + 2.1\,O_2 + 19.18\,N_2$$

The air-fuel mass ratio is

$$AF = \frac{m_{air}}{m_{fuel}} = \frac{(5.1 \times 4.76 \times 29)\text{ kg}}{(1 \times 46)\text{ kg}} = \frac{704.0\text{ kg}}{46\text{ kg}} = \mathbf{15.30\ kg\ air/kg\ fuel}$$

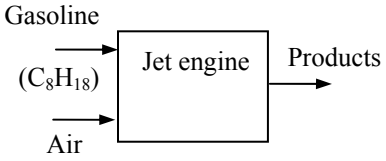
15-24 Gasoline is burned steadily with air in a jet engine. The AF ratio is given. The percentage of excess air used is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2 only.

Properties The molar masses of C, H_2, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The theoretical combustion equation in this case can be written as

$$C_8H_{18} + a_{th}[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + 3.76a_{th}\,N_2$$



where a_{th} is the stoichiometric coefficient for air. It is determined from

O_2 balance: $a_{th} = 8 + 4.5 \longrightarrow a_{th} = 12.5$

The air-fuel ratio for the theoretical reaction is determined by taking the ratio of the mass of the air to the mass of the fuel for,

$$AF_{th} = \frac{m_{air,th}}{m_{fuel}} = \frac{(12.5 \times 4.76\text{ kmol})(29\text{ kg/kmol})}{(8\text{ kmol})(12\text{ kg/kmol}) + (9\text{ kmol})(2\text{ kg/kmol})} = 15.14\text{ kg air/kg fuel}$$

Then the percent theoretical air used can be determined from

$$\text{Percent theoretical air} = \frac{AF_{act}}{AF_{th}} = \frac{18\text{ kg air/kg fuel}}{15.14\text{ kg air/kg fuel}} = \mathbf{119\%}$$

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15-25E Ethylene is burned with 175 percent theoretical air during a combustion process. The AF ratio and the dew-point temperature of the products are to be determined.

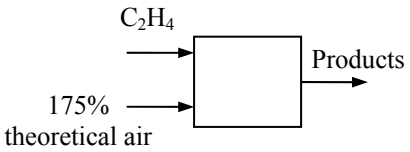
Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, and air are 12 lbm/lbmol, 2 lbm/lbmol, and 29 lbm/lbmol, respectively (Table A-1E).

Analysis (*a*) The combustion equation in this case can be written as

$$C_2H_4 + 1.75a_{th}[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 2H_2O + 0.75a_{th}O_2 + (1.75\times 3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient for air. It is determined from



O_2 balance: $1.75a_{th} = 2 + 1 + 0.75a_{th} \longrightarrow a_{th} = 3$

Substituting,

$$C_2H_4 + 5.25[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 2H_2O + 2.25O_2 + 19.74N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$AF = \frac{m_{air}}{m_{fuel}} = \frac{(5.25\times 4.76 \text{ lbmol})(29 \text{ lbm/lbmol})}{(2 \text{ lbmol})(12 \text{ lbm/lbmol}) + (2 \text{ lbmol})(2 \text{ lbm/lbmol})} = \mathbf{25.9 \text{ lbm air/lbm fuel}}$$

(*b*) The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{prod}}\right)P_{prod} = \left(\frac{2 \text{ lbmol}}{25.99 \text{ lbmol}}\right)(14.5 \text{ psia}) = 1.116 \text{ psia}$$

Thus,

$$T_{dp} = T_{sat@1.116 \text{ psia}} = \mathbf{105.4°F}$$

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15-26 Propylene is burned with 50 percent excess air during a combustion process. The AF ratio and the temperature at which the water vapor in the products will start condensing are to be determined.

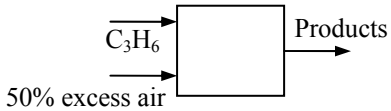
Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis (*a*) The combustion equation in this case can be written as

$$C_3H_6 + 1.5a_{th}[O_2 + 3.76N_2] \longrightarrow 3CO_2 + 3H_2O + 0.5a_{th}O_2 + (1.5\times 3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient for air. It is determined from



O_2 balance: $1.5a_{th} = 3 + 1.5 + 0.5a_{th} \longrightarrow a_{th} = 4.5$

Substituting,

$$C_3H_6 + 6.75[O_2 + 3.76N_2] \longrightarrow 3CO_2 + 3H_2O + 2.25O_2 + 25.38N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$AF = \frac{m_{air}}{m_{fuel}} = \frac{(6.75\times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{(3 \text{ kmol})(12 \text{ kg/kmol}) + (3 \text{ kmol})(2 \text{ kg/kmol})} = \mathbf{22.2\ kg\ air/kg\ fuel}$$

(*b*) The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{prod}}\right)P_{prod} = \left(\frac{3 \text{ kmol}}{33.63 \text{ kmol}}\right)(105 \text{ kPa}) = 9.367 \text{ kPa}$$

Thus,

$$T_{dp} = T_{sat@9.367\ kPa} = \mathbf{44.5°C}$$

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15-27 Butane C_4H_{10} is burned with 200 percent theoretical air. The kmol of water that needs to be sprayed into the combustion chamber per kmol of fuel is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction equation for 200% theoretical air without the additional water is

$$C_4H_{10} + 2a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + E\,O_2 + F\,N_2$$

where a_{th} is the stoichiometric coefficient for air. We have automatically accounted for the 100% excess air by using the factor $2a_{th}$ instead of a_{th} for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

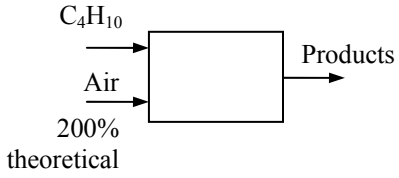
Carbon balance: $B = 4$

Hydrogen balance: $2D = 10 \longrightarrow D = 5$

Oxygen balance: $2 \times 2a_{th} = 2B + D + 2E$

$a_{th} = E$

Nitrogen balance: $2a_{th} \times 3.76 = F$



Solving the above equations, we find the coefficients (E = 6.5, F = 48.88, and a_{th} = 6.5) and write the balanced reaction equation as

$$C_4H_{10} + 13\left[O_2 + 3.76N_2\right] \longrightarrow 4\,CO_2 + 5\,H_2O + 6.5\,O_2 + 48.88\,N_2$$

With the additional water sprayed into the combustion chamber, the balanced reaction equation is

$$C_4H_{10} + 13\left[O_2 + 3.76N_2\right] + N_v\,H_2O \longrightarrow 4\,CO_2 + (5 + N_v)\,H_2O + 6.5\,O_2 + 48.88\,N_2$$

The partial pressure of water in the saturated product mixture at the dew point is

$$P_{v,\text{prod}} = P_{\text{sat@60°C}} = 19.95\ \text{kPa}$$

The vapor mole fraction is

$$y_v = \frac{P_{v,\text{prod}}}{P_{\text{prod}}} = \frac{19.95\ \text{kPa}}{100\ \text{kPa}} = 0.1995$$

The amount of water that needs to be sprayed into the combustion chamber can be determined from

$$y_v = \frac{N_{\text{water}}}{N_{\text{total,product}}} \longrightarrow 0.1995 = \frac{5 + N_v}{4 + 5 + N_v + 6.5 + 48.88} \longrightarrow N_v = \mathbf{9.796\ kmol}$$

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15-28 A fuel mixture of 60% by mass methane, CH_4, and 40% by mass ethanol, C_2H_6O, is burned completely with theoretical air. The required flow rate of air is to be determined.

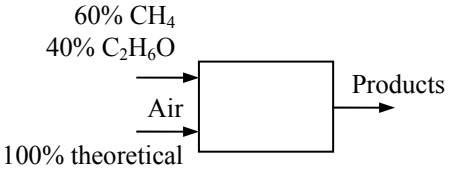
Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis For 100 kg of fuel mixture, the mole numbers are

$$N_{CH4} = \frac{\text{mf}_{CH4}}{M_{CH4}} = \frac{60\text{ kg}}{16\text{ kg/kmol}} = 3.75\text{ kmol}$$

$$N_{C2H6O} = \frac{\text{mf}_{C2H6O}}{M_{C2H6O}} = \frac{40\text{ kg}}{46\text{ kg/kmol}} = 0.8696\text{ kmol}$$



The mole fraction of methane and ethanol in the fuel mixture are

$$x = \frac{N_{CH4}}{N_{CH4} + N_{C2H6O}} = \frac{3.75\text{ kmol}}{(3.75 + 0.8696)\text{ kmol}} = 0.8118$$

$$y = \frac{N_{C2H6O}}{N_{CH4} + N_{C2H6O}} = \frac{0.8696\text{ kmol}}{(3.75 + 0.8696)\text{ kmol}} = 0.1882$$

The combustion equation in this case can be written as

$$x\,CH_4 + y\,C_2H_6O + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + F\,N_2$$

where a_{th} is the stoichiometric coefficient for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

Carbon balance: $x + 2y = B$

Hydrogen balance: $4x + 6y = 2D$

Oxygen balance: $2a_{th} + y = 2B + D$

Nitrogen balance: $3.76a_{th} = F$

Substituting x and y values into the equations and solving, we find the coefficients as

$x = 0.8118$ $\quad B = 1.188$

$y = 0.1882$ $\quad D = 2.188$

$a_{th} = 2.188$ $\quad F = 8.228$

Then, we write the balanced reaction equation as

$$0.8118\,CH_4 + 0.1882\,C_2H_6O + 2.188\left[O_2 + 3.76N_2\right] \longrightarrow 1.188\,CO_2 + 2.188\,H_2O + 8.228\,N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\text{AF} = \frac{m_{air}}{m_{fuel}} = \frac{(2.188 \times 4.76\text{ kmol})(29\text{ kg/kmol})}{(0.8118\text{ kmol})(12 + 4 \times 1)\text{kg/kmol} + (0.1882\text{ kmol})(2 \times 12 + 6 \times 1 + 16)\text{kg/kmol}}$$
$$= 13.94\text{ kg air/kg fuel}$$

Then, the required flow rate of air becomes

$$\dot{m}_{air} = \text{AF}\dot{m}_{fuel} = (13.94)(10\text{ kg/s}) = \mathbf{139.4\ kg/s}$$

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15-29 The volumetric fractions of the constituents of a certain natural gas are given. The AF ratio is to be determined if this gas is burned with the stoichiometric amount of dry air.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2 only.

Properties The molar masses of C, H_2, N_2, O_2, and air are 12 kg/kmol, 2 kg/kmol, 28 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis Considering 1 kmol of fuel, the combustion equation can be written as

$$(0.65\text{CH}_4 + 0.08\text{H}_2 + 0.18\text{N}_2 + 0.03\text{O}_2 + 0.06\text{CO}_2) + a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2) \longrightarrow x\text{CO}_2 + y\text{H}_2\text{O} + z\text{N}_2$$

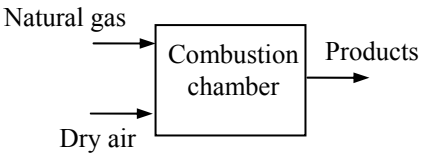
The unknown coefficients in the above equation are determined from mass balances,

$$\text{C}: \ 0.65 + 0.06 = x \qquad \longrightarrow x = 0.71$$

$$\text{H}: \ 0.65 \times 4 + 0.08 \times 2 = 2y \qquad \longrightarrow y = 1.38$$

$$\text{O}_2: \ 0.03 + 0.06 + a_{\text{th}} = x + y/2 \qquad \longrightarrow a_{\text{th}} = 1.31$$

$$\text{N}_2: \ 0.18 + 3.76a_{\text{th}} = z \qquad \longrightarrow z = 5.106$$



Thus,

$$(0.65\text{CH}_4 + 0.08\text{H}_2 + 0.18\text{N}_2 + 0.03\text{O}_2 + 0.06\text{CO}_2) + 1.31(\text{O}_2 + 3.76\text{N}_2) \longrightarrow 0.71\text{CO}_2 + 1.38\text{H}_2\text{O} + 5.106\text{N}_2$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\text{air}} = (1.31 \times 4.76 \text{ kmol})(29 \text{ kg/kmol}) = 180.8 \text{ kg}$$

$$m_{\text{fuel}} = (0.65 \times 16 + 0.08 \times 2 + 0.18 \times 28 + 0.03 \times 32 + 0.06 \times 44)\text{kg} = 19.2 \text{ kg}$$

and

$$\text{AF}_{\text{th}} = \frac{m_{\text{air,th}}}{m_{\text{fuel}}} = \frac{180.8 \text{ kg}}{19.2 \text{ kg}} = \mathbf{9.42 \ kg \ air/kg \ fuel}$$

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15-30 The composition of a certain natural gas is given. The gas is burned with stoichiometric amount of moist air. The AF ratio is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2 only.

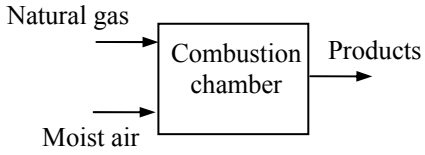
Properties The molar masses of C, H_2, N_2, O_2, and air are 12 kg/kmol, 2 kg/kmol, 28 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The fuel is burned completely with the stoichiometric amount of air, and thus the products will contain only H_2O, CO_2 and N_2, but no free O_2. The moisture in the air does not react with anything; it simply shows up as additional H_2O in the products. Therefore, we can simply balance the combustion equation using dry air, and then add the moisture to both sides of the equation. Considering 1 kmol of fuel, the combustion equation can be written as

$$(0.65\text{CH}_4 + 0.08\text{H}_2 + 0.18\text{N}_2 + 0.03\text{O}_2 + 0.06\text{CO}_2) + a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2) \longrightarrow x\text{CO}_2 + y\text{H}_2\text{O} + z\text{N}_2$$

The unknown coefficients in the above equation are determined from mass balances,

$$\text{C}: \ 0.65 + 0.06 = x \longrightarrow x = 0.71$$
$$\text{H}: \ 0.65 \times 4 + 0.08 \times 2 = 2y \longrightarrow y = 1.38$$
$$\text{O}_2: \ 0.03 + 0.06 + a_{\text{th}} = x + y/2 \longrightarrow a_{\text{th}} = 1.31$$
$$\text{N}_2: \ 0.18 + 3.76a_{\text{th}} = z \longrightarrow z = 5.106$$



Thus,

$$(0.65\text{CH}_4 + 0.08\text{H}_2 + 0.18\text{N}_2 + 0.03\text{O}_2 + 0.06\text{CO}_2) + 1.31(\text{O}_2 + 3.76\text{N}_2) \longrightarrow 0.71\text{CO}_2 + 1.38\text{H}_2\text{O} + 5.106\text{N}_2$$

Next we determine the amount of moisture that accompanies $4.76a_{\text{th}} = (4.76)(1.31) = 6.24$ kmol of dry air. The partial pressure of the moisture in the air is

$$P_{v,\text{in}} = \phi_{\text{air}} P_{\text{sat@25°C}} = (0.85)(3.1698 \text{ kPa}) = 2.694 \text{ kPa}$$

Assuming ideal gas behavior, the number of moles of the moisture in the air ($N_{v,\text{in}}$) is determined to be

$$N_{v,\text{in}} = \left(\frac{P_{v,\text{in}}}{P_{\text{total}}}\right) N_{\text{total}} = \left(\frac{2.694 \text{ kPa}}{101.325 \text{ kPa}}\right)(6.24 + N_{v,\text{in}}) \longrightarrow N_{v,\text{air}} = 0.17 \text{ kmol}$$

The balanced combustion equation is obtained by substituting the coefficients determined earlier and adding 0.17 kmol of H_2O to both sides of the equation,

$$(0.65\text{CH}_4 + 0.08\text{H}_2 + 0.18\text{N}_2 + 0.03\text{O}_2 + 0.06\text{CO}_2) + 1.31(\text{O}_2 + 3.76\text{N}_2) + 0.17\text{H}_2\text{O} \longrightarrow 0.71\text{CO}_2 + 1.55\text{H}_2\text{O} + 5.106\text{N}_2$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\text{air}} = (1.31 \times 4.76 \text{ kmol})(29 \text{ kg/kmol}) + (0.17 \text{ kmol} \times 18 \text{ kg/kmol}) = 183.9 \text{ kg}$$
$$m_{\text{fuel}} = (0.65 \times 16 + 0.08 \times 2 + 0.18 \times 28 + 0.03 \times 32 + 0.06 \times 44)\text{kg} = 19.2 \text{ kg}$$

and

$$\text{AF}_{\text{th}} = \frac{m_{\text{air,th}}}{m_{\text{fuel}}} = \frac{183.9\text{kg}}{19.2\text{kg}} = \mathbf{9.58 \text{ kg air/kg fuel}}$$

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15-31 The composition of a gaseous fuel is given. It is burned with 130 percent theoretical air. The AF ratio and the fraction of water vapor that would condense if the product gases were cooled are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, N_2, and air are 12 kg/kmol, 2 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis (*a*) The fuel is burned completely with excess air, and thus the products will contain H_2O, CO_2, N_2, and some free O_2. Considering 1 kmol of fuel, the combustion equation can be written as

$$(0.45\text{CH}_4 + 0.35\text{H}_2 + 0.20\text{N}_2) + 1.3a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2) \longrightarrow x\text{CO}_2 + y\text{H}_2\text{O} + 0.3a_{\text{th}}\text{O}_2 + z\text{N}_2$$

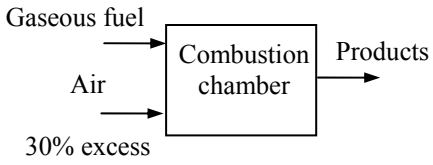
The unknown coefficients in the above equation are determined from mass balances,

$$\text{C}: \ 0.45 = x \qquad \longrightarrow x = 0.45$$

$$\text{H}: \ 0.45 \times 4 + 0.35 \times 2 = 2y \qquad \longrightarrow y = 1.2$$

$$\text{O}_2: \ 1.3a_{\text{th}} = x + y/2 + 0.3a_{\text{th}} \qquad \longrightarrow a_{\text{th}} = 1.05$$

$$\text{N}_2: \ 0.20 + 3.76 \times 1.3a_{\text{th}} = z \qquad \longrightarrow z = 5.332$$



Thus,

$$(0.45\text{CH}_4 + 0.35\text{H}_2 + 0.20\text{N}_2) + 1.365(\text{O}_2 + 3.76\text{N}_2) \longrightarrow 0.45\text{CO}_2 + 1.2\text{H}_2\text{O} + 0.315\text{O}_2 + 5.332\text{N}_2$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\text{air}} = (1.365 \times 4.76 \text{ kmol})(29 \text{ kg/kmol}) = 188.4 \text{ kg}$$

$$m_{\text{fuel}} = (0.45 \times 16 + 0.35 \times 2 + 0.2 \times 28)\text{kg} = 13.5 \text{ kg}$$

and

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{188.4 \text{ kg}}{13.5 \text{ kg}} = \mathbf{13.96 \ kg \ air/kg \ fuel}$$

(*b*) For each kmol of fuel burned, 0.45 + 1.2 + 0.315 + 5.332 = 7.297 kmol of products are formed, including 1.2 kmol of H_2O. Assuming that the dew-point temperature of the products is above 25°C, some of the water vapor will condense as the products are cooled to 25°C. If N_w kmol of H_2O condenses, there will be 1.2 - N_w kmol of water vapor left in the products. The mole number of the products in the gas phase will also decrease to 7.297 − N_w as a result. Treating the product gases (including the remaining water vapor) as ideal gases, N_w is determined by equating the mole fraction of the water vapor to its pressure fraction,

$$\frac{N_v}{N_{\text{prod,gas}}} = \frac{P_v}{P_{\text{prod}}} \longrightarrow \frac{1.2 - N_w}{7.297 - N_w} = \frac{3.1698 \text{ kPa}}{101.325 \text{ kPa}} \longrightarrow N_w = 1.003 \text{ kmol}$$

since $P_v = P_{\text{sat @ 25°C}} = 3.1698$ kPa. Thus the fraction of water vapor that condenses is 1.003/1.2 = 0.836 or **84%**.

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15-32 Problem 15-31 is reconsidered. The effects of varying the percentages of CH4, H2 and N2 making up the fuel and the product gas temperature are to be studied.

Analysis The problem is solved using EES, and the solution is given below.

```
Let's modify this problem to include the fuels butane, ethane, methane, and propane in
pull down menu. Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  aCxHy+bH2+cN2  +  (a*y/4 + a*x+b/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> a*xCO2  +  ((a*y/2)+b) H2O  +  (c+3.76  (a*y/4 + a*x+b/2) (Theo_air/100))  N2 +  (a*y/4 + a*x+b/2)
(Theo_air/100 - 1) O2

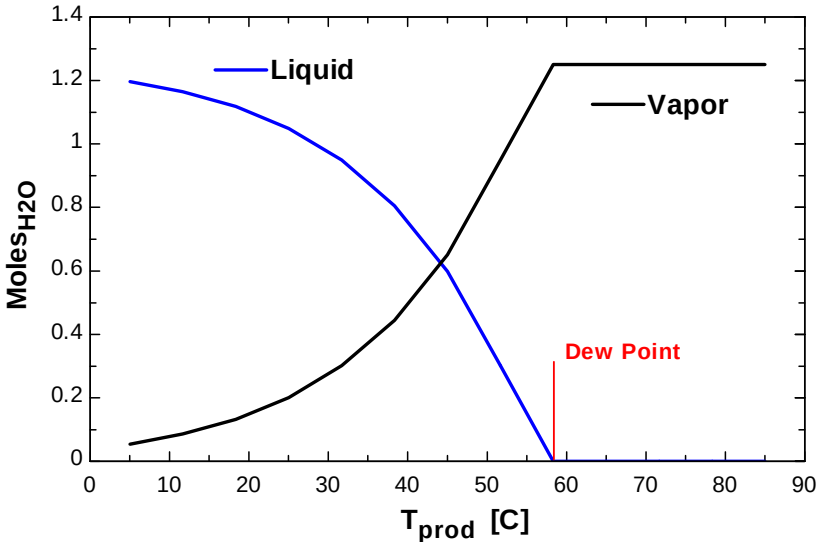
T_prod is the product gas temperature.
Theo_air is the % theoretical air. "
Procedure
H20Cond(P_prod,T_prod,Moles_H2O,M_other:T_DewPoint,Moles_H2O_vap,Moles_H2O_liq,Result$)
P_v = Moles_H2O/(M_other+Moles_H2O)*P_prod
T_DewPoint = temperature(steam,P=P_v,x=0)
IF T_DewPoint <= T_prod then
Moles_H2O_vap = Moles_H2O
Moles_H2O_liq=0
Result$='No condensation occurred'
ELSE
Pv_new=pressure(steam,T=T_prod,x=0)
Moles_H2O_vap=Pv_new/P_prod*M_other/(1-Pv_new/P_prod)
Moles_H2O_liq = Moles_H2O - Moles_H2O_vap
Result$='There is condensation'
ENDIF
END

"Input data from the diagram window"
{P_prod = 101.325 [kPa]
Theo_air = 130 "[%]"
a=0.45
b=0.35
c=0.20
T_prod = 25 [C]}
Fuel$='CH4'
x=1
y=4

"Composition of Product gases:"
A_th = a*y/4 +a* x+b/2
AF_ratio = 4.76*A_th*Theo_air/100*molarmass(Air)/(a*16+b*2+c*28) "[kg_air/kg_fuel]"
Moles_O2=(a*y/4 +a* x+b/2) *(Theo_air/100 - 1)
Moles_N2=c+(3.76*(a*y/4 + a*x+b/2))* (Theo_air/100)
Moles_CO2=a*x
Moles_H2O=a*y/2+b
M_other=Moles_O2+Moles_N2+Moles_CO2
Call H20Cond(P_prod,T_prod,Moles_H2O,M_other:T_DewPoint,Moles_H2O_vap,Moles_H2O_liq,Result$)
Frac_cond = Moles_H2O_liq/Moles_H2O*Convert(, %) "[%]"
"Reaction:  aCxHy+bH2+cN2  +  A_th Theo_air/100 (O2 + 3.76 N2)
<--> a*xCO2  +  (a*y/2+b) H2O  +  (c+3.76  A_th Theo_air/100) N2 +  A_th (Theo_air/100 - 1) O2"
```

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AF_{ratio} [kg_{air}/ kg_{fuel}]	$Frac_{cond}$ [%]	$Moles_{H2O,liq}$	$Moles_{H2O,vap}$	T_{prod} [C]
14.27	95.67	1.196	0.05409	5
14.27	93.16	1.165	0.08549	11.67
14.27	89.42	1.118	0.1323	18.33
14.27	83.92	1.049	0.201	25
14.27	75.94	0.9492	0.3008	31.67
14.27	64.44	0.8055	0.4445	38.33
14.27	47.92	0.599	0.651	45
14.27	24.06	0.3008	0.9492	51.67
14.27	0	0	1.25	58.33
14.27	0	0	1.25	65
14.27	0	0	1.25	71.67
14.27	0	0	1.25	78.33
14.27	0	0	1.25	85



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15-33 Carbon is burned with dry air. The volumetric analysis of the products is given. The AF ratio and the percentage of theoretical air used are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, O_2, and N_2 only.

Properties The molar masses of C, H_2, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis Considering 100 kmol of dry products, the combustion equation can be written as

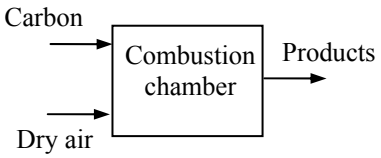
$$xC + a[O_2 + 3.76N_2] \longrightarrow 10.06CO_2 + 0.42CO + 10.69O_2 + 78.83N_2$$

The unknown coefficients *x* and *a* are determined from mass balances,

$$N_2: \ 3.76a = 78.83 \longrightarrow a = 20.965$$

$$C: \ x = 10.06 + 0.42 \longrightarrow x = 10.48$$

$$(\text{Check} O_2: \ a = 10.06 + 0.21 + 10.69 \longrightarrow 20.96 = 20.96)$$



Thus,

$$10.48C + 20.96[O_2 + 3.76N_2] \longrightarrow 10.06CO_2 + 0.42CO + 10.69O_2 + 78.83N_2$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 10.48,

$$C + 2.0[O_2 + 3.76N_2] \longrightarrow 0.96CO_2 + 0.04CO + 1.02O_2 + 7.52N_2$$

(*a*) The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(2.0 \times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{(1 \text{ kmol})(12 \text{ kg/kmol})} = \mathbf{23.0 \ kg \ air/kg \ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$C + 1[O_2 + 3.76N_2] \longrightarrow CO_2 + 3.76N_2$$

Then,

$$\text{Percent theoretical air} = \frac{m_{\text{air,act}}}{m_{\text{air,th}}} = \frac{N_{\text{air,act}}}{N_{\text{air,th}}} = \frac{(2.0)(4.76) \text{ kmol}}{(1.0)(4.76) \text{ kmol}} = \mathbf{200\%}$$

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15-34 Methane is burned with dry air. The volumetric analysis of the products is given. The AF ratio and the percentage of theoretical air used are to be determined.

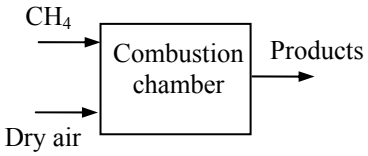
Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis Considering 100 kmol of dry products, the combustion equation can be written as

$$x\text{CH}_4 + a\left[\text{O}_2 + 3.76\text{N}_2\right] \longrightarrow 5.20\text{CO}_2 + 0.33\text{CO} + 11.24\text{O}_2 + 83.23\text{N}_2 + b\text{H}_2\text{O}$$

The unknown coefficients *x*, *a*, and *b* are determined from mass balances,



$$\text{N}_2:\ 3.76a = 83.23 \longrightarrow a = 22.14$$

$$\text{C}:\ x = 5.20 + 0.33 \longrightarrow x = 5.53$$

$$\text{H}:\ 4x = 2b \longrightarrow b = 11.06$$

$$(\text{Check } \text{O}_2:\ a = 5.20 + 0.165 + 11.24 + b/2 \longrightarrow 22.14 = 22.14)$$

Thus,

$$5.53\text{CH}_4 + 22.14\left[\text{O}_2 + 3.76\text{N}_2\right] \longrightarrow 5.20\text{CO}_2 + 0.33\text{CO} + 11.24\text{O}_2 + 83.23\text{N}_2 + 11.06\text{H}_2\text{O}$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 5.53,

$$\text{CH}_4 + 4.0\left[\text{O}_2 + 3.76\text{N}_2\right] \longrightarrow 0.94\text{CO}_2 + 0.06\text{CO} + 2.03\text{O}_2 + 15.05\text{N}_2 + 2\text{H}_2\text{O}$$

(*a*) The air-fuel ratio is determined from its definition,

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(4.0 \times 4.76\ \text{kmol})(29\ \text{kg/kmol})}{(1\ \text{kmol})(12\ \text{kg/kmol}) + (2\ \text{kmol})(2\ \text{kg/kmol})} = \mathbf{34.5\ kg\ air/kg\ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$\text{CH}_4 + a_{\text{th}}\left[\text{O}_2 + 3.76\text{N}_2\right] \longrightarrow \text{CO}_2 + 2\text{H}_2\text{O} + 3.76a_{\text{th}}\text{N}_2$$

$$\text{O}_2: \quad a_{\text{th}} = 1 + 1 \longrightarrow a_{\text{th}} = 2.0$$

Then,

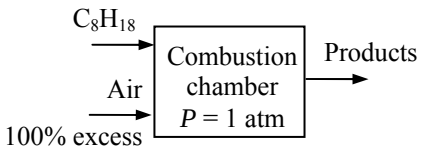
$$\text{Percent theoretical air} = \frac{m_{\text{air,act}}}{m_{\text{air,th}}} = \frac{N_{\text{air,act}}}{N_{\text{air,th}}} = \frac{(4.0)(4.76)\ \text{kmol}}{(2.0)(4.76)\ \text{kmol}} = \mathbf{200\%}$$

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15-35 n-Octane is burned with 100% excess air. The combustion is incomplete. The mole fractions of products and the dew-point temperature of the water vapor in the products are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).



Analysis The combustion reaction for stoichiometric air is

$$C_8H_{18} + 12.5[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + (12.5 \times 3.76)N_2$$

The combustion equation with 100% excess air and incomplete combustion is

$$C_8H_{18} + 2 \times 12.5[O_2 + 3.76N_2] \longrightarrow (0.85 \times 8)\,CO_2 + (0.15 \times 8)\,CO + 9H_2O + x\,O_2 + (2 \times 12.5 \times 3.76)\,N_2$$

The coefficient for CO is determined from a mass balance,

O_2 balance: $25 = 0.85 \times 8 + 0.5 \times 0.15 \times 8 + 0.5 \times 9 + x \longrightarrow x = 13.1$

Substituting,

$$C_8H_{18} + 25[O_2 + 3.76N_2] \longrightarrow 6.8\,CO_2 + 1.2\,CO + 9\,H_2O + 13.1\,O_2 + 94\,N_2$$

The mole fractions of the products are

$$N_{prod} = 6.8 + 1.2 + 9 + 13.1 + 94 = 124.1\,\text{kmol}$$

$$y_{CO2} = \frac{N_{CO2}}{N_{prod}} = \frac{6.8\,\text{kmol}}{124.1\,\text{kmol}} = \mathbf{0.0548}$$

$$y_{CO} = \frac{N_{CO}}{N_{prod}} = \frac{1.2\,\text{kmol}}{124.1\,\text{kmol}} = \mathbf{0.0097}$$

$$y_{H2O} = \frac{N_{H2O}}{N_{prod}} = \frac{9\,\text{kmol}}{124.1\,\text{kmol}} = \mathbf{0.0725}$$

$$y_{O2} = \frac{N_{O2}}{N_{prod}} = \frac{13.1\,\text{kmol}}{124.1\,\text{kmol}} = \mathbf{0.1056}$$

$$y_{N2} = \frac{N_{N2}}{N_{prod}} = \frac{94\,\text{kmol}}{124.1\,\text{kmol}} = \mathbf{0.7575}$$

The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{prod}}\right) P_{prod} = \left(\frac{9\,\text{kmol}}{124.1\,\text{kmol}}\right)(101.325\,\text{kPa}) = 7.348\,\text{kPa}$$

Thus,

$$T_{dp} = T_{sat@7.348\,\text{kPa}} = \mathbf{39.9°C} \qquad \text{(Table A-5 or EES)}$$

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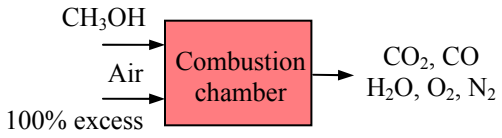
15-36 Methyl alcohol is burned with 100% excess air. The combustion is incomplete. The balanced chemical reaction is to be written and the air-fuel ratio is to be determined.

Assumptions **1** Combustion is incomplete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The balanced reaction equation for stoichiometric air is

$$CH_3OH + a_{th}[O_2 + 3.76N_2] \longrightarrow CO_2 + 2\,H_2O + a_{th} \times 3.76\,N_2$$



The stoicihiometric coefficient a_{th} is determined from an O_2 balance:

$$0.5 + a_{th} = 1 + 1 \longrightarrow a_{th} = 1.5$$

Substituting,

$$CH_3OH + 1.5[O_2 + 3.76N_2] \longrightarrow CO_2 + 2\,H_2O + 1.5 \times 3.76\,N_2$$

The reaction with 100% excess air and incomplete combustion can be written as

$$CH_3OH + 2 \times 1.5[O_2 + 3.76N_2] \longrightarrow 0.60\,CO_2 + 0.40\,CO + 2\,H_2O + x\,O_2 + 2 \times 1.5 \times 3.76\,N_2$$

The coefficient for O_2 is determined from a mass balance,

O_2 balance: $0.5 + 2 \times 1.5 = 0.6 + 0.2 + 1 + x \longrightarrow x = 1.7$

Substituting,

$$CH_3OH + 3[O_2 + 3.76N_2] \longrightarrow 0.6\,CO_2 + 0.4\,CO + 2\,H_2O + 1.7\,O_2 + 11.28\,N_2$$

The air-fuel mass ratio is

$$AF = \frac{m_{air}}{m_{fuel}} = \frac{(3 \times 4.76 \times 29)\,kg}{(1 \times 32)\,kg} = \frac{414.1\,kg}{32\,kg} = \mathbf{12.94\ kg\ air/kg\ fuel}$$

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15-37 Ethyl alcohol is burned with stoichiometric amount of air. The combustion is incomplete. The apparent molecular weight of the products is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, OH, and N_2 only.

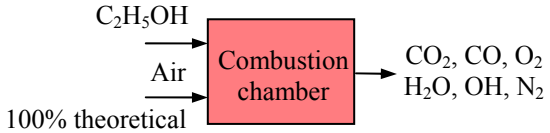
Properties The molar masses of C, H_2, OH, N_2 and air are 12 kg/kmol, 2 kg/kmol, 17 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction with stoichiometric air is

$$C_2H_5OH + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 2\,CO_2 + 3H_2O + a_{th} \times 3.76\,N_2$$

where

$$0.5 + a_{th} = 2 + 1.5 \longrightarrow a_{th} = 3$$



Substituting,

$$C_2H_5OH + 3\left[O_2 + 3.76N_2\right] \longrightarrow 2\,CO_2 + 3H_2O + 3 \times 3.76\,N_2$$

The balanced reaction equation with incomplete combustion is

$$C_2H_5OH + 3\left[O_2 + 3.76N_2\right] \longrightarrow 2(0.90\,CO_2 + 0.10\,CO) + 3\left(0.95H_2O + 0.1\,OH\right) + bO_2 + 3 \times 3.76\,N_2$$

O_2 balance: $0.5+3 = 1.8+0.1+3.15/2+b \;\rightarrow\; b = 0.025$

which can be written as

$$C_2H_5OH + 3\left[O_2 + 3.76N_2\right] \longrightarrow 1.80\,CO_2 + 0.20\,CO + 2.85\,H_2O + 0.3\,OH + 0.025O_2 + 11.28\,N_2$$

The total moles of the products is

$$N_m = 1.8 + 0.2 + 2.85 + 0.3 + 0.025 + 11.28 = 16.64\ \text{kmol}$$

The apparent molecular weight of the product gas is

$$M_m = \frac{m_m}{N_m} = \frac{(1.8 \times 44 + 0.20 \times 28 + 2.85 \times 18 + 0.3 \times 17 + 0.025 \times 32 + 11.28 \times 28)\ \text{kg}}{16.64\ \text{kmol}} = \mathbf{27.83\ kg/kmol}$$

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15-38 Coal whose mass percentages are specified is burned with stoichiometric amount of air. The mass fractions of the products and the air-fuel ratio are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, SO_2, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, O_2, S, and air are 12, 2, 32, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kg of coal for simplicity. Noting that the mass percentages in this case correspond to the masses of the constituents, the mole numbers of the constituent of the coal are determined to be

$$N_C = \frac{m_C}{M_C} = \frac{79.61\text{ kg}}{12\text{ kg/kmol}} = 6.634\text{ kmol}$$

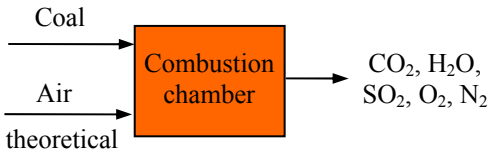
$$N_{H2} = \frac{m_{H2}}{M_{H2}} = \frac{4.66\text{ kg}}{2\text{ kg/kmol}} = 2.33\text{ kmol}$$

$$N_{O2} = \frac{m_{O2}}{M_{O2}} = \frac{4.76\text{ kg}}{32\text{ kg/kmol}} = 0.1488\text{ kmol}$$

$$N_{N2} = \frac{m_{N2}}{M_{N2}} = \frac{1.83\text{ kg}}{28\text{ kg/kmol}} = 0.06536\text{ kmol}$$

$$N_S = \frac{m_S}{M_S} = \frac{0.52\text{ kg}}{32\text{ kg/kmol}} = 0.01625\text{ kmol}$$

79.61% C
4.66% H_2
4.76% O_2
1.83% N_2
0.52% S
8.62% ash
(by mass)



Ash consists of the non-combustible matter in coal. Therefore, the mass of ash content that enters the combustion chamber is equal to the mass content that leaves. Disregarding this non-reacting component for simplicity, the combustion equation may be written as

$$6.634C + 2.33H_2 + 0.1488O_2 + 0.06536N_2 + 0.01625S + a_{th}(O_2 + 3.76N_2) \longrightarrow xCO_2 + yH_2O + zSO_2 + wN_2$$

Performing mass balances for the constituents gives

C balance: $x = 6.634$

H_2 balance: $y = 2.33$

S balance: $z = 0.01625$

O_2 balance: $0.1488 + a_{th} = x + 0.5y + z \longrightarrow a_{th} = 6.634 + 0.5(2.33) + 0.01625 - 0.1488 = 7.667$

N_2 balance: $w = 0.06536 + 3.76a_{th} = 0.06536 + 3.76 \times 7.667 = 28.89$

Substituting, the balanced combustion equation without the ash becomes

$$6.634C + 2.33H_2 + 0.1488O_2 + 0.06536N_2 + 0.01625S + 7.667(O_2 + 3.76N_2) \longrightarrow 6.634CO_2 + 2.33H_2O + 0.01625SO_2 + 28.89N_2$$

The mass fractions of the products are

$$m_{total} = 6.634 \times 44 + 2.33 \times 18 + 0.01625 \times 64 + 228.89 \times 28 = 1144\text{ kg}$$

$$mf_{CO2} = \frac{m_{CO2}}{m_{total}} = \frac{(6.634 \times 44)\text{ kg}}{1144\text{ kg}} = \mathbf{0.2552}$$

$$mf_{H2O} = \frac{m_{H2O}}{m_{total}} = \frac{(2.33 \times 18)\text{ kg}}{1144\text{ kg}} = \mathbf{0.0367}$$

$$mf_{SO2} = \frac{m_{SO2}}{m_{total}} = \frac{(0.01625 \times 64)\text{ kg}}{1144\text{ kg}} = \mathbf{0.00091}$$

$$mf_{N2} = \frac{m_{N2}}{m_{total}} = \frac{(28.89 \times 28)\text{ kg}}{1144\text{ kg}} = \mathbf{0.7072}$$

The air-fuel mass ratio is then

$$AF = \frac{m_{air}}{m_{fuel}} = \frac{(7.667 \times 4.76 \times 29)\text{ kg}}{(6.634 \times 12 + 2.33 \times 2 + 0.1488 \times 32 + 0.06536 \times 28 + 0.01625 \times 32)\text{ kg}} = \frac{1058\text{ kg}}{91.38\text{ kg}} = \mathbf{11.58\text{ kg air/kg fuel}}$$

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15-39 Coal whose mass percentages are specified is burned with 40% excess air. The air-fuel ratio and the apparent molecular weight of the product gas are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, SO_2, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, O_2, S, and air are 12, 2, 32, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kg of coal for simplicity. Noting that the mass percentages in this case correspond to the masses of the constituents, the mole numbers of the constituent of the coal are determined to be

$$N_{\text{C}} = \frac{m_{\text{C}}}{M_{\text{C}}} = \frac{67.40\ \text{kg}}{12\ \text{kg/kmol}} = 5.617\ \text{kmol}$$

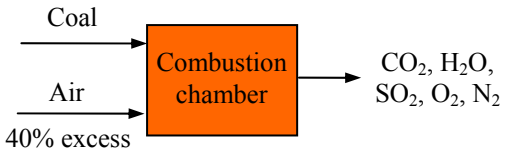
$$N_{\text{H2}} = \frac{m_{\text{H2}}}{M_{\text{H2}}} = \frac{5.31\ \text{kg}}{2\ \text{kg/kmol}} = 2.655\ \text{kmol}$$

$$N_{\text{O2}} = \frac{m_{\text{O2}}}{M_{\text{O2}}} = \frac{15.11\ \text{kg}}{32\ \text{kg/kmol}} = 0.4722\ \text{kmol}$$

$$N_{\text{N2}} = \frac{m_{\text{N2}}}{M_{\text{N2}}} = \frac{1.44\ \text{kg}}{28\ \text{kg/kmol}} = 0.05143\ \text{kmol}$$

$$N_{\text{S}} = \frac{m_{\text{S}}}{M_{\text{S}}} = \frac{2.36\ \text{kg}}{32\ \text{kg/kmol}} = 0.07375\ \text{kmol}$$

67.40% C
5.31% H_2
15.11% O_2
1.44% N_2
2.36% S
8.38% ash
(by mass)



The mole number of the mixture and the mole fractions are

$$N_m = 5.617 + 2.655 + 0.4722 + 0.05143 + 0.07375 = 8.869\ \text{kmol}$$

$$y_{\text{C}} = \frac{N_{\text{C}}}{N_m} = \frac{5.617\ \text{kmol}}{8.869\ \text{kmol}} = 0.6333$$

$$y_{\text{H2}} = \frac{N_{\text{H2}}}{N_m} = \frac{2.655\ \text{kmol}}{8.869\ \text{kmol}} = 0.2994$$

$$y_{\text{O2}} = \frac{N_{\text{O2}}}{N_m} = \frac{0.4722\ \text{kmol}}{8.869\ \text{kmol}} = 0.05323$$

$$y_{\text{N2}} = \frac{N_{\text{N2}}}{N_m} = \frac{0.05143\ \text{kmol}}{8.869\ \text{kmol}} = 0.00580$$

$$y_{\text{S}} = \frac{N_{\text{S}}}{N_m} = \frac{0.07375\ \text{kmol}}{8.869\ \text{kmol}} = 0.00832$$

Ash consists of the non-combustible matter in coal. Therefore, the mass of ash content that enters the combustion chamber is equal to the mass content that leaves. Disregarding this non-reacting component for simplicity, the combustion equation may be written as

$$0.6333\text{C} + 0.2994\text{H}_2 + 0.05323\text{O}_2 + 0.00580\text{N}_2 + 0.00832\text{S} + 1.4a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2) \longrightarrow x\text{CO}_2 + y\text{H}_2\text{O} + z\text{SO}_2 + k\text{N}_2 + m\text{O}_2$$

According to the species balances,

C balance: $x = 0.6333$

H_2 balance: $y = 0.2994$

S balance: $z = 0.00832$

O_2 balance:

$$0.05323 + a_{\text{th}} = x + 0.5y + z$$

$$a_{\text{th}} = 0.6333 + 0.5 \times 0.2994 + 0.00832 - 0.05323 = 0.7381$$

N_2 balance: $k = 0.00580 + 1.4 \times 3.76a_{\text{th}} = 0.00580 + 1.4 \times 3.76 \times 0.7381 = 3.891$

$$m = 0.4a_{\text{th}} = 0.4 \times 0.7381 = 0.2952$$

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Substituting,

$$0.6333\text{C} + 0.2994\text{H}_2 + 0.05323\text{O}_2 + 0.00580\text{N}_2 + 0.00832\text{S} + 1.033(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.6333\text{CO}_2 + 0.2994\text{H}_2\text{O} + 0.00832\text{SO}_2 + 3.891\text{N}_2 + 0.2952\text{O}_2$$

The total mass of the products is

$$m_{\text{total}} = 0.6333 \times 44 + 0.2994 \times 18 + 0.00832 \times 64 + 3.891 \times 28 + 0.2952 \times 32 = 152.2 \text{ kg}$$

The total mole number of the products is

$$N_m = 0.6333 + 0.2994 + 0.00832 + 3.891 + 0.2952 = 5.127 \text{ kmol}$$

The apparent molecular weight of the product gas is

$$M_m = \frac{m_m}{N_m} = \frac{152.2 \text{ kg}}{5.127 \text{ kmol}} = \mathbf{29.68 \text{ kg/kmol}}$$

The air-fuel mass ratio is then

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(1.033 \times 4.76 \times 29) \text{ kg}}{(0.6333 \times 12 + 0.2994 \times 2 + 0.05323 \times 32 + 0.00580 \times 28 + 0.00832 \times 32) \text{ kg}}$$
$$= \frac{142.6 \text{ kg}}{10.33 \text{ kg}}$$
$$= \mathbf{13.80 \text{ kg air/kg fuel}}$$

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Enthalpy of Formation and Enthalpy of Combustion

15-40C For combustion processes the enthalpy of reaction is referred to as the enthalpy of combustion, which represents the amount of heat released during a steady-flow combustion process.

15-41C Enthalpy of formation is the enthalpy of a substance due to its chemical composition. The enthalpy of formation is related to elements or compounds whereas the enthalpy of combustion is related to a particular fuel.

15-42C The heating value is called the higher heating value when the H_2O in the products is in the liquid form, and it is called the lower heating value when the H_2O in the products is in the vapor form. The heating value of a fuel is equal to the absolute value of the enthalpy of combustion of that fuel.

15-43C If the combustion of a fuel results in a single compound, the enthalpy of formation of that compound is identical to the enthalpy of combustion of that fuel.

15-44C Yes.

15-45C No. The enthalpy of formation of N_2 is simply assigned a value of zero at the standard reference state for convenience.

15-46C 1 kmol of H_2. This is evident from the observation that when chemical bonds of H_2 are destroyed to form H_2O a large amount of energy is released.

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15-47 The enthalpy of combustion of methane at a 25°C and 1 atm is to be determined using the data from Table A-26 and to be compared to the value listed in Table A-27.

Assumptions The water in the products is in the liquid phase.

Analysis The stoichiometric equation for this reaction is

$$CH_4 + 2[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O(\ell) + 7.52N_2$$

Both the reactants and the products are at the standard reference state of 25°C and 1 atm. Also, N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then the enthalpy of combustion of CH_4 becomes

$$h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO_2} + \left(N\bar{h}^\circ_f\right)_{H_2O} - \left(N\bar{h}^\circ_f\right)_{CH_4}$$

Using $\bar{h}^\circ_f$ values from Table A-26,

$$\begin{aligned} h_C &= (1\text{ kmol})(-393{,}520\text{ kJ/kmol}) + (2\text{ kmol})(-285{,}830\text{ kJ/kmol}) \\ &\quad - (1\text{ kmol})(-74{,}850\text{ kJ/kmol}) \\ &= \mathbf{-890{,}330\text{ kJ}}\ (\text{per kmol } CH_4) \end{aligned}$$

The listed value in Table A-27 is -890,868 kJ/kmol, which is almost identical to the calculated value. Since the water in the products is assumed to be in the liquid phase, this h_c value corresponds to the higher heating value of CH_4.

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15-48 Problem 15-47 is reconsidered. The effect of temperature on the enthalpy of combustion is to be studied.

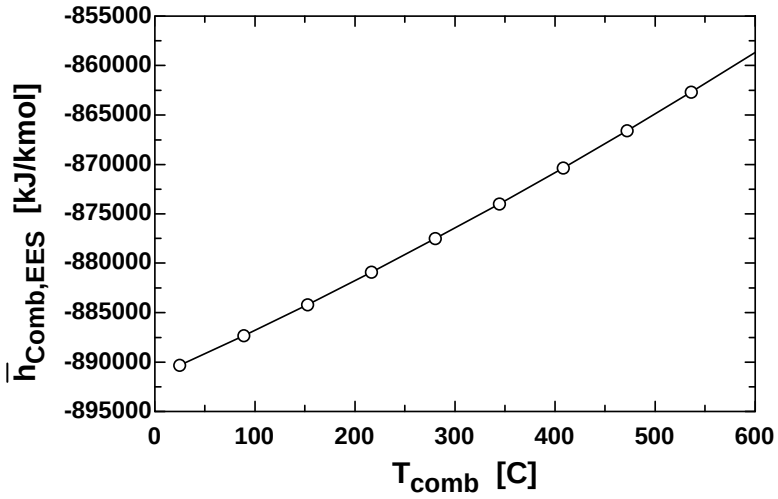
Analysis The problem is solved using EES, and the solution is given below.

```
Fuel$ = 'Methane (CH4)'
T_comb =25 [C]
T_fuel = T_comb +273 "[K]"
T_air1 = T_comb +273 "[K]"
T_prod =T_comb +273 "[K]"
h_bar_comb_TableA27 = -890360 [kJ/kmol]
"For  theoretical dry air, the complete combustion equation is"
"CH4 + A_th(O2+3.76 N2)=1 CO2+2 H2O +  A_th (3.76) N2 "

A_th*2=1*2+2*1 "theoretical O balance"

"Apply First Law SSSF"
h_fuel_EES=enthalpy(CH4,T=298) "[kJ/kmol]"
h_fuel_TableA26=-74850 "[kJ/kmol]"
h_bar_fg_H2O=enthalpy(Steam_iapws,T=298,x=1)-enthalpy(Steam_iapws,T=298,x=0) "[kJ/kmol]"
HR=h_fuel_EES+ A_th*enthalpy(O2,T=T_air1)+A_th*3.76 *enthalpy(N2,T=T_air1) "[kJ/kmol]"
HP=1*enthalpy(CO2,T=T_prod)+2*(enthalpy(H2O,T=T_prod)-h_bar_fg_H2O)+A_th*3.76*
enthalpy(N2,T=T_prod) "[kJ/kmol]"
h_bar_Comb_EES=(HP-HR) "[kJ/kmol]"
PercentError=ABS(h_bar_Comb_EES-h_bar_comb_TableA27)/ABS(h_bar_comb_TableA27)*Convert(, %) "[%]"
```

$h_{CombEES}$ [kJ/kmol]	T_{Comb} [C]
-890335	25
-887336	88.89
-884186	152.8
-880908	216.7
-877508	280.6
-873985	344.4
-870339	408.3
-866568	472.2
-862675	536.1
-858661	600



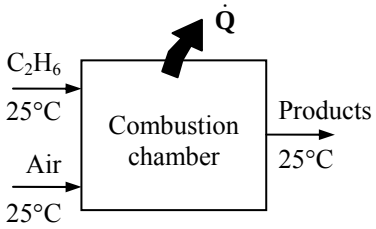
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15-49 Ethane is burned with stoichiometric amount of air. The heat transfer is to be determined if both the reactants and products are at 25°C.

Assumptions The water in the products is in the vapor phase.

Analysis The stoichiometric equation for this reaction is

$$C_2H_6 + 3.5[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 3H_2O + 13.16N_2$$



Since both the reactants and the products are at the standard reference state of 25°C and 1 atm, the heat transfer for this process is equal to enthalpy of combustion. Note that N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then,

$$Q = h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO2} + \left(N\bar{h}^\circ_f\right)_{H2O} - \left(N\bar{h}^\circ_f\right)_{C2H6}$$

Using $\bar{h}^\circ_f$ values from Table A-26,

$$Q = h_C = (2\text{ kmol})(-393{,}520\text{ kJ/kmol}) + (3\text{ kmol})(-241{,}820\text{ kJ/kmol}) - (1\text{ kmol})(-84{,}680\text{ kJ/kmol})$$
$$= \mathbf{-1{,}427{,}820\text{ kJ/kmol C2H6}}$$

15-50 Ethane is burned with stoichiometric amount of air at 1 atm and 25°C. The minimum pressure of the products which will assure that the water in the products will be in vapor form is to be determined.

Assumptions The water in the products is in the vapor phase.

Analysis The stoichiometric equation for this reaction is

$$C_2H_6 + 3.5[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 3H_2O + 13.16N_2$$

At the minimum pressure, the product mixture will be saturated with water vapor and

$$P_v = P_{\text{sat@25°C}} = 3.1698\text{ kPa}$$

The mole fraction of water in the products is

$$y_v = \frac{N_{H2O}}{N_{prod}} = \frac{3\text{ kmol}}{(2+3+13.16)\text{ kmol}} = 0.1652$$

The minimum pressure of the products is then

$$P_{min} = \frac{P_v}{y_v} = \frac{3.1698\text{ kPa}}{0.1652} = \mathbf{19.2\text{ kPa}}$$

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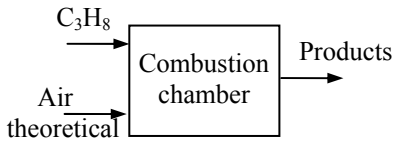
15-51 The higher and lower heating values of liquid propane are to be determined and compared to the listed values.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, O_2, H_2, and air are 12, 32, 2, and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion reaction with stoichiometric air is

$$C_3H_8(l) + 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$



Both the reactants and the products are taken to be at the standard reference state of 25°C and 1 atm for the calculation of heating values. The heat transfer for this process is equal to enthalpy of combustion. Note that N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then,

$$q = h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO2} + \left(N\bar{h}^\circ_f\right)_{H2O} - \left(N\bar{h}^\circ_f\right)_{C3H8}$$

The $\bar{h}^\circ_f$ of liquid propane is obtained by adding $\bar{h}_{fg}$ of propane at 25°C to $\bar{h}^\circ_f$ of gas propane (103,850 + 44.097 × 335 = 118,620 kJ/kmol). For the HHV, the water in the products is taken to be liquid. Then,

$$\begin{aligned} h_C &= (3\text{ kmol})(-393{,}520\text{ kJ/kmol}) + (4\text{ kmol})(-285{,}830\text{ kJ/kmol}) - (1\text{ kmol})(-118{,}620\text{ kJ/kmol}) \\ &= -2{,}205{,}260\text{ kJ/kmol propane} \end{aligned}$$

The HHV of the liquid propane is

$$\text{HHV} = \frac{-h_C}{M_m} = \frac{2{,}205{,}260\text{ kJ/kmol } C_3H_8}{44.097\text{ kg/kmol } C_3H_8} = \mathbf{50{,}010\text{ kJ/kg } C_3H_8}$$

The listed value from Table A-27 is **50,330 kJ/kg**. For the LHV, the water in the products is taken to be vapor. Then,

$$\begin{aligned} h_C &= (3\text{ kmol})(-393{,}520\text{ kJ/kmol}) + (4\text{ kmol})(-241{,}820\text{ kJ/kmol}) - (1\text{ kmol})(-118{,}620\text{ kJ/kmol}) \\ &= -2{,}029{,}220\text{ kJ/kmol propane} \end{aligned}$$

The LHV of the propane is then

$$\text{LHV} = \frac{-h_C}{M_m} = \frac{2{,}029{,}220\text{ kJ/kmol } C_3H_8}{44.097\text{ kg/kmol } C_3H_8} = \mathbf{46{,}020\text{ kJ/kg } C_3H_8}$$

The listed value from Table A-27 is **46,340 kJ/kg**. The calculated and listed values are practically identical.

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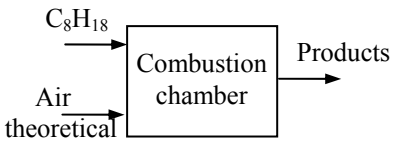
15-52 The higher and lower heating values of gaseous octane are to be determined and compared to the listed values.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, O_2, H_2, and air are 12, 32, 2, and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion reaction with stoichiometric air is

$$C_8H_{18} + 12.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 47N_2$$



Both the reactants and the products are taken to be at the standard reference state of 25°C and 1 atm for the calculation of heating values. The heat transfer for this process is equal to enthalpy of combustion. Note that N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then,

$$q = h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO2} + \left(N\bar{h}^\circ_f\right)_{H2O} - \left(N\bar{h}^\circ_f\right)_{C8H18}$$

For the HHV, the water in the products is taken to be liquid. Then,

$$\begin{aligned} h_C &= (8\,\text{kmol})(-393{,}520\,\text{kJ/kmol}) + (9\,\text{kmol})(-285{,}830\,\text{kJ/kmol}) - (1\,\text{kmol})(-208{,}450\,\text{kJ/kmol}) \\ &= -5{,}512{,}180\,\text{kJ/kmol octane} \end{aligned}$$

The HHV of the gaseous octane is

$$\text{HHV} = \frac{-h_C}{M_m} = \frac{5{,}512{,}180\,\text{kJ/kmol}\,C_8H_{18}}{114.231\,\text{kg/kmol}\,C_8H_{18}} = \mathbf{48{,}250\ kJ/kg\ C_8H_{18}}$$

The listed value for liquid octane from Table A-27 is 47,890 kJ/kg. Adding the enthalpy of vaporization of octane to this value (47,890+363=48,253), the higher heating value of gaseous octane becomes **48,253 kJ/kg** octane. This value is practically identical to the calculated value. For the LHV, the water in the products is taken to be vapor. Then,

$$\begin{aligned} h_C &= (8\,\text{kmol})(-393{,}520\,\text{kJ/kmol}) + (9\,\text{kmol})(-241{,}820\,\text{kJ/kmol}) - (1\,\text{kmol})(-208{,}450\,\text{kJ/kmol}) \\ &= -5{,}116{,}090\,\text{kJ/kmol octane} \end{aligned}$$

The LHV of the gaseous octane is then

$$\text{HHV} = \frac{-h_C}{M_m} = \frac{5{,}116{,}090\,\text{kJ/kmol}\,C_8H_{18}}{114.231\,\text{kg/kmol}\,C_8H_{18}} = \mathbf{44{,}790\ kJ/kg\ C_8H_{18}}$$

The listed value for liquid octane from Table A-27 is 44,430 kJ/kg. Adding the enthalpy of vaporization of octane to this value (44,430+363=44,793), the lower heating value of gaseous octane becomes **44,793 kJ/kg** octane. This value is practically identical to the calculated value.

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15-53 The higher and lower heating values of coal from Illinois are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, SO_2, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, O_2, S, and air are 12, 2, 32, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kg of coal for simplicity. Noting that the mass percentages in this case correspond to the masses of the constituents, the mole numbers of the constituent of the coal are determined to be

$$N_{\text{C}} = \frac{m_{\text{C}}}{M_{\text{C}}} = \frac{67.40\text{ kg}}{12\text{ kg/kmol}} = 5.617\text{ kmol}$$

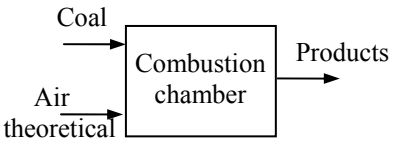
$$N_{\text{H2}} = \frac{m_{\text{H2}}}{M_{\text{H2}}} = \frac{5.31\text{ kg}}{2\text{ kg/kmol}} = 2.655\text{ kmol}$$

$$N_{\text{O2}} = \frac{m_{\text{O2}}}{M_{\text{O2}}} = \frac{15.11\text{ kg}}{32\text{ kg/kmol}} = 0.4722\text{ kmol}$$

$$N_{\text{N2}} = \frac{m_{\text{N2}}}{M_{\text{N2}}} = \frac{1.44\text{ kg}}{28\text{ kg/kmol}} = 0.05143\text{ kmol}$$

$$N_{\text{S}} = \frac{m_{\text{S}}}{M_{\text{S}}} = \frac{2.36\text{ kg}}{32\text{ kg/kmol}} = 0.07375\text{ kmol}$$

67.40% C
5.31% H_2
15.11% O_2
1.44% N_2
2.36% S
8.38% ash
(by mass)



The mole number of the mixture and the mole fractions are

$$N_m = 5.617 + 2.655 + 0.4722 + 0.05143 + 0.07375 = 8.869\text{ kmol}$$

$$y_{\text{C}} = \frac{N_{\text{C}}}{N_m} = \frac{5.617\text{ kmol}}{8.869\text{ kmol}} = 0.6333$$

$$y_{\text{H2}} = \frac{N_{\text{H2}}}{N_m} = \frac{2.655\text{ kmol}}{8.869\text{ kmol}} = 0.2994$$

$$y_{\text{O2}} = \frac{N_{\text{O2}}}{N_m} = \frac{0.4722\text{ kmol}}{8.869\text{ kmol}} = 0.05323$$

$$y_{\text{N2}} = \frac{N_{\text{N2}}}{N_m} = \frac{0.05143\text{ kmol}}{8.869\text{ kmol}} = 0.00580$$

$$y_{\text{S}} = \frac{N_{\text{S}}}{N_m} = \frac{0.07375\text{ kmol}}{8.869\text{ kmol}} = 0.00832$$

Ash consists of the non-combustible matter in coal. Therefore, the mass of ash content that enters the combustion chamber is equal to the mass content that leaves. Disregarding this non-reacting component for simplicity, the combustion equation may be written as

$$0.6333\text{C} + 0.2994\text{H}_2 + 0.05323\text{O}_2 + 0.00580\text{N}_2 + 0.00832\text{S} + a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2) \longrightarrow x\text{CO}_2 + y\text{H}_2\text{O} + z\text{SO}_2 + k\text{N}_2$$

According to the species balances,

C balance: $x = 0.6333$

H_2 balance: $y = 0.2994$

S balance: $z = 0.00832$

O_2 balance:

$0.05323 + a_{\text{th}} = x + 0.5y + z$

$a_{\text{th}} = 0.6333 + 0.5 \times 0.2994 + 0.00832 - 0.05323 = 0.7381$

N_2 balance: $k = 0.00580 + 3.76a_{\text{th}} = 0.00580 + 3.76 \times 0.7381 = 2.781$

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Substituting,

$$0.6333\text{C} + 0.2994\text{H}_2 + 0.05323\text{O}_2 + 0.00580\text{N}_2 + 0.00832\text{S} + 0.7381(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.6333\text{CO}_2 + 0.2994\text{H}_2\text{O} + 0.00832\text{SO}_2 + 2.781\text{N}_2$$

Both the reactants and the products are taken to be at the standard reference state of 25°C and 1 atm for the calculation of heating values. The heat transfer for this process is equal to enthalpy of combustion. Note that C, S, H_2, N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then,

$$q = h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{\text{CO2}} + \left(N\bar{h}^\circ_f\right)_{\text{H2O}} + \left(N\bar{h}^\circ_f\right)_{\text{SO2}}$$

For the HHV, the water in the products is taken to be liquid. Then,

$$\begin{aligned} h_C &= (0.6333\,\text{kmol})(-393{,}520\,\text{kJ/kmol}) + (0.2994\,\text{kmol})(-285{,}830\,\text{kJ/kmol}) \\ &\quad + (0.00832\,\text{kmol})(-297{,}100\,\text{kJ/kmol}) \\ &= -337{,}270\,\text{kJ/kmol coal} \end{aligned}$$

The apparent molecular weight of the coal is

$$\begin{aligned} M_m &= \frac{m_m}{N_m} = \frac{(0.6333\times 12 + 0.2994\times 2 + 0.05323\times 32 + 0.00580\times 28 + 0.00832\times 32)\,\text{kg}}{(0.6333 + 0.2994 + 0.05323 + 0.00580 + 0.00832)\,\text{kmol}} \\ &= \frac{10.33\,\text{kg}}{1.000\,\text{kmol}} = 10.33\,\text{kg/kmol coal} \end{aligned}$$

The HHV of the coal is then

$$\text{HHV} = \frac{-h_C}{M_m} = \frac{337{,}270\,\text{kJ/kmol coal}}{10.33\,\text{kg/kmol coal}} = \mathbf{32{,}650\ kJ/kg\ coal}$$

For the LHV, the water in the products is taken to be vapor. Then,

$$\begin{aligned} h_C &= (0.6333\,\text{kmol})(-393{,}520\,\text{kJ/kmol}) + (0.2994\,\text{kmol})(-241{,}820\,\text{kJ/kmol}) \\ &\quad + (0.00832\,\text{kmol})(-297{,}100\,\text{kJ/kmol}) \\ &= -324{,}090\,\text{kJ/kmol coal} \end{aligned}$$

The LHV of the coal is then

$$\text{LHV} = \frac{-h_C}{M_m} = \frac{324{,}090\,\text{kJ/kmol coal}}{10.33\,\text{kg/kmol coal}} = \mathbf{31{,}370\ kJ/kg\ coal}$$

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First Law Analysis of Reacting Systems

15-54C In this case $\Delta U + W_b = \Delta H$, and the conservation of energy relation reduces to the form of the steady-flow energy relation.

15-55C The heat transfer will be the same for all cases. The excess oxygen and nitrogen enters and leaves the combustion chamber at the same state, and thus has no effect on the energy balance.

15-56C For case (*b*), which contains the maximum amount of nonreacting gases. This is because part of the chemical energy released in the combustion chamber is absorbed and transported out by the nonreacting gases.

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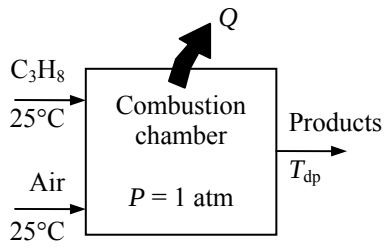
15-57 Propane is burned with an air-fuel ratio of 25. The heat transfer per kilogram of fuel burned when the temperature of the products is such that liquid water just begins to form in the products is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete. **5** The reactants are at 25°C and 1 atm. **6** The fuel is in vapor phase.

Properties The molar masses of propane and air are 44 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis The mass of air per kmol of fuel is

$$m_{\text{air}} = (\text{AF})m_{\text{fuel}}$$
$$= (25\ \text{kg air/kg fuel})(1\times 44\ \text{kg/kmol fuel}) = 1100\ \text{kg air/kmol fuel}$$



The mole number of air per kmol of fuel is then

$$N_{\text{air}} = \frac{m_{\text{air}}}{M_{\text{air}}} = \frac{1100\ \text{kg air/kmol fuel}}{29\ \text{kg air/kmol air}} = 37.93\ \text{kmol air/kmol fuel}$$

The combustion equation can be written as

$$C_3H_8 + (37.93/4.76)(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + xO_2 + (37.93/4.76)\times 3.76N_2$$

The coefficient for O_2 is obtained from O_2 balance:

$$(37.93/4.76) = 3 + 2 + x \longrightarrow x = 2.968$$

Substituting, $\quad C_3H_8 + 7.968(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 2.968O_2 + 29.96N_2$

The mole fraction of water in the products is

$$y_v = \frac{N_{\text{H2O}}}{N_{\text{prod}}} = \frac{4\ \text{kmol}}{(3+4+2.968+29.96)\ \text{kmol}} = \frac{4\ \text{kmol}}{39.93\ \text{kmol}} = 0.1002$$

The partial pressure of water vapor at 1 atm total pressure is

$$P_v = y_v P = (0.1002)(101.325\ \text{kPa}) = 10.15\ \text{kPa}$$

When this mixture is at the dew-point temperature, the water vapor pressure is the same as the saturation pressure. Then,

$$T_{\text{dp}} = T_{\text{sat@10.15 kPa}} = 46.1°\text{C} = 319.1\ \text{K} \cong 320\ \text{K}$$

We obtain properties at 320 K (instead of 319.1 K) to avoid iterations in the ideal gas tables. The heat transfer for this combustion process is determined from the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$, kJ/kmol	$\bar{h}_{298\text{K}}$, kJ/kmol	$\bar{h}_{320\text{K}}$, kJ/kmol
C_3H_8	-103,850	---	---
O_2	0	8682	9325
N_2	0	8669	9306
H_2O (g)	-241,820	9904	10,639
CO_2	-393,520	9364	10,186

Substituting,

$$-\bar{Q}_{\text{out}} = (3)(-393{,}520 + 10{,}186 - 9364) + (4)(-241{,}820 + 10{,}639 - 9904) + (2.968)(0 + 9325 - 8682)$$
$$+ (29.96)(0 + 9306 - 8669) - (1)(-103{,}850) - 0$$
$$= -2{,}017{,}590\ \text{kJ/kmol}\ C_3H_8$$

or $\quad \bar{Q}_{\text{out}} = 2{,}017{,}590\ \text{kJ/kmol}\ C_3H_8$

Then the heat transfer per kg of fuel is

$$Q_{\text{out}} = \frac{\bar{Q}_{\text{out}}}{M_{\text{fuel}}} = \frac{2{,}017{,}590\ \text{kJ/kmol fuel}}{44\ \text{kg/kmol}} = \mathbf{45{,}850\ kJ/kg\ C_3H_8}$$

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15-58 n-Octane is burned with 100 percent excess air. The heat transfer per kilogram of fuel burned for a product temperature of 257°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete. **5** The fuel is in vapor phase.

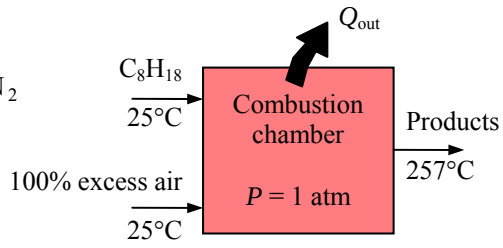
Properties The molar masses of propane and air are 44 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion reaction for stoichiometric air is

$$C_8H_{18} + 12.5[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + (12.5 \times 3.76)N_2$$

The combustion equation with 100% excess air is

$$C_8H_{18} + 25[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + 12.5\,O_2 + 94N_2$$



The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol	$\bar{h}_{530K}$ kJ/kmol
C_8H_{18} (g)	-208,450	---	---
O_2	0	8682	15,708
N_2	0	8669	15,469
H_2O (g)	-241,820	9904	17,889
CO_2	-393,520	9364	19,029

Substituting,

$$\begin{aligned}-\bar{Q}_{out} &= (8)(-393{,}520 + 19{,}029 - 9364) + (9)(-241{,}820 + 17{,}889 - 9904) + (12.5)(0 + 15{,}708 - 8682)\\ &\quad + (94)(0 + 15{,}469 - 8669) - (1)(-208{,}450) - 0 - 0\\ &= -4{,}239{,}880 \text{ kJ/kmol } C_8H_{18}\end{aligned}$$

or $\bar{Q}_{out} = 4{,}239{,}880 \text{ kJ/kmol } C_8H_{18}$

Then the heat transfer per kg of fuel is

$$Q_{out} = \frac{\bar{Q}_{out}}{M_{fuel}} = \frac{4{,}239{,}880 \text{ kJ/kmol fuel}}{114 \text{ kg/kmol}} = \mathbf{37{,}200 \text{ kJ/kg } C_8H_{18}}$$

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15-59 Propane is burned with 50 percent excess air during a steady-flow combustion process. The rate of heat transfer in the combustion chamber is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

Properties The molar masses of propane and air are 44 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion equation can be written as

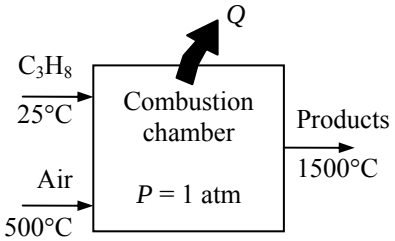
$$C_3H_8 + 1.5a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 0.5a_{th}O_2 + 1.5a_{th} \times 3.76N_2$$

The stoichiometric coefficient is obtained from O_2 balance:

$$1.5a_{th} = 3 + 2 + 0.5a_{th} \longrightarrow a_{th} = 5$$

Substituting,

$$C_3H_8 + 7.5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 2.5O_2 + 28.2N_2$$



The specific volume of the air entering the system is

$$\nu_{air} = \frac{R_{air}T}{P} = \frac{(0.287\ \text{kJ/kmol} \cdot \text{K})(773\ \text{K})}{101.3\ \text{kPa}} = 2.190\ \text{m}^3/\text{kg}$$

and the mass flow rate of this air is

$$\dot{m}_{air} = \frac{\dot{V}_{air}}{\nu_{air}} = \frac{1\ \text{m}^3/\text{s}}{2.190\ \text{m}^3/\text{kg}} = 0.4566\ \text{kg/s}$$

The air-fuel ratio for this combustion process is

$$\text{AF} = \frac{m_{air}}{m_{fuel}} = \frac{(7.5 \times 4.76\ \text{kmol})(29\ \text{kg/kmol})}{(3\ \text{kmol})(12\ \text{kg/kmol}) + (4\ \text{kmol})(2\ \text{kg/kmol})} = 23.53\ \text{kg air/kg fuel}$$

The mass flow rate of fuel is

$$\dot{m}_{fuel} = \frac{\dot{m}_{air}}{\text{AF}} = \frac{0.4566\ \text{kg/s}}{23.53} = 0.01941\ \text{kg/s}$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_P - \sum N_R(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol	$\bar{h}_{773\ K}$ kJ/kmol	$\bar{h}_{1773\ K}$ kJ/kmol
C_3H_8	-103,850	---	---	---
O_2	0	8682	23,614	59,364
N_2	0	8669	22,866	56,689
H_2O (g)	-241,820	9904	---	71,177
CO_2	-393,520	9364	---	87,195

Substituting,

$$\begin{aligned}-Q_{out} &= (3)(-393{,}520 + 87{,}195 - 9364) + (4)(-241{,}820 + 71{,}177 - 9904) + (2.5)(0 + 59{,}364 - 8682) \\ &\quad + (28.2)(0 + 56{,}689 - 8669) - (1)(-103{,}850 + h_{298} - h_{298}) - (7.5)(0 + 23{,}614 - 8682) \\ &\quad - (28.2)(0 + 22{,}866 - 8669) \\ &= -596{,}881\ \text{kJ/kmol}\ C_3H_8\end{aligned}$$

or $\quad Q_{out} = 596{,}881\ \text{kJ/kmol}\ C_3H_8$

Then the rate of heat transfer for a mass flow rate of 0.01941 kg/s for the propane becomes

$$\dot{Q}_{out} = \dot{N}Q_{out} = \left(\frac{\dot{m}}{M}\right)Q_{out} = \left(\frac{0.01941\ \text{kg/s}}{44\ \text{kg/kmol}}\right)(596{,}881\ \text{kJ/kmol}) = \mathbf{263.3\ kW}$$

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15-60 Methane is burned completely during a steady-flow combustion process. The heat transfer from the combustion chamber is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

Analysis The fuel is burned completely with the stoichiometric amount of air, and thus the products will contain only H_2O, CO_2 and N_2, but no free O_2. Considering 1 kmol of fuel, the theoretical combustion equation can be written as

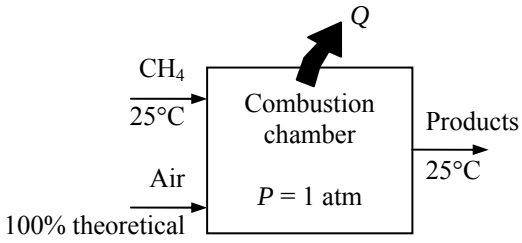
$$CH_4 + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$

where a_{th} is determined from the O_2 balance,

$$a_{th} = 1 + 1 = 2$$

Substituting,

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 5.64N_2$$



The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_P - \sum N_R(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_R = \sum N_P \bar{h}_{f,P}^\circ - \sum N_R \bar{h}_{f,R}^\circ$$

since both the reactants and the products are at 25°C and both the air and the combustion gases can be treated as ideal gases. From the tables,

Substance	$\bar{h}_f^\circ$ **kJ/kmol**
CH_4	-74,850
O_2	0
N_2	0
H_2O (ℓ)	-285,830
CO_2	-393,520

Thus,

$$-Q_{out} = (1)(-393{,}520) + (2)(-285{,}830) + 0 - (1)(-74{,}850) - 0 - 0 = -890{,}330 \text{ kJ / kmol } CH_4$$

or

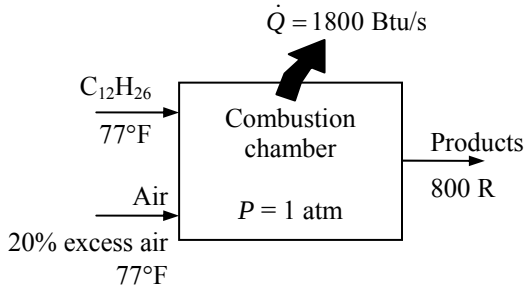
$$Q_{out} = \mathbf{890{,}330\ kJ / kmol\ CH_4}$$

If combustion is achieved with 100% excess air, the answer would still be the same since it would enter and leave at 25°C, and absorb no energy.

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15-61E Diesel fuel is burned with 20 percent excess air during a steady-flow combustion process. The required mass flow rate of the diesel fuel for a specified heat transfer rate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.



Analysis The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol of $C_{12}H_{26}$, the combustion equation can be written as

$$C_{12}H_{26} + 1.2a_{th}(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 0.2a_{th}O_2 + (1.2)(3.76a_{th})N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.2a_{th} = 12 + 6.5 + 0.2a_{th} \quad \longrightarrow \quad a_{th} = 18.5$$

Substituting,

$$C_{12}H_{26} + 22.2(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 3.7O_2 + 83.47N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\bar{h}_{f,R}^\circ$$

since all of the reactants are at 77°F. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ Btu/lbmol	$\bar{h}_{537\ R}$ Btu/lbmol	$\bar{h}_{800\ R}$ Btu/lbmol
$C_{12}H_{26}$	-125,190	---	---
O_2	0	3725.1	5602.0
N_2	0	3729.5	5564.4
H_2O (g)	-104,040	4258.0	6396.9
CO_2	-169,300	4027.5	6552.9

Thus,

$$\begin{aligned}-Q_{out} &= (12)(-169{,}300 + 6552.9 - 4027.5) + (13)(-104{,}040 + 6396.9 - 4258) \\ &\quad + (3.7)(0 + 5602.0 - 3725.1) + (83.47)(0 + 5564.4 - 3729.5) - (1)(-125{,}190) - 0 - 0 \\ &= -3{,}040{,}716 \text{ Btu/lbmol } C_{12}H_{26}\end{aligned}$$

or $$Q_{out} = 3{,}040{,}716 \text{ Btu/lbmol } C_{12}H_{26}$$

Then the required mass flow rate of fuel for a heat transfer rate of 1800 Btu/s becomes

$$\dot{m} = \dot{N}M = \left(\frac{\dot{Q}}{Q}\right)M = \left(\frac{1800 \text{ Btu/s}}{3{,}040{,}716 \text{ Btu/lbmol}}\right)(170 \text{ lbm/lbmol}) = \mathbf{0.1006\ lbm/s}$$

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15-62 A certain coal is burned steadily with 40% excess air. The heat transfer for a given product temperature is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, SO_2, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, N_2, O_2, S, and air are 12, 2, 28, 32, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kg of coal for simplicity. Noting that the mass percentages in this case correspond to the masses of the constituents, the mole numbers of the constituent of the coal are determined to be

$$N_{\text{C}} = \frac{m_{\text{C}}}{M_{\text{C}}} = \frac{39.25\,\text{kg}}{12\,\text{kg/kmol}} = 3.271\,\text{kmol}$$

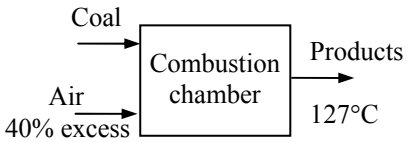
$$N_{\text{H2}} = \frac{m_{\text{H2}}}{M_{\text{H2}}} = \frac{6.93\,\text{kg}}{2\,\text{kg/kmol}} = 3.465\,\text{kmol}$$

$$N_{\text{O2}} = \frac{m_{\text{O2}}}{M_{\text{O2}}} = \frac{41.11\,\text{kg}}{32\,\text{kg/kmol}} = 1.285\,\text{kmol}$$

$$N_{\text{N2}} = \frac{m_{\text{N2}}}{M_{\text{N2}}} = \frac{0.72\,\text{kg}}{28\,\text{kg/kmol}} = 0.0257\,\text{kmol}$$

$$N_{\text{S}} = \frac{m_{\text{S}}}{M_{\text{S}}} = \frac{0.79\,\text{kg}}{32\,\text{kg/kmol}} = 0.0247\,\text{kmol}$$

39.25% C
6.93% H_2
41.11% O_2
0.72% N_2
0.79% S
11.20% ash
(by mass)



The mole number of the mixture and the mole fractions are

$$N_m = 3.271 + 3.465 + 1.285 + 0.0257 + 0.0247 = 8.071\,\text{kmol}$$

$$y_{\text{C}} = \frac{N_{\text{C}}}{N_m} = \frac{3.271\,\text{kmol}}{8.071\,\text{kmol}} = 0.4052$$

$$y_{\text{H2}} = \frac{N_{\text{H2}}}{N_m} = \frac{3.465\,\text{kmol}}{8.071\,\text{kmol}} = 0.4293$$

$$y_{\text{O2}} = \frac{N_{\text{O2}}}{N_m} = \frac{1.285\,\text{kmol}}{8.071\,\text{kmol}} = 0.1592$$

$$y_{\text{N2}} = \frac{N_{\text{N2}}}{N_m} = \frac{0.0257\,\text{kmol}}{8.071\,\text{kmol}} = 0.00319$$

$$y_{\text{S}} = \frac{N_{\text{S}}}{N_m} = \frac{0.0247\,\text{kmol}}{8.071\,\text{kmol}} = 0.00306$$

Ash consists of the non-combustible matter in coal. Therefore, the mass of ash content that enters the combustion chamber is equal to the mass content that leaves. Disregarding this non-reacting component for simplicity, the combustion equation may be written as

$$0.4052\text{C} + 0.4293\text{H}_2 + 0.1592\text{O}_2 + 0.00319\text{N}_2 + 0.00306\text{S} + 1.4a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.4052\text{CO}_2 + 0.4293\text{H}_2\text{O} + 0.4a_{\text{th}}\text{O}_2 + 0.00306\text{SO}_2 + 1.4a_{\text{th}} \times 3.76\text{N}_2$$

According to the O_2 mass balance,

$$0.1592 + 1.4a_{\text{th}} = 0.4052 + 0.5 \times 0.4293 + 0.4a_{\text{th}} + 0.00306 \longrightarrow a_{\text{th}} = 0.4637$$

Substituting,

$$0.4052\text{C} + 0.4293\text{H}_2 + 0.1592\text{O}_2 + 0.00319\text{N}_2 + 0.00306\text{S} + 0.6492(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.4052\text{CO}_2 + 0.4293\text{H}_2\text{O} + 0.1855\text{O}_2 + 0.00306\text{SO}_2 + 2.441\text{N}_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{\text{out}} = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

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Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ \mathrm{K}}$ kJ/kmol	$\bar{h}_{400\ \mathrm{K}}$ kJ/kmol
O_2	0	8682	11,711
N_2	0	8669	11,640
H_2O (g)	-241,820	9904	13,356
CO_2	-393,520	9364	13,372
SO_2	-297,100	-	-

The enthalpy change of sulfur dioxide between the standard temperature and the product temperature using constant specific heat assumption is

$$\Delta\bar{h}_{SO2} = c_p \Delta T = (41.7\ \mathrm{kJ/kmol \cdot K})(127 - 25)\mathrm{K} = 4253\ \mathrm{kJ/kmol}$$

Substituting into the energy balance relation,

$$\begin{aligned} -\bar{Q}_{\text{out}} &= (0.4052)(-393{,}520 + 13{,}372 - 9364) + (0.4293)(-241{,}820 + 13{,}356 - 9904) \\ &\quad + (0.1855)(0 + 11{,}711 - 8682) + (2.441)(0 + 11{,}640 - 8669) + (0.00306)(-297{,}100 + 4253) - 0 \\ &= -253{,}244\ \mathrm{kJ/kmol\ C_8H_{18}} \end{aligned}$$

or $\bar{Q}_{\text{out}} = 253{,}244\ \mathrm{kJ/kmol\ fuel}$

Then the heat transfer per kg of fuel is

$$\begin{aligned} Q_{\text{out}} &= \frac{\bar{Q}_{\text{out}}}{M_{\text{fuel}}} = \frac{253{,}244\ \mathrm{kJ/kmol\ fuel}}{(0.4052 \times 12 + 0.4293 \times 2 + 0.1592 \times 32 + 0.00319 \times 28 + 0.00306 \times 32)\ \mathrm{kg/kmol}} \\ &= \frac{253{,}244\ \mathrm{kJ/kmol\ fuel}}{11.00\ \mathrm{kg/kmol}} \\ &= \mathbf{23{,}020\ kJ/kg\ coal} \end{aligned}$$

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15-63 Octane gas is burned with 30 percent excess air during a steady-flow combustion process. The heat transfer per unit mass of octane is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

Properties The molar mass of C_8H_{18} is 114 kg/kmol (Table A-1).

Analysis The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. The moisture in the air does not react with anything; it simply shows up as additional H_2O in the products. Therefore, for simplicity, we will balance the combustion equation using dry air, and then add the moisture to both sides of the equation. Considering 1 kmol of C_8H_{18}, the combustion equation can be written as

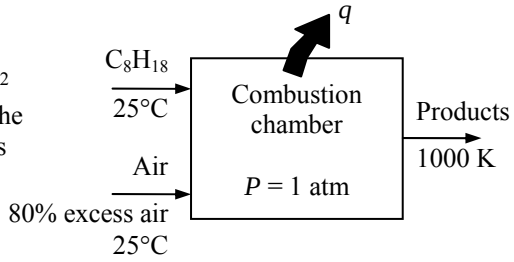
$$C_8H_{18}(g) + 1.8a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 0.8a_{th}O_2 + (1.8)(3.76a_{th})N_2$$

where a_{th} is the stoichiometric coefficient for air. It is determined from

$$O_2 \text{ balance}: 1.8a_{th} = 8 + 4.5 + 0.8a_{th} \longrightarrow a_{th} = 12.5$$

Thus,

$$C_8H_{18}(g) + 22.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 10O_2 + 84.6N_2$$



Therefore, 22.5 × 4.76 = 107.1 kmol of dry air will be used per kmol of the fuel. The partial pressure of the water vapor present in the incoming air is

$$P_{v,\text{in}} = \phi_{\text{air}} P_{\text{sat@25°C}} = (0.40)(3.1698 \text{ kPa}) = 1.268 \text{ kPa}$$

Assuming ideal gas behavior, the number of moles of the moisture that accompanies 107.1 kmol of incoming dry air is determined to be

$$N_{v,\text{in}} = \left(\frac{P_{v,\text{in}}}{P_{\text{total}}}\right) N_{\text{total}} = \left(\frac{1.268 \text{ kPa}}{101.325 \text{ kPa}}\right)(107.1 + N_{v,\text{in}}) \longrightarrow N_{v,\text{in}} = 1.36 \text{ kmol}$$

The balanced combustion equation is obtained by adding 1.36 kmol of H_2O to both sides of the equation,

$$C_8H_{18}(g) + 22.5(O_2 + 3.76N_2) + 1.36H_2O \longrightarrow 8CO_2 + 10.36H_2O + 10O_2 + 84.6N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{\text{out}} = \sum N_P (\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_P - \sum N_R (\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_R = \sum N_P (\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_P - \sum N_R \bar{h}_{f,R}^\circ$$

since all of the reactants are at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$, kJ/kmol	$\bar{h}_{298\text{ K}}$, kJ/kmol	$\bar{h}_{1000\text{ K}}$, kJ/kmol
C_8H_{18} (g)	-208,450	---	---
O_2	0	8682	31,389
N_2	0	8669	30,129
H_2O (g)	-241,820	9904	35,882
CO_2	-393,520	9364	42,769

Substituting,

$$\begin{aligned} -Q_{\text{out}} &= (8)(-393{,}520 + 42{,}769 - 9364) + (10.36)(-241{,}820 + 35{,}882 - 9904) \\ &\quad + (10)(0 + 31{,}389 - 8682) + (84.6)(0 + 30{,}129 - 8669) \\ &\quad - (1)(-208{,}450) - (1.36)(-241{,}820) - 0 - 0 \\ &= -2{,}537{,}130 \text{ kJ/kmol } C_8H_{18} \end{aligned}$$

Thus 2,537,130 kJ of heat is transferred from the combustion chamber for each kmol (114 kg) of C_8H_{18}. Then the heat transfer per kg of C_8H_{18} becomes

$$q = \frac{Q_{\text{out}}}{M} = \frac{2{,}537{,}130 \text{ kJ}}{114 \text{kg}} = \mathbf{22{,}260 \text{ kJ/kg } C_8H_{18}}$$

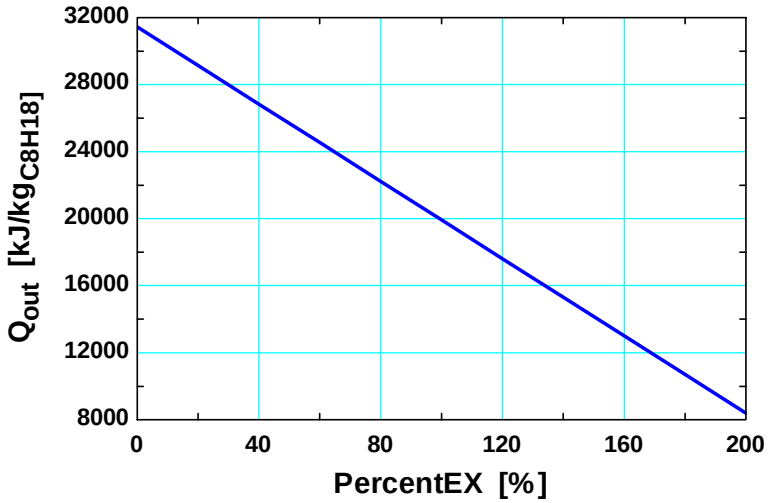
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15-64 Problem 15-63 is reconsidered. The effect of the amount of excess air on the heat transfer for the combustion process is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
Fuel$ = 'Octane (C8H18)'
T_fuel = (25+273) "[K]"
{PercentEX = 80 "[%]"}
Ex = PercentEX/100 "[%Excess air/100]"
P_air1 = 101.3 [kPa]
T_air1 = 25+273 "[K]"
RH_1 = 40/100  "[%]"
T_prod = 1000 [K]
M_air = 28.97 [kg/kmol]
M_water = 18 [kg/kmol]
M_C8H18=(8*12+18*1) "[kg/kmol]"
"For  theoretical dry air, the complete combustion equation is"
"C8H18 + A_th(O2+3.76 N2)=8 CO2+9 H2O +  A_th (3.76) N2 "
A_th*2=8*2+9*1 "theoretical O balance"
"now to find the amount of water vapor associated with the dry air"
w_1=HUMRAT(AirH2O,T=T_air1,P=P_air1,R=RH_1) "Humidity ratio, kgv/kga"
N_w=w_1*(A_th*4.76*M_air)/M_water "Moles of water in the atmoshperic air, kmol/kmol_fuel"
"The balanced combustion equation with Ex% excess moist air is"
"C8H18 + (1+EX)[A_th(O2+3.76 N2)+N_w H2O]=8 CO2+(9+(1+Ex)*N_w) H2O + (1+Ex) A_th (3.76) N2+ Ex(
A_th) O2 "
"Apply First Law SSSF"
H_fuel = -208450 [kJ/kmol] "from Table A-26"
HR=H_fuel+ (1+Ex)*A_th*enthalpy(O2,T=T_air1)+(1+Ex)*A_th*3.76
*enthalpy(N2,T=T_air1)+(1+Ex)*N_w*enthalpy(H2O,T=T_air1)
HP=8*enthalpy(CO2,T=T_prod)+(9+(1+Ex)*N_w)*enthalpy(H2O,T=T_prod)+(1+Ex)*A_th*3.76*
enthalpy(N2,T=T_prod)+Ex*A_th*enthalpy(O2,T=T_prod)
Q_net=(HP-HR)"kJ/kmol"/(M_C8H18 "kg/kmol")   "[kJ/kg_C8H18]"
Q_out = -Q_net "[kJ/kg_C8H18]"
"This solution used the humidity ratio form psychrometric data to determine the moles of water vapor in
atomspheric air.  One should calculate the moles of water contained in the atmospheric air by the method shown
in Chapter 14 which uses the relative humidity to find the partial pressure of the water vapor and, thus, the moles
of water vapor. Explore what happens to the results as you vary the percent excess air, relative humidity, and
product temperature."
```

PercentEX [%]	Q_{out} [kJ/kgC8H18]
0	31444
20	29139
40	26834
60	24529
80	22224
100	19919
120	17614
140	15309
160	13003
180	10698
200	8393



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15-65 Propane gas is burned with 100% excess air. The combustion is incomplete. The balanced chemical reaction is to be written, and the dew-point temperature of the products and the heat transfer from the combustion chamber are to be determined.

Assumptions **1** Combustion is incomplete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

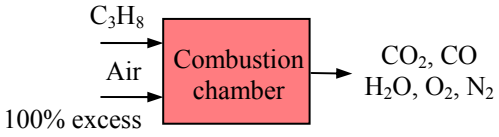
Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis (a) The balanced reaction equation for stoichiometric air is

$$C_3H_8 + a_{th}[O_2 + 3.76N_2] \longrightarrow 3\,CO_2 + 4\,H_2O + a_{th} \times 3.76\,N_2$$

The stoicihiometric coefficient a_{th} is determined from an O_2 balance:

$$a_{th} = 3 + 2 = 5$$



Substituting,

$$C_3H_8 + 5[O_2 + 3.76N_2] \longrightarrow 3\,CO_2 + 4\,H_2O + 5 \times 3.76\,N_2$$

The reaction with 100% excess air and incomplete combustion can be written as

$$C_3H_8 + 2 \times 5[O_2 + 3.76N_2] \longrightarrow 0.90 \times 3\,CO_2 + 0.10 \times 3\,CO + 4\,H_2O + x\,O_2 + 2 \times 5 \times 3.76\,N_2$$

The coefficient for O_2 is determined from a mass balance,

O_2 balance: $\quad 10 = 0.9 \times 3 + 0.05 \times 3 + 2 + x \longrightarrow x = 5.15$

Substituting,

$$C_3H_8 + 10[O_2 + 3.76N_2] \longrightarrow 2.7\,CO_2 + 0.3\,CO + 4\,H_2O + 5.15\,O_2 + 37.6\,N_2$$

(b) The partial pressure of water vapor is

$$P_v = \frac{N_{H2O}}{N_{total}} P_{total} = \frac{4}{2.7 + 0.3 + 4 + 5.15 + 37.6}(100\text{ kPa}) = \frac{4\text{ kmol}}{49.75\text{ kmol}}(100\text{ kPa}) = 8.040\text{ kPa}$$

The dew point temperature of the product gases is the saturation temperature of water at this pressure:

$$T_{dp} = T_{sat@8.04\,kPa} = \mathbf{41.5°C} \quad \text{(Table A-5)}$$

(c) The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Both the reactants and products are at 25 °C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. Also, since the temperature of products (25 °C) is lower than the dew-po,nt temperature, some water vapor will condense. Noting from Table A-5 that $P_{sat@25°C} = 3.17$ kPa , the molar amount of water that remain as vapor is determined from

$$P_v = \frac{N_{H2O}}{N_{total}} P_{total} \longrightarrow 3.17\text{ kPa} = \frac{N_{H2O,vapor}}{N_{H2O,vapor} + 2.7 + 0.3 + 5.15 + 37.6}(100\text{ kPa}) \longrightarrow N_{H2O,vapor} = 1.5\text{ kmol}$$

Thus, $N_{H2O,liquid} = 4 - 1.5 = 2.5$ kmol

Then, using the values given in the table,

$$\begin{aligned} -Q_{out} &= (2.7)(-393{,}520) + (0.3)(-110{,}530) + (1.5)(-241{,}820) + (2.5)(-285{,}830) - (1)(-103{,}850) \\ &= -2{,}069{,}120\text{ kJ/kmol } C_3H_8 \end{aligned}$$

or $\quad Q_{out} = 2{,}069{,}120\text{ kJ/kmol } C_3H_8$

Then the heat transfer for a 100 kmol fuel becomes

$$Q_{out} = NQ_{out} = \left(\frac{\dot{m}}{N}\right)Q_{out} = (100\text{ kmol fuel})(2{,}069{,}120\text{ kJ/kmol fuel}) = \mathbf{2.069 \times 10^8\ kJ}$$

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15-66 A mixture of propane and methane is burned with theoretical air. The balanced chemical reaction is to be written, and the amount of water vapor condensed and the the required air flow rate for a given heat transfer rate are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

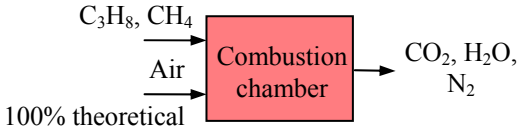
Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis (a) The balanced reaction equation for stoichiometric air is

$$0.4\,C_3H_8 + 0.6\,CH_4 + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 1.8\,CO_2 + 2.8\,H_2O + a_{th} \times 3.76\,N_2$$

The stoicihiometric coefficient a_{th} is determined from an O_2 balance:

$$a_{th} = 1.8 + 1.4 = 3.2$$



Substituting,

$$0.4\,C_3H_8 + 0.6\,CH_4 + 3.2\left[O_2 + 3.76N_2\right] \longrightarrow 1.8\,CO_2 + 2.8\,H_2O + 12.032\,N_2$$

(b) The partial pressure of water vapor is

$$P_v = \frac{N_{H2O}}{N_{total}} P_{total} = \frac{2.8}{1.8 + 2.8 + 12.032}(100\text{ kPa}) = \frac{2.8\text{ kmol}}{16.632\text{ kmol}}(100\text{ kPa}) = 16.84\text{ kPa}$$

The dew point temperature of the product gases is the saturation temperature of water at this pressure:

$$T_{dp} = T_{sat@16.84\,kPa} = 56.2°C \quad \text{(Table A-5)}$$

Since the temperature of the product gases are at 398 K (125°C), there will be no condensation of water vapor.

(c) The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

The products are at 125 °C, and the enthalpy of products can be expressed as

$$\left(\bar{h} - \bar{h}^\circ\right) = \bar{c}_p \Delta T$$

where $\Delta T = 125 - 25 = 100°C = 100\text{ K}$. Then, using the values given in the table,

$$\begin{aligned} -Q_{out} &= (1.8)(-393{,}520 + 41.16 \times 100) + (2.8)(-241{,}820 + 34.28 \times 100) + (12.032)(0 + 29.27 \times 100) \\ &\quad - (0.4)(-103{,}850) - (0.6)(-74{,}850) \\ &= -1{,}246{,}760\text{ kJ/kmol fuel} \end{aligned}$$

or $$Q_{out} = 1{,}246{,}760\text{ kJ/kmol fuel}$$

For a heat transfer rate of 97,000 kJ/h, the molar flow rate of fuel is

$$\dot{N}_{fuel} = \frac{\dot{Q}_{out}}{Q_{out}} = \frac{97{,}000\text{ kJ/h}}{1{,}246{,}760\text{ kJ/kmol fuel}} = 0.07780\text{ kmol fuel/h}$$

The molar mass of the fuel mixture is

$$M_{fuel} = 0.4 \times 44 + 0.6 \times 16 = 27.2\text{ kg/kmol}$$

The mass flow rate of fuel is

$$\dot{m}_{fuel} = \dot{N}_{fuel} M_{fuel} = (0.07780\text{ kmol/h})(27.2\text{ kg/kmol}) = 2.116\text{ kg/h}$$

The air-fuel ratio is

$$\text{AF} = \frac{m_{air}}{m_{fuel}} = \frac{(3.2 \times 4.76 \times 29)\text{ kg}}{(0.4 \times 44 + 0.6 \times 16)\text{ kg}} = 16.24\text{ kg air/kg fuel}$$

The mass flow rate of air is then

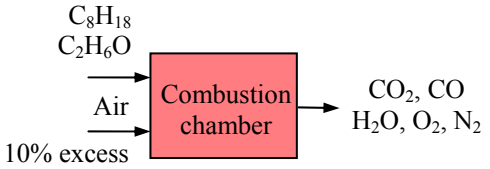
$$\dot{m}_{air} = \dot{m}_{fuel}\text{AF} = (2.116\text{ kg/h})16.24) = \mathbf{34.4\text{ kg/h}}$$

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15-67 A mixture of ethanol and octane is burned with 10% excess air. The combustion is incomplete. The balanced chemical reaction is to be written, and the dew-point temperature of the products, the heat transfer for the process, and the relative humidity of atmospheric air for specified conditions are to be determined.

Assumptions **1** Combustion is incomplete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).



Analysis (a) The balanced reaction equation for stoichiometric air is

$$0.1\,C_2H_6O + 0.9\,C_8H_{18} + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 7.4\,CO_2 + 8.4\,H_2O + a_{th} \times 3.76\,N_2$$

The stoicihiometric coefficient a_{th} is determined from an O_2 balance:

$$0.1/2 + a_{th} = 7.4 + 8.4/2 \longrightarrow a_{th} = 11.55$$

Substituting,

$$0.1\,C_2H_6O + 0.9\,C_8H_{18} + 11.55\left[O_2 + 3.76N_2\right] \longrightarrow 7.4\,CO_2 + 8.4\,H_2O + 11.55 \times 3.76\,N_2$$

The reaction with 10% excess air and incomplete combustion can be written as

$$0.1\,C_2H_6O + 0.9\,C_8H_{18} + 1.1 \times 11.55\left[O_2 + 3.76N_2\right]$$
$$\longrightarrow 0.9 \times 7.4\,CO_2 + 0.1 \times 7.4\,CO + 8.4\,H_2O + x\,O_2 + 1.1 \times 11.55 \times 3.76\,N_2$$

The coefficient for O_2 is determined from a mass balance,

O_2 balance: $$0.5 \times 0.1 + 1.1 \times 11.55 = 0.9 \times 7.4 + 0.5 \times (0.1 \times 7.4) + 0.5 \times 8.4 + x \longrightarrow x = 1.525$$

Substituting,

$$0.1\,C_2H_6O + 0.9\,C_8H_{18} + 12.705\left[O_2 + 3.76N_2\right]$$
$$\longrightarrow 6.66\,CO_2 + 0.74\,CO + 8.4\,H_2O + 1.525\,O_2 + 47.77\,N_2$$

(b) The partial pressure of water vapor is

$$P_v = \frac{N_{H2O}}{N_{total}} P_{total} = \frac{8.4}{6.66 + 0.74 + 8.4 + 1.525 + 47.77}(100\text{ kPa}) = \frac{8.4\text{ kmol}}{65.10\text{ kmol}}(100\text{ kPa}) = 12.9\text{ kPa}$$

The dew point temperature of the product gases is the saturation temperature of water at this pressure:

$$T_{dp} = T_{sat@12.9\,kPa} = \mathbf{50.5°C} \quad \text{(Table A-5)}$$

(c) The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Both the reactants and products are at 25 °C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. Then, using the values given in the table,

$$-Q_{out} = (6.66)(-393{,}520) + (0.74)(-110{,}530) + (8.4)(-241{,}820) - (0.1)(-235{,}310) - (0.9)(-208{,}450)$$
$$= -4{,}522{,}790\text{ kJ/kmol fuel}$$

or $$Q_{out} = 4{,}522{,}790\text{ kJ/kmol fuel}$$

The molar mass of the fuel is

$$M = 0.1 \times 46 + 0.9 \times 114 = 107.2\text{ kg/kmol}$$

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Then the heat transfer for a 2.5 kg of fuel becomes

$$Q_{\text{out}} = NQ_{\text{out}} = \left(\frac{m}{M}\right)Q_{\text{out}} = \frac{2.5\text{ kg}}{107.2\text{ kg/kmol}}(4{,}522{,}790\text{ kJmol}) = \mathbf{105{,}480\text{ kJ}}$$

(d) For 9.57 kmol of water vapor in the products, the air must carry 9.57 − 8.4 = 1.17 kmol of water vapor in the atmospheric air. The partial pressure of this water vapor in the stmospheric air is

$$P_v = \frac{N_{\text{H2O}}}{N_{\text{total}}}P_{\text{total}} = \frac{1.17}{12.705 \times 4.76 + 1.17}(100\text{ kPa}) = \frac{1.17\text{ kmol}}{61.65\text{ kmol}}(100\text{ kPa}) = 1.8979\text{ kPa}$$

The saturation pressure of water at 25°C is 3.17 kPa (Table A-4). The relative humidity of water vapor in the atmospheric air is then

$$\phi = \frac{P_v}{P_{\text{total}}} = \frac{1.8979\text{ kPa}}{3.17\text{ kPa}} = 0.599 = \mathbf{59.9\%}$$

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15-68 A mixture of methane and oxygen contained in a tank is burned at constant volume. The final pressure in the tank and the heat transfer during this process are to be determined.

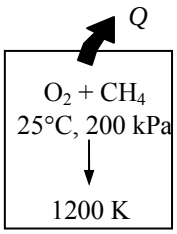
Assumptions **1** Air and combustion gases are ideal gases. **2** Combustion is complete.

Properties The molar masses of CH_4 and O_2 are 16 kg/kmol and 32 kg/kmol, respectively (Table A-1).

Analysis (*a*) The combustion is assumed to be complete, and thus all the carbon in the methane burns to CO_2 and all of the hydrogen to H_2O. The number of moles of CH_4 and O_2 in the tank are

$$N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{0.12 \text{ kg}}{16 \text{ kg/kmol}} = 7.5\times10^{-3} \text{ kmol} = 7.5 \text{ mol}$$

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{0.6 \text{ kg}}{32 \text{ kg/kmol}} = 18.75\times10^{-3} \text{ kmol} = 18.75 \text{ mol}$$



Then the combustion equation can be written as

$$7.5CH_4 + 18.75O_2 \longrightarrow 7.5CO_2 + 15H_2O + 3.75O_2$$

At 1200 K, water exists in the gas phase. Assuming both the reactants and the products to be ideal gases, the final pressure in the tank is determined to be

$$\left.\begin{aligned} P_R V &= N_R R_u T_R \\ P_P V &= N_P R_u T_P \end{aligned}\right\} P_P = P_R\left(\frac{N_P}{N_R}\right)\left(\frac{T_P}{T_R}\right)$$

Substituting,

$$P_P = (200 \text{ kPa})\left(\frac{26.25 \text{ mol}}{26.25 \text{ mol}}\right)\left(\frac{1200 \text{ K}}{298 \text{ K}}\right) = \mathbf{805 \text{ kPa}}$$

which is relatively low. Therefore, the ideal gas assumption utilized earlier is appropriate.

(*b*) The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and products are assumed to be ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by R_uT. It yields

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h}_{1200\text{ K}} - \bar{h}_{298\text{ K}} - R_uT\right)_P - \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\text{ K}}$ kJ/kmol	$\bar{h}_{1200\text{ K}}$ kJ/kmol
CH_4	-74,850	---	---
O_2	0	8682	38,447
H_2O (*g*)	-241,820	9904	44,380
CO_2	-393,520	9364	53,848

Thus,

$$\begin{aligned} -Q_{out} &= (7.5)(-393{,}520 + 53{,}848 - 9364 - 8.314\times1200) \\ &\quad + (15)(-241{,}820 + 44{,}380 - 9904 - 8.314\times1200) \\ &\quad + (3.75)(0 + 38{,}447 - 8682 - 8.314\times1200) \\ &\quad - (7.5)(-74{,}850 - 8.314\times298) - (18.75)(-8.314\times298) \\ &= -5{,}251{,}791 \text{ J} = -5252 \text{ kJ} \end{aligned}$$

Thus $Q_{out} = \mathbf{5252 \text{ kJ}}$ of heat is transferred from the combustion chamber as 120 g of CH_4 burned in this combustion chamber.

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15-69 Problem 15-68 is reconsidered. The effect of the final temperature on the final pressure and the heat transfer for the combustion process is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
T_reac = (25+273) "[K]"                "reactant mixture temperature"
P_reac = 200 [kPa]                     "reactant mixture pressure"
{T_prod = 1200 [K]}                    "product mixture temperature"
m_O2=0.600 [kg]                        "initial mass of O2"
Mw_O2 = 32 [kg/kmol]
m_CH4 = 0.120 [kg]                     "initial mass of CH4"
Mw_CH4=(1*12+4*1) "[kg/kmol]"
R_u = 8.314 [kJ/kmol-K]                "universal gas constant"

"For  theoretical oxygen, the complete combustion equation is"
"CH4 + A_th O2=1 CO2+2 H2O  "
2*A_th=1*2+2*1"theoretical O balance"

"now to find the actual moles of O2 supplied  per mole of fuel"
N_O2 = m_O2/Mw_O2/N_CH4
N_CH4= m_CH4/Mw_CH4

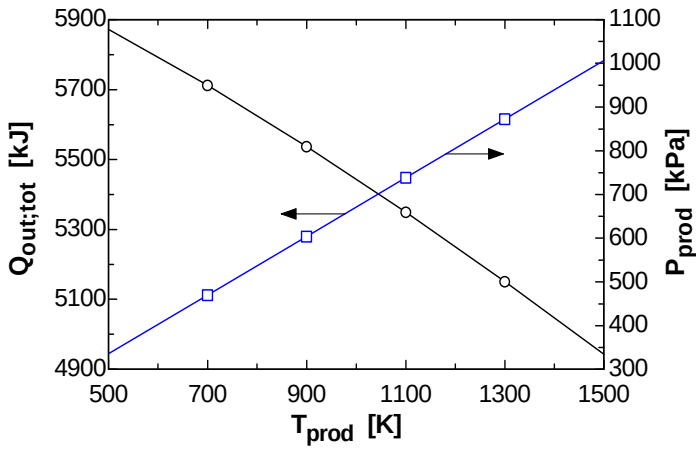
"The balanced complete combustion equation with Ex% excess O2 is"
"CH4 + (1+EX) A_th O2=1 CO2+ 2 H2O + Ex( A_th) O2 "
N_O2 = (1+Ex)*A_th

"Apply First Law to the closed system combustion chamber and assume ideal gas
behavior. (At 1200 K, water exists in the gas phase.)"
E_in - E_out = DELTAE_sys
E_in = 0
E_out = Q_out "kJ/kmol_CH4"  "No work is done because volume is constant"
DELTAE_sys = U_prod - U_reac "neglect KE and PE and note: U = H - PV = N(h - R_u T)"
U_reac = 1*(enthalpy(CH4, T=T_reac) - R_u*T_reac) +(1+EX)*A_th*(enthalpy(O2,T=T_reac) - R_u*T_reac)
U_prod = 1*(enthalpy(CO2, T=T_prod) - R_u*T_prod) +2*(enthalpy(H2O, T=T_prod) -
R_u*T_prod)+EX*A_th*(enthalpy(O2,T=T_prod) - R_u*T_prod)

"The total heat transfer out, in kJ, is:"
Q_out_tot=Q_out"kJ/kmol_CH4"/(Mw_CH4 "kg/kmol_CH4") *m_CH4"kg"  "kJ"

"The final pressure in the tank is the pressure of the product gases.  Assuming
ideal gas behavior for the gases in the constant volume tank, the ideal gas law gives:"
P_reac*V =N_reac * R_u *T_reac
P_prod*V = N_prod * R_u * T_prod
N_reac = N_CH4*(1 + N_O2)
N_prod = N_CH4*(1 + 2 + Ex*A_th)
```

T_{prod} [K]	$Q_{out,tot}$ [kJ]	P_{prod} [kPa]
500	5872	335.6
700	5712	469.8
900	5537	604
1100	5349	738.3
1300	5151	872.5
1500	4943	1007



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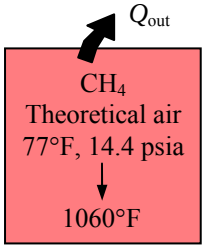
15-70E Methane is burned with stoichiometric amount of air in a rigid container. The heat rejected from the container is to be determined.

Assumptions **1** Air and combustion gases are ideal gases. **2** Combustion is complete.

Properties The molar masses of CH_4 and air are 16 lbm/lbmol and 29 lbm/lbmol, respectively (Table A-1E).

Analysis The combustion equation for 1 lbmol of fuel is

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 7.52N_2$$



The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and products are assumed to be ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by R_uT. It yields

$$-Q_{out} = \sum N_P \left(\bar{h}_f^\circ + \bar{h}_{1520\,R} - \bar{h}_{537\,R} - R_uT\right)_P - \sum N_R \left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 77°F. From the tables,

Substance	$\bar{h}_f^\circ$ Btu/lbmol	$\bar{h}_{537R}$ Btu/lbmol	$\bar{h}_{1520R}$ Btu/lbmol
CH_4	-32,210	---	---
O_2	0	3725.1	11,179.6
N_2	0	3729.5	10,800.4
H_2O (g)	-104,040	4258.0	12,738.8
CO_2	-169,300	4027.5	14,824.9

Thus,

$$\begin{aligned}-Q_{out} &= (1)(-169{,}300 + 14{,}824.9 - 4027.5 - 1.9858 \times 1520) \\ &\quad + (2)(-104{,}040 + 12{,}738.8 - 4258.0 - 1.9858 \times 1520) \\ &\quad + (7.52)(0 + 10{,}800.4 - 3729.5 - 1.9858 \times 1520) \\ &\quad - (1)(-32{,}210 - 1.9858 \times 537) - (2)(-1.9858 \times 537) - (7.52)(-1.9858 \times 537) \\ &= -284{,}800 \text{ Btu/lbmol } CH_4\end{aligned}$$

Thus

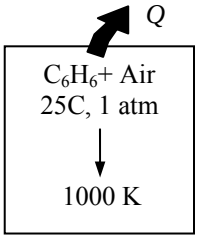
$$Q_{out} = \mathbf{284{,}800\ Btu/lbmol\ CH_4}$$

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15-71 A mixture of benzene gas and 30 percent excess air contained in a constant-volume tank is ignited. The heat transfer from the combustion chamber is to be determined.

Assumptions **1** Both the reactants and products are ideal gases. **2** Combustion is complete.

Analysis The theoretical combustion equation of C_6H_6 with stoichiometric amount of air is



$$C_6H_6(g) + a_{th}(O_2 + 3.76N_2) \longrightarrow 6CO_2 + 3H_2O + 3.76a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$a_{th} = 6 + 1.5 = 7.5$$

Then the actual combustion equation with 30% excess air becomes

$$C_6H_6(g) + 9.75(O_2 + 3.76N_2) \longrightarrow 5.52CO_2 + 0.48CO + 3H_2O + 2.49O_2 + 36.66N_2$$

The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ - P\overline{v}\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ - P\overline{v}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\overline{v}$ terms in this equation can be replaced by R_uT.

It yields

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h}_{1000\ K} - \overline{h}_{298\ K} - R_uT\right)_P - \sum N_R\left(\overline{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

Substance	$\overline{h}_f^\circ$ kJ/kmol	$\overline{h}_{298\ K}$ kJ/kmol	$\overline{h}_{1000\ K}$ kJ/kmol
C_6H_6 (*g*)	82,930	---	---
O_2	0	8682	31,389
N_2	0	8669	30,129
H_2O (*g*)	-241,820	9904	35,882
CO	-110,530	8669	30,355
CO_2	-393,520	9364	42,769

Thus,

$$\begin{aligned}-Q_{out} &= (5.52)(-393{,}520 + 42{,}769 - 9364 - 8.314\times 1000)\\ &\quad + (0.48)(-110{,}530 + 30{,}355 - 8669 - 8.314\times 1000)\\ &\quad + (3)(-241{,}820 + 35{,}882 - 9904 - 8.314\times 1000)\\ &\quad + (2.49)(0 + 31{,}389 - 8682 - 8.314\times 1000)\\ &\quad + (36.66)(0 + 30{,}129 - 8669 - 8.314\times 1000)\\ &\quad - (1)(82{,}930 - 8.314\times 298) - (9.75)(4.76)(-8.314\times 298)\\ &= -2{,}200{,}433\ \text{kJ}\end{aligned}$$

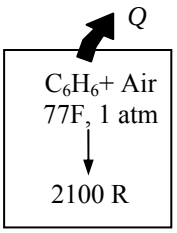
or $Q_{out} = \mathbf{2{,}200{,}433\ kJ}$

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15-72E A mixture of benzene gas and 60 percent excess air contained in a constant-volume tank is ignited. The heat transfer from the combustion chamber is to be determined.

Assumptions **1** Both the reactants and products are ideal gases. **2** Combustion is complete.

Analysis The theoretical combustion equation of C_6H_6 with stoichiometric amount of air is



$$C_6H_6(g) + a_{th}(O_2 + 3.76N_2) \longrightarrow 6CO_2 + 3H_2O + 3.76a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$a_{th} = 6 + 1.5 = 7.5$$

Then the actual combustion equation with 60% excess air becomes

$$C_6H_6(g) + 12(O_2 + 3.76N_2) \longrightarrow 5.52CO_2 + 0.48CO + 3H_2O + 4.74O_2 + 45.12N_2$$

The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with W = 0. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by R_uT.

It yields

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h}_{1800\,R} - \bar{h}_{537\,R} - R_uT\right)_P - \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 77°F. From the tables,

Substance	$\bar{h}_f^\circ$ Btu/lbmol	$\bar{h}_{537\,R}$ Btu/lbmol	$\bar{h}_{2100R}$ Btu/lbmol
C_6H_6 (*g*)	35,680	---	---
O_2	0	3725.1	16,011
N_2	0	3729.5	15,334
H_2O (*g*)	-104,040	4258.0	18,467
CO	-47,540	3725.1	15,463
CO_2	-169,300	4027.5	22,353

Thus,

$$\begin{aligned}-Q_{out} &= (5.52)(-169{,}300 + 22{,}353 - 4027.5 - 1.986 \times 2100)\\ &\quad + (0.48)(-47{,}540 + 15{,}463 - 3725.1 - 1.986 \times 2100)\\ &\quad + (3)(-104{,}040 + 18{,}467 - 4258.0 - 1.986 \times 2100)\\ &\quad + (4.74)(0 + 16{,}011 - 3725.1 - 1.986 \times 2100)\\ &\quad + (45.12)(0 + 15{,}334 - 3729.5 - 1.986 \times 2100)\\ &\quad - (1)(35{,}680 - 1.986 \times 537) - (12)(4.76)(-1.986 \times 537)\\ &= -757{,}400 \text{ Btu}\end{aligned}$$

or $Q_{out} = \mathbf{757{,}400\ Btu}$

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15-73 A high efficiency gas furnace burns gaseous propane C_3H_8 with 140 percent theoretical air. The volume flow rate of water condensed from the product gases is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction equation for 40% excess air (140% theoretical air) is

$$C_3H_8 + 1.4a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + E\,O_2 + F\,N_2$$

where a_{th} is the stoichiometric coefficient for air. We have automatically accounted for the 40% excess air by using the factor $1.4a_{th}$ instead of a_{th} for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

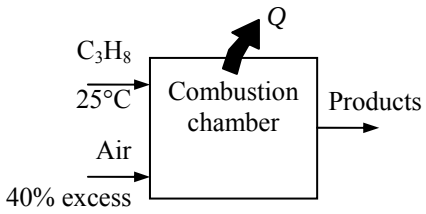
Carbon balance: $B = 3$

Hydrogen balance: $2D = 8 \longrightarrow D = 4$

Oxygen balance: $2 \times 1.4a_{th} = 2B + D + 2E$

$0.4a_{th} = E$

Nitrogen balance: $1.4a_{th} \times 3.76 = F$



Solving the above equations, we find the coefficients ($E = 2$, $F = 26.32$, and $a_{th} = 5$) and write the balanced reaction equation as

$$C_3H_8 + 7\left[O_2 + 3.76N_2\right] \longrightarrow 3\,CO_2 + 4\,H_2O + 2\,O_2 + 26.32\,N_2$$

The partial pressure of water in the saturated product mixture at the dew point is

$$P_{v,\text{prod}} = P_{\text{sat@40°C}} = 7.3851\,\text{kPa}$$

The vapor mole fraction is

$$y_v = \frac{P_{v,\text{prod}}}{P_{\text{prod}}} = \frac{7.3851\,\text{kPa}}{100\,\text{kPa}} = 0.07385$$

The kmoles of water condensed is determined from

$$y_v = \frac{N_{\text{water}}}{N_{\text{total,product}}} \longrightarrow 0.07385 = \frac{4 - N_w}{3 + 4 - N_w + 2 + 26.32} \longrightarrow N_w = 1.503\,\text{kmol}$$

The steady-flow energy balance is expressed as

$$\dot{N}_{\text{fuel}} H_R = \dot{Q}_{\text{fuel}} + \dot{N}_{\text{fuel}} H_P$$

where $\dot{Q}_{\text{fuel}} = \dfrac{\dot{Q}_{\text{out}}}{\eta_{\text{furnace}}} = \dfrac{31{,}650\,\text{kJ/h}}{0.96} = 32{,}969\,\text{kJ/h}$

$$H_R = \bar{h}^o_{f\,\text{fuel@25°C}} + 7\bar{h}_{\text{O2@25°C}} + 26.32\bar{h}_{\text{N2@25°C}}$$
$$= (-103{,}847\,\text{kJ/kmol}) + 7(0) + 26.32(0) = -103{,}847\,\text{kJ/kmol}$$

$$H_P = 3\bar{h}_{\text{CO2@25°C}} + 4\bar{h}_{\text{H2O@25°C}} + 2\bar{h}_{\text{O2@25°C}} + 26.32\bar{h}_{\text{N2@25°C}} + N_w(\bar{h}^o_{f\,\text{H2O(liq)}})$$
$$= 3(-393{,}520\,\text{kJ/kmol}) + 4(\text{-}241{,}820\,\text{kJ/kmol}) + 2(0) + 26.32(0) + 1.503(-285{,}830\,\text{kJ/kmol})$$
$$= -2.577 \times 10^6\ \text{kJ/kmol}$$

Substituting into the energy balance equation,

$$\dot{N}_{\text{fuel}} H_R = \dot{Q}_{\text{fuel}} + \dot{N}_{\text{fuel}} H_P$$

$$\dot{N}_{\text{fuel}}(-103{,}847\,\text{kJ/kmol}) = 32{,}969\,\text{kJ/h} + \dot{N}_{\text{fuel}}(-2.577 \times 10^6\ \text{kJ/kmol}) \longrightarrow \dot{N}_{\text{fuel}} = 0.01333\,\text{kmol/h}$$

The molar and mass flow rates of the liquid water are

$$\dot{N}_w = N_w \dot{N}_{\text{fuel}} = (1.503\ \text{kmol/kmol fuel})(0.01333\ \text{kmol fuel/h}) = 0.02003\ \text{kmol/h}$$

$$\dot{m}_w = \dot{N}_w M_w = (0.02003\ \text{kmol/h})(18\ \text{kg/kmol}) = 0.3608\ \text{kg/h}$$

The volume flow rate of liquid water is

$$\dot{V}_w = (v_{f\,@25°C})\dot{m}_w = (0.001003\ \text{m}^3/\text{kg})(0.3608\ \text{kg/h}) = 0.0003619\ \text{m}^3/\text{h} = \mathbf{8.7\ L/day}$$

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15-74 Wheat straw that is being considered as an alternative fuel is tested in a bomb calorimeter. The heating value of this straw is to be determined and compared to the higher heating value of propane.

Assumptions **1** Combustion is complete.

Analysis The heat released by the combustion is

$$Q = mc_v \Delta T = (100\ \text{kJ/K})(1.8\ \text{K}) = 180\ \text{kJ}$$

The heating value is then

$$\text{HV} = \frac{Q}{m} = \frac{180\ \text{kJ}}{0.010\ \text{kg}} = \mathbf{18{,}000\ kJ/kg}$$

From Table A-27, the higher heating value of propane is

HHV = **50,330 kJ/kg**

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Adiabatic Flame Temperature

15-75C For the case of stoichiometric amount of pure oxygen since we have the same amount of chemical energy released but a smaller amount of mass to absorb it.

15-76C Under the conditions of complete combustion with stoichiometric amount of air.

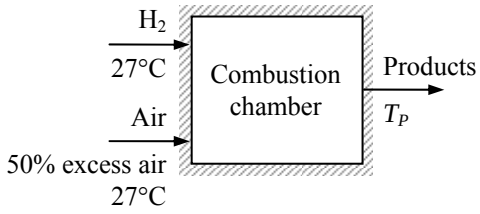
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15-77 Hydrogen is burned with 50 percent excess air during a steady-flow combustion process. The exit temperature of product gases is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

Analysis Adiabatic flame temperature is the temperature at which the products leave the combustion chamber under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$). Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$



The combustion equation of H_2 with 50% excess air is

$$H_2 + 0.75(O_2 + 3.76N_2) \longrightarrow H_2O + 0.25O_2 + 2.82N_2$$

From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{300\text{K}}$ kJ/kmol	$\bar{h}_{298\text{ K}}$ kJ/kmol
H_2	0	8522	8468
O_2	0	8736	8682
N_2	0	8723	8669
H_2O (*g*)	-241,820	9966	9904

Thus,

$$(1)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (0.25)(0 + \bar{h}_{O_2} - 8682) + (2.82)(0 + \bar{h}_{N_2} - 8669)$$
$$= (1)(0 + 8522 - 8468) + (0.75)(0 + 8736 - 8682) + (2.82)(0 + 8723 - 8669)$$

It yields

$$\bar{h}_{H_2O} + 0.25\bar{h}_{O_2} + 2.82\bar{h}_{N_2} = 278{,}590 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 278,590/(1 + 0.25 + 2.82) = 68,450 kJ/kmol. This enthalpy value corresponds to about 2100 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2100 K, but somewhat under it because of the higher specific heat of H_2O.

At 2000 K: $\bar{h}_{H_2O} + 0.25\bar{h}_{O_2} + 2.82\bar{h}_{N_2} = (1)(82{,}593) + (0.25)(67{,}881) + (2.82)(64{,}810)$
$= 282{,}330$ kJ (Higher than 278,590 kJ)

At 1960 K: $\bar{h}_{H_2O} + 0.25\bar{h}_{O_2} + 2.826\bar{h}_{N_2} = (1)(80{,}555) + (0.25)(66{,}374) + (2.82)(63{,}381)$
$= 275{,}880$ kJ (Lower than 278,590 kJ)

By interpolation, T_P = **1977 K**

Discussion The adiabatic flame temperature cam be obtained by using EES without a trial and error approach. We found the temperature to be **1978 K** by EES. The results are practically identical.

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15-78 Problem 15-77 is reconsidered. This problem is to be modified to include the fuels butane, ethane, methane, and propane as well as H2; to include the effects of inlet air and fuel temperatures; and the percent theoretical air supplied.

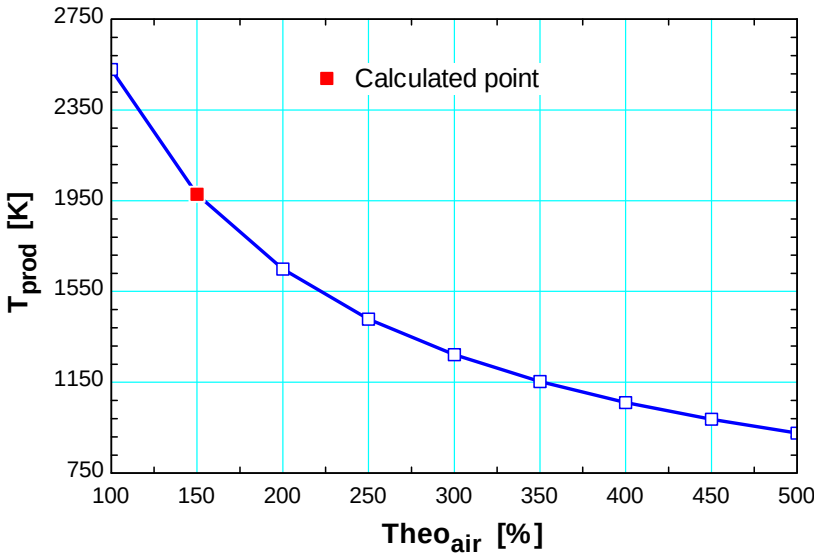
Analysis The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHy  +  (y/4 + x) (Theo_air/100) (O2 + 3.76 N2)
  <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x) (Theo_air/100)  N2 +  (y/4 + x) (Theo_air/100 - 1) O2
T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. " "The initial guess value of T_prod = 450K ."
Procedure Fuel(Fuel$:x,y,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H6' then
          x=2;y=6
             Name$='ethane'
else
If fuel$='C3H8' then
          x=3; y=8
             Name$='propane'
else
If fuel$='C4H10' then
          x=4; y=10
            Name$='butane'
else
if fuel$='CH4' then
          x=1; y=4
            Name$='methane'
else
if fuel$='H2' then
          x=0; y=2
            Name$='hydrogen'
endif; endif; endif; endif; endif
end

{"Input data from the diagram window"
T_fuel = 300 [K]
T_air = 300  [K]
Theo_air = 150  "%"
Fuel$='H2'}
Call Fuel(fuel$:x,y,Name$)
HR=enthalpy(Fuel$,T=T_fuel)+ (y/4 + x) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(y/4 + x) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=x*enthalpy(CO2,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(y/4 + x)*
(Theo_air/100)*enthalpy(N2,T=T_prod)+(y/4 + x) *(Theo_air/100 - 1)*enthalpy(O2,T=T_prod)
Moles_O2=(y/4 + x) *(Theo_air/100 - 1)
Moles_N2=3.76*(y/4 + x)* (Theo_air/100)
Moles_CO2=x;   Moles_H2O=y/2
T[1]=T_prod;  xa[1]=Theo_air  "array variable are plotted in Plot Window 1"
```

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$Theo_{air}$ [%]	T_{prod} [K]
100	2528
150	1978
200	1648
250	1428
300	1271
350	1153
400	1060
450	986.3
500	925.5



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15-79 Acetylene is burned with stoichiometric amount of oxygen. The adiabatic flame temperature is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

Analysis Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $Q = W = 0$ reduces to

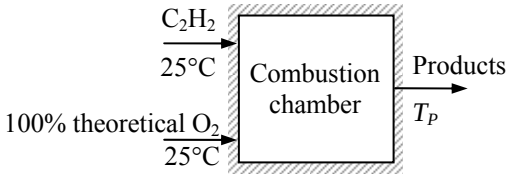
$$\sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ$$

since all the reactants are at the standard reference temperature of 25°C. Then, for the stoichiometric oxygen

$$C_2H_2 + 2.5O_2 \longrightarrow 2\,CO_2 + 1\,H_2O$$

From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol
C_2H_2 (*g*)	226,730	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO_2	-393,520	9364



Thus,

$$(2)\left(-393{,}520 + \bar{h}_{\text{CO2}} - 9364\right) + (1)\left(-241{,}820 + \bar{h}_{\text{H2O}} - 9904\right) = (1)(226{,}730) + 0 + 0$$

It yields $2\bar{h}_{\text{CO2}} + 1\bar{h}_{\text{H2O}} = 1{,}284{,}220$ kJ

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,284,220/(2+1) = 428,074 kJ/kmol. The ideal gas tables do not list enthalpy values this high. Therefore, we cannot use the tables to estimate the adiabatic flame temperature. In Table A-2b, the highest available value of specific heat is c_p = 1.234 kJ/kg·K for CO_2 at 1000 K. The specific heat of water vapor is c_p = 1.8723 kJ/kg·K (Table A-2a). Using these specific heat values,

$$(2)\left(-393{,}520 + \bar{c}_p \Delta T\right) + (1)\left(-241{,}820 + \bar{c}_p \Delta T\right) = (1)(226{,}730) + 0 + 0$$

where $\Delta T = (T_{\text{af}} - 25)°\text{C}$. The specific heats on a molar base are

$$\bar{c}_{p,\text{CO2}} = \bar{c}_p M = (1.234\ \text{kJ/kg} \cdot \text{K})(44\ \text{kg/kmol}) = 54.3\ \text{kJ/kmol} \cdot \text{K}$$
$$\bar{c}_{p,\text{H2O}} = \bar{c}_p M = (1.8723\ \text{kJ/kg} \cdot \text{K})(18\ \text{kg/kmol}) = 33.7\ \text{kJ/kmol} \cdot \text{K}$$

Substituting,

$$(2)\left(-393{,}520 + 54.3\Delta T\right) + (1)\left(-241{,}820 + 33.7\Delta T\right) = 226{,}730$$
$$(2 \times 54.3)\Delta T + 33.7\Delta T = 1{,}255{,}590$$
$$\Delta T = \frac{1{,}255{,}590\ \text{kJ/kmol}}{(2 \times 54.3 + 33.7)\ \text{kJ/kmol} \cdot \text{K}} = 8824\ \text{K}$$

Then the adiabatic flame temperature is estimated as

$$T_{\text{af}} = \Delta T + 25 = 8824 + 25 = \mathbf{8849°C}$$

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15-80 Propane is burned with stoichiometric and 50 percent excess air. The adiabatic flame temperature is to be determined for both cases.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

Analysis Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $Q = W = 0$ reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ$$

since all the reactants are at the standard reference temperature of 25°C. Then, for the stoicihiometric air

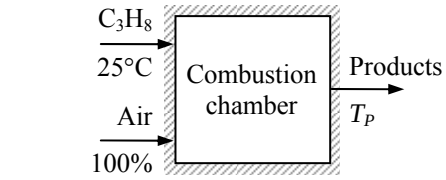
$$C_3H_8 + a_{\text{th}}(O_2 + 3.76\,N_2) \longrightarrow 3\,CO_2 + 4\,H_2O + a_{\text{th}} \times 3.76\,N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$a_{\text{th}} = 3 + 2 = 5$$

Thus,

$$C_3H_8 + 5(O_2 + 3.76\,N_2) \longrightarrow 3\,CO_2 + 4\,H_2O + 18.8\,N_2$$



From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol
C_3H_8 (*g*)	-103,850	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO_2	-393,520	9364

Thus,

$$(3)\left(-393{,}520 + \bar{h}_{CO2} - 9364\right) + (4)\left(-241{,}820 + \bar{h}_{H2O} - 9904\right) + (18.8)\left(0 + \bar{h}_{N2} - 8669\right) = (1)\left(-103{,}850\right) + 0 + 0$$

It yields $3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} = 2{,}274{,}680$ kJ

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,274,680/(3 + 4 + 18.8) = 88,166 kJ/kmol. This enthalpy value corresponds to about 2650 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2650 K, but somewhat under it because of the higher specific heat of H_2O.

At 2500 K:

$$\begin{aligned} 3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} &= 3 \times 131{,}290 + 4 \times 108{,}868 + 18.8 \times 82{,}981 \\ &= 2{,}389{,}380 \text{ kJ} \quad \text{(Higher than 2,274,680 kJ)} \end{aligned}$$

At 2450 K:

$$\begin{aligned} 3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} &= 3 \times 128{,}219 + 4 \times 106{,}183 + 18.8 \times 81{,}149 \\ &= 2{,}334{,}990 \text{ kJ} \quad \text{(Higher than 2,274,680 kJ)} \end{aligned}$$

At 2400 K:

$$\begin{aligned} 3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} &= 3 \times 125{,}152 + 4 \times 103{,}508 + 18.8 \times 79{,}320 \\ &= 2{,}280{,}704 \text{ kJ} \quad \text{(Higher than 2,274,680 kJ)} \end{aligned}$$

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At 2350 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} = 3\times 122{,}091 + 4\times 100{,}846 + 18.8\times 77{,}496$$
$$= 2{,}226{,}580 \text{ kJ (Lower than 2,274,680 kJ)}$$

By interpolation of the two results,

T_P = **2394 K = 2121°C**

When propane is burned with 50% excess air, the reaction equation may be written as

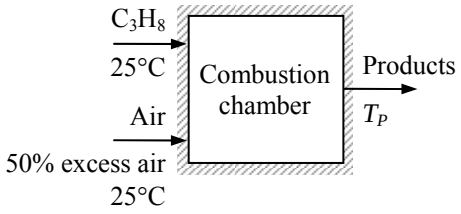
$$C_3H_8 + 1.5\times a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 0.5\times a_{th}O_2 + 1.5\times a_{th}\times 3.76N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.5a_{th} = 3 + 2 + 0.5a_{th} \longrightarrow a_{th} = 5$$

Thus,

$$C_3H_8 + 7.5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 2.5O_2 + 28.2N_2$$



Using the values in the table,

$$(3)\left(-393{,}520 + \bar{h}_{CO2} - 9364\right) + (4)\left(-241{,}820 + \bar{h}_{H2O} - 9904\right) + (2.5)\left(0 + \bar{h}_{O2} - 8682\right)$$
$$+ (28.2)\left(0 + \bar{h}_{N2} - 8669\right) = (1)\left(-103{,}850\right) + 0 + 0$$

It yields $3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 2.5\bar{h}_{O2} + 28.2\bar{h}_{N2} = 2{,}377{,}870$ kJ

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,377,870/(3+4+2.5+28.2) = 63,073 kJ/kmol. This enthalpy value corresponds to about 1960 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 1960 K, but somewhat under it because of the higher specific heat of H_2O.

At 1800 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 2.5\bar{h}_{O2} + 28.2\bar{h}_{N2} = 3\times 88{,}806 + 4\times 72{,}513 + 2.5\times 60{,}371 + 28.2\times 57{,}651$$
$$= 2{,}333{,}160 \text{ kJ (Lower than 2,377,870 kJ)}$$

At 1840 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 2.5\bar{h}_{O2} + 28.2\bar{h}_{N2} = 3\times 91{,}196 + 4\times 74{,}506 + 2.5\times 61{,}866 + 28.2\times 59{,}075$$
$$= 2{,}392{,}190 \text{ kJ (Higher than 2,377,870 kJ)}$$

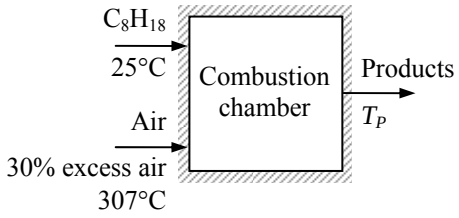
By interpolation,

T_P = **1830 K = 1557°C**

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15-81 Octane is burned with 40 percent excess air adiabatically during a steady-flow combustion process. The exit temperature of product gases is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.



Analysis Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $Q = W = 0$ reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

since all the reactants are at the standard reference temperature of 25°C. Then,

$$C_8H_{18}(l) + 1.4a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 0.4a_{th}O_2 + (1.4)(3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.4a_{th} = 8 + 4.5 + 0.4a_{th} \longrightarrow a_{th} = 12.5$$

Thus, $C_8H_{18}(g) + 17.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 5O_2 + 65.8N_2$

From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol	$\bar{h}_{580K}$ kJ/kmol
C_8H_{18} (l)	-249,950	---	---
O_2	0	8682	17,290
N_2	0	8669	16,962
H_2O (g)	-241,820	9904	---
CO_2	-393,520	9364	---

Thus,

$$(8)(-393{,}520 + \bar{h}_{CO2} - 9364) + (9)(-241{,}820 + \bar{h}_{H2O} - 9904) + (5)(0 + \bar{h}_{O2} - 8682) + (65.8)(0 + \bar{h}_{N2} - 8669) = (1)(-249{,}950) + (17.5)(0 + 17{,}290 - 8682) + (65.8)(16{,}962 - 8669)$$

It yields $8\bar{h}_{CO2} + 9\bar{h}_{H2O} + 5\bar{h}_{O2} + 65.8\bar{h}_{N2} = 6{,}548{,}788$ kJ

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 6,548,788/(8 + 9 + 5 + 65.8) = 74,588 kJ/kmol. This enthalpy value corresponds to about 2250 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2250 K, but somewhat under it because of the higher specific heat of H_2O.

At 2100 K:

$$8\bar{h}_{CO2} + 9\bar{h}_{H2O} + 5\bar{h}_{O2} + 65.8\bar{h}_{N2} = (8)(106{,}864) + (9)(87{,}735) + (5)(71{,}668) + (65.8)(68{,}417) = 6{,}504{,}706 \text{ kJ (Lower than 6,548,788 kJ)}$$

At 2150 K:

$$8\bar{h}_{CO2} + 9\bar{h}_{H2O} + 5\bar{h}_{O2} + 65.8\bar{h}_{N2} = (8)(109{,}898) + (9)(90{,}330) + (5)(73{,}573) + (65.8)(70{,}226) = 6{,}680{,}890 \text{ kJ (Higher than 6,548,788 kJ)}$$

By interpolation,

T_P = **2113 K = 1840°C**

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15-82 A certain coal is burned with 100 percent excess air adiabatically during a steady-flow combustion process. The temperature of product gases is to be determined for complete combustion and incomplete combustion cases.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

Properties The molar masses of C, H_2, N_2, O_2, S, and air are 12, 2, 28, 32, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kg of coal for simplicity. Noting that the mass percentages in this case correspond to the masses of the constituents, the mole numbers of the constituent of the coal are determined to be

$$N_{\mathrm{C}} = \frac{m_{\mathrm{C}}}{M_{\mathrm{C}}} = \frac{84.36\ \mathrm{kg}}{12\ \mathrm{kg/kmol}} = 7.03\ \mathrm{kmol}$$

$$N_{\mathrm{H2}} = \frac{m_{\mathrm{H2}}}{M_{\mathrm{H2}}} = \frac{1.89\ \mathrm{kg}}{2\ \mathrm{kg/kmol}} = 0.945\ \mathrm{kmol}$$

$$N_{\mathrm{O2}} = \frac{m_{\mathrm{O2}}}{M_{\mathrm{O2}}} = \frac{4.40\ \mathrm{kg}}{32\ \mathrm{kg/kmol}} = 0.1375\ \mathrm{kmol}$$

$$N_{\mathrm{N2}} = \frac{m_{\mathrm{N2}}}{M_{\mathrm{N2}}} = \frac{0.63\ \mathrm{kg}}{28\ \mathrm{kg/kmol}} = 0.0225\ \mathrm{kmol}$$

$$N_{\mathrm{S}} = \frac{m_{\mathrm{S}}}{M_{\mathrm{S}}} = \frac{0.89\ \mathrm{kg}}{32\ \mathrm{kg/kmol}} = 0.0278\ \mathrm{kmol}$$

84.36% C
1.89% H_2
4.40% O_2
0.63% N_2
0.89% S
7.83% ash
(by mass)

The mole number of the mixture and the mole fractions are

$$N_m = 7.03 + 0.945 + 0.1375 + 0.0225 + 0.0278 = 8.163\ \mathrm{kmol}$$

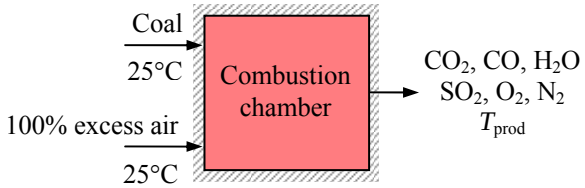
$$y_{\mathrm{C}} = \frac{N_{\mathrm{C}}}{N_m} = \frac{7.03\ \mathrm{kmol}}{8.163\ \mathrm{kmol}} = 0.8611$$

$$y_{\mathrm{H2}} = \frac{N_{\mathrm{H2}}}{N_m} = \frac{0.945\ \mathrm{kmol}}{8.163\ \mathrm{kmol}} = 0.1158$$

$$y_{\mathrm{O2}} = \frac{N_{\mathrm{O2}}}{N_m} = \frac{0.1375\ \mathrm{kmol}}{8.163\ \mathrm{kmol}} = 0.01684$$

$$y_{\mathrm{N2}} = \frac{N_{\mathrm{N2}}}{N_m} = \frac{0.0225\ \mathrm{kmol}}{8.163\ \mathrm{kmol}} = 0.00276$$

$$y_{\mathrm{S}} = \frac{N_{\mathrm{S}}}{N_m} = \frac{0.0278\ \mathrm{kmol}}{8.163\ \mathrm{kmol}} = 0.003407$$



Ash consists of the non-combustible matter in coal. Therefore, the mass of ash content that enters the combustion chamber is equal to the mass content that leaves. Disregarding this non-reacting component for simplicity, the combustion equation may be written as

$$0.8611\mathrm{C} + 0.1158\mathrm{H_2} + 0.01684\mathrm{O_2} + 0.00276\mathrm{N_2} + 0.00341\mathrm{S} + 2a_{\mathrm{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow x(0.97\mathrm{CO_2} + 0.03\mathrm{CO}) + y\mathrm{H_2O} + z\mathrm{SO_2} + k\mathrm{N_2} + a_{\mathrm{th}}\mathrm{O_2}$$

According to the species balances,

C balance: $x = 0.8611$

H_2 balance: $y = 0.1158$

S balance: $z = 0.00341$

O_2 balance:

$$0.01684 + 2a_{\mathrm{th}} = 0.97x + 0.015x + 0.5y + z + a_{\mathrm{th}}$$

$$a_{\mathrm{th}} = (0.97)(0.8611) + (0.015)(0.8611) + (0.5)(0.1158) + 0.00341 - 0.01684 = 0.8927$$

N_2 balance: $0.00276 + 2 \times 3.76 a_{\mathrm{th}} = k \longrightarrow k = 0.00276 + 2 \times 3.76 \times 0.8927 = 6.72$

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Substituting,

$$0.8611\text{C} + 0.1158\text{H}_2 + 0.01684\text{O}_2 + 0.00276\text{N}_2 + 0.003407\text{S} + 1.785(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.8353\text{CO}_2 + 0.0258\text{CO} + 0.1158\text{H}_2\text{O} + 0.00341\text{SO}_2 + 6.72\text{N}_2 + 0.8927\text{O}_2$$

Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $Q = W = 0$ reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ$$

From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\text{K}}$ kJ/kmol
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO	-110,530	8669
CO_2	-110,530	8669

Thus,

$$(0.8353)(-393{,}520 + \bar{h}_{\text{CO2}} - 9364) + (0.0258)(-110{,}530 + \bar{h}_{\text{CO}} - 8669) + (0.1158)(-241{,}820 + \bar{h}_{\text{H2O}} - 9904)$$
$$+ (0.8927)(0 + \bar{h}_{\text{O2}} - 8682) + (6.72)(0 + \bar{h}_{\text{N2}} - 8669) = 0$$

It yields $$0.8353\bar{h}_{\text{CO2}} + 0.0258\bar{h}_{\text{CO}} + 0.1158\bar{h}_{\text{H2O}} + 0.8927\bar{h}_{\text{O2}} + 6.72\bar{h}_{\text{N2}} = 434{,}760 \text{ kJ}$$

The product temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields

434,760/(0.8353+0.0258+0.1158+0.00341+6.72+0.8927) = 50,595 kJ/kmol.

This enthalpy value corresponds to about 1600 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 1600 K, but somewhat under it because of the higher specific heat of H_2O.

At 1500 K:

$$0.8353\bar{h}_{\text{CO2}} + 0.0258\bar{h}_{\text{CO}} + 0.1158\bar{h}_{\text{H2O}} + 0.9095\bar{h}_{\text{O2}} + 6.842\bar{h}_{\text{N2}}$$
$$= (0.8353)(71{,}078) + (0.0258)(47{,}517) + (0.1158)(57{,}999) + (0.8927)(49{,}292) + (6.72)(47{,}073)$$
$$= 427{,}647 \text{ kJ} \quad \text{(Lower than 434,760 kJ)}$$

At 1520 K:

$$0.8353\bar{h}_{\text{CO2}} + 0.0258\bar{h}_{\text{CO}} + 0.1158\bar{h}_{\text{H2O}} + 0.9095\bar{h}_{\text{O2}} + 6.842\bar{h}_{\text{N2}}$$
$$= (0.8353)(72{,}246) + (0.0258)(48{,}222) + (0.1158)(58{,}942) + (0.8927)(50{,}024) + (6.72)(47{,}771)$$
$$= 434{,}094 \text{ kJ} \quad \text{(Lower than 434,760 kJ)}$$

By extrapolation, T_P = **1522 K = 1249°C**

We repeat the calculations for the complete combustion now:

The combustion equation in this case may be written as

$$0.8611\text{C} + 0.1158\text{H}_2 + 0.01684\text{O}_2 + 0.00276\text{N}_2 + 0.00341\text{S} + 2a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow x\text{CO}_2 + y\text{H}_2\text{O} + z\text{SO}_2 + k\text{N}_2 + a_{\text{th}}\text{O}_2$$

According to the species balances,

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C balance: $x = 0.8611$

H_2 balance: $y = 0.1158$

S balance: $z = 0.00341$

O_2 balance:

$$0.01684 + 2a_{th} = x + 0.5y + z + a_{th} \longrightarrow a_{th} = 0.8611 + (0.5)(0.1158) + 0.00341 - 0.01684 = 0.9056$$

N_2 balance: $0.00276 + 2 \times 3.76 a_{th} = k \longrightarrow k = 0.00276 + 2 \times 3.76 \times 0.9056 = 6.81$

Substituting,

$$0.8611\text{C} + 0.1158\text{H}_2 + 0.01684\text{O}_2 + 0.00276\text{N}_2 + 0.003407\text{S} + 1.819(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.8611\text{CO}_2 + 0.1158\text{H}_2\text{O} + 0.00341\text{SO}_2 + 6.81\text{N}_2 + 0.9056\text{O}_2$$

Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $Q = W = 0$ reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ$$

From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol
O_2	0	8682
N_2	0	8669
H_2O (g)	-241,820	9904
CO	-110,530	8669
CO_2	0	8682

Thus,

$$(0.8611)(-393{,}520 + \bar{h}_{CO2} - 9364) + (0.1158)(-241{,}820 + \bar{h}_{H2O} - 9904)$$
$$+ (0.9056)(0 + \bar{h}_{O2} - 8682) + (6.81)(0 + \bar{h}_{N2} - 8669) = 0$$

It yields $0.8611\bar{h}_{CO2} + 0.1158\bar{h}_{H2O} + 0.9056\bar{h}_{O2} + 6.81\bar{h}_{N2} = 442{,}971 \text{ kJ}$

The product temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields

442,971/(0.8611+0.1158+0.00341+6.81+0.9056) = 50,940 kJ/kmol.

This enthalpy value corresponds to about 1600 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 1600 K, but somewhat under it because of the higher specific heat of H_2O.

At 1500 K:

$$0.8611\bar{h}_{CO2} + 0.1158\bar{h}_{H2O} + 0.9056\bar{h}_{O2} + 6.81\bar{h}_{N2}$$
$$= (0.8611)(71{,}078) + (0.1158)(57{,}999) + (0.9056)(49{,}292) + (6.81)(47{,}073)$$
$$= 433{,}128 \text{ kJ} \quad \text{(Lower than 442,971 kJ)}$$

At 1520 K:

$$0.8611\bar{h}_{CO2} + 0.1158\bar{h}_{H2O} + 0.9056\bar{h}_{O2} + 6.81\bar{h}_{N2}$$
$$= (0.8611)(72{,}246) + (0.1158)(58{,}942) + (0.9056)(50{,}024) + (6.81)(47{,}771)$$
$$= 439{,}658 \text{ kJ} \quad \text{(Lower than 442,971 kJ)}$$

By extrapolation, T_P = **1530 K = 1257°C**

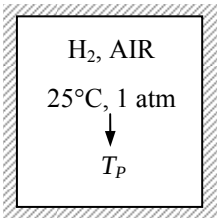
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15-83 A mixture of hydrogen and the stoichiometric amount of air contained in a constant-volume tank is ignited. The final temperature in the tank is to be determined.

Assumptions **1** The tank is adiabatic. **2** Both the reactants and products are ideal gases. **3** There are no work interactions. **4** Combustion is complete.

Analysis The combustion equation of H_2 with stoichiometric amount of air is

$$H_2 + 0.5(O_2 + 3.76N_2) \longrightarrow H_2O + 1.88N_2$$



The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$),

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P = \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by R_uT.

It yields

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298\ K} - R_uT\right)_P = \sum N_R \left(\bar{h}_f^\circ R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol
H_2	0	8468
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904

Thus,

$$(1)(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314 \times T_P) + (1.88)(0 + \bar{h}_{N_2} - 8669 - 8.314 \times T_P)$$
$$= (1)(0 - 8.314 \times 298) + (0.5)(0 - 8.314 \times 298) + (1.88)(0 - 8.314 \times 298)$$

It yields $\bar{h}_{H_2O} + 1.88\bar{h}_{N_2} - 23.94 \times T_P = 259{,}648$ kJ

The temperature of the product gases is obtained from a trial and error solution,

At 3050 K: $\bar{h}_{H_2O} + 1.88\bar{h}_{N_2} - 23.94 \times T_P = (1)(139{,}051) + (1.88)(103{,}260) - (23.94)(3050)$
$= 260{,}163$ kJ (Higher than 259,648 kJ)

At 3000 K: $\bar{h}_{H_2O} + 1.88\bar{h}_{N_2} - 23.94 \times T_P = (1)(136{,}264) + (1.88)(101{,}407) - (23.94)(3000)$
$= 255{,}089$ kJ (Lower than 259,648 kJ)

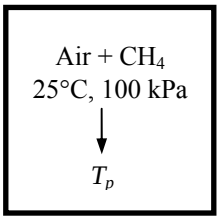
By interpolation, T_P = **3045 K**

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15-84 Methane is burned with 300 percent excess air adiabatically in a constant volume container. The final pressure and temperature of product gases are to be determined.

Assumptions **1** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

Analysis The combustion equation is



$$CH_4 + 4a_{th}[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O + 4a_{th} \times 3.76\,N_2 + 3a_{th}O_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$4a_{th} = 1 + 1 + 3a_{th} \longrightarrow a_{th} = 2$$

Substituting, $$CH_4 + 8[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O + 30.08\,N_2 + 6O_2$$

For this constant-volume process, the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $Q = W = 0$ reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and products are assumed to be ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by R_uT. It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - R_uT\right)_R$$

From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol
CH_4 (g)	-74,850	---
O_2	0	8682
N_2	0	8669
H_2O (g)	-241,820	9904
CO_2	-393,520	9364

Thus,

$$(1)(-393{,}520 + \bar{h}_{CO2} - 9364 - 8.314 \times T_p) + (2)(-241{,}820 + \bar{h}_{H2O} - 9904 - 8.314 \times T_p)$$
$$+ (6)(0 + \bar{h}_{O2} - 8682 - 8.314 \times T_p) + (30.08)(0 + \bar{h}_{N2} - 8669 - 8.314 \times T_p)$$
$$= (1)(-74{,}850 - 8.314 \times 298) + (8)(0 - 8.314 \times 298) + (30.08)(-8.314 \times 298)$$

It yields $$\bar{h}_{CO2} + 2\bar{h}_{H2O} + 6\bar{h}_{O2} + 30.08\bar{h}_{N2} - 324.9T_p = -171{,}674 + 1{,}219{,}188 = 1{,}047{,}514 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess may be obtained by assuming all the products are nitrogen and using nitrogen enthalpy in the above equation. That is,

$$39.08\bar{h}_{N2} - 324.9T_p = 1{,}047{,}514 \text{ kJ}$$

An investigation of Table A-18 shows that this equation is satisfied at a temperature close to 1200 K but it will be somewhat under it because of the higher specific heat of H_2O.

At 1100 K: $$(48{,}258) + (2)(40{,}071) + (6)(34{,}899) + (30.08)(33{,}426) - (324.9)(1100) = 985{,}858$$
(Lower than 1,047,514 kJ)

At 1200 K: $$(53{,}848) + (2)(44{,}380) + (6)(38{,}447) + (30.08)(36{,}777) - (324.9)(1200) = 1{,}089{,}662$$
(Higher than 1,047,514 kJ)

By interpolation, T_P = **1159 K**

The volume of reactants when 1 kmol of fuel is burned is

$$V = V_{fuel} + V_{air} = (N_{fuel} + N_{air})\frac{R_uT}{P} = (1 + 38.08)\text{ kmol})\frac{(8.314\text{ kJ/kmol}\cdot\text{K})(298\text{ K})}{101.3\text{ kPa}} = 955.8\text{ m}^3$$

The final pressure is then

$$P = N_{prod}\frac{R_uT}{V} = (39.08\text{ kmol})\frac{(8.314\text{ kJ/kmol}\cdot\text{K})(1159\text{ K})}{955.8\text{ m}^3} = \mathbf{394\text{ kPa}}$$

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Entropy Change and Second Law Analysis of Reacting Systems

15-85C Assuming the system exchanges heat with the surroundings at T_0, the increase-in-entropy principle can be expressed as

$$S_{\text{gen}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{\text{out}}}{T_0}$$

15-86C By subtracting $R\ln(P/P_0)$ from the tabulated value at 1 atm. Here P is the actual pressure of the substance and P_0 is the atmospheric pressure.

15-87C It represents the reversible work associated with the formation of that compound.

15-88 Hydrogen is burned steadily with oxygen. The reversible work and exergy destruction (or irreversibility) are to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis The combustion equation is

$$2\text{H}_2 + 1\text{O}_2 \longrightarrow 2\text{H}_2\text{O}.$$

The H_2, the O_2, and the H_2O are at 25°C and 1 atm, which is the standard reference state and also the state of the surroundings. Therefore, the reversible work in this case is simply the difference between the Gibbs function of formation of the reactants and that of the products,

$$\begin{aligned} W_{\text{rev}} &= \sum N_R \bar{g}^\circ_{f,R} - \sum N_P \bar{g}^\circ_{f,P} = N_{\text{H}_2} \bar{g}^\circ_{f,\text{H}_2}{}^{\nearrow 0} + N_{\text{O}_2} \bar{g}^\circ_{f,\text{O}_2}{}^{\nearrow 0} - N_{\text{H}_2\text{O}} \bar{g}^\circ_{f,\text{H}_2\text{O}} = -N_{\text{H}_2\text{O}} \bar{g}^\circ_{f,\text{H}_2\text{O}} \\ &= -(2\text{ kmol})(-237{,}180\text{ kJ/kmol}) = \mathbf{474{,}360\text{ kJ}} \quad (\text{for 2 kmol of } \text{H}_2) \end{aligned}$$

since the g°_f of stable elements at 25°C and 1 atm is zero. Therefore, 474,360 kJ of work could be done as 2 kmol of H_2 is burned with 1 kmol of O_2 at 25°C and 1 atm in an environment at the same state. The reversible work in this case represents the exergy of the reactants since the product (the H_2O) is at the state of the surroundings.

This process involves no actual work. Therefore, the reversible work and exergy destruction are identical,

$X_{\text{destruction}} =$ **474,360 kJ** (for 2 kmol of H_2)

We could also determine the reversible work without involving the Gibbs function,

$$\begin{aligned} W_{\text{rev}} &= \sum N_R \left(\bar{h}^\circ_f + \bar{h} - \bar{h}^\circ - T_0 \bar{s}\right)_R - \sum N_P \left(\bar{h}^\circ_f + \bar{h} - \bar{h}^\circ - T_0 \bar{s}\right)_P \\ &= \sum N_R \left(\bar{h}^\circ_f - T_0 \bar{s}\right)_R - \sum N_P \left(\bar{h}^\circ_f - T_0 \bar{s}\right)_P \\ &= N_{\text{H}_2} \left(\bar{h}^\circ_f - T_0 \bar{s}^\circ\right)_{\text{H}_2} + N_{\text{O}_2} \left(\bar{h}^\circ_f - T_0 \bar{s}^\circ\right)_{\text{O}_2} - N_{\text{H}_2\text{O}} \left(\bar{h}^\circ_f - T_0 \bar{s}^\circ\right)_{\text{H}_2\text{O}} \end{aligned}$$

Substituting,

$$W_{\text{rev}} = (2)(0 - 298 \times 130.58) + (1)(0 - 298 \times 205.03) - (2)(-285{,}830 - 298 \times 69.92) = 474{,}400\text{ kJ}$$

which is almost identical to the result obtained before.

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15-89 Ethylene gas is burned steadily with 20 percent excess air. The temperature of products, the entropy generation, and the exergy destruction (or irreversibility) are to be determined.

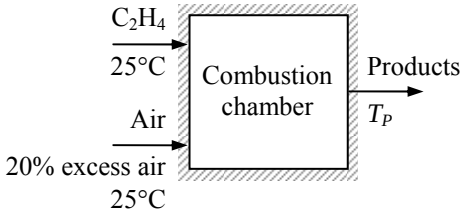
Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis (*a*) The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol of C_2H_4, the combustion equation can be written as

$$C_2H_4(g) + 1.2a_{th}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 2H_2O + 0.2a_{th}O_2 + (1.2)(3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.2a_{th} = 2 + 1 + 0.2a_{th} \quad \longrightarrow \quad a_{th} = 3$$



Thus,

$$C_2H_4(g) + 3.6(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 2H_2O + 0.6O_2 + 13.54N_2$$

Under steady-flow conditions, the exit temperature of the product gases can be determined from the steady-flow energy equation, which reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ = \left(N\bar{h}_f^\circ\right)_{C_2H_4}$$

since all the reactants are at the standard reference state, and for O_2 and N_2. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol
C_2H_4 (*g*)	52,280	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO_2	-393,520	9364

Substituting,

$$(2)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (0.6)(0 + \bar{h}_{O_2} - 8682) + (13.54)(0 + \bar{h}_{N_2} - 8669) = (1)(52{,}280)$$

or,

$$2\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 0.6\bar{h}_{O_2} + 13.54\bar{h}_{N_2} = 1{,}484{,}083 \text{ kJ}$$

By trial and error,

T_P = **2269.6 K**

(*b*) The entropy generation during this adiabatic process is determined from

$$S_{gen} = S_P - S_R = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R$$

The C_2H_4 is at 25°C and 1 atm, and thus its absolute entropy is 219.83 kJ/kmol·K (Table A-26). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where y_i is the mole fraction of component *i* . Also,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

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The entropy calculations can be presented in tabular form as

	N_i	y_i	$\bar{s}_i^\circ(T,1\text{atm})$	$R_u \ln(y_i P_m)$	$N_i \bar{s}_i$
C_2H_4	1	1.00	219.83	---	219.83
O_2	3.6	0.21	205.14	-12.98	784.87
N_2	13.54	0.79	191.61	-1.96	2620.94
					S_R = 3625.64 kJ/K
CO_2	2	0.1103	316.881	-18.329	670.42
H_2O	2	0.1103	271.134	-18.329	578.93
O_2	0.6	0.0331	273.467	-28.336	181.08
N_2	13.54	0.7464	256.541	-2.432	3506.49
					S_P = 4936.92 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R = 4936.92 - 3625.64 = \mathbf{1311.28\ kJ/kmol \cdot K}$$

and

(*c*) $$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\text{ K})(1311.28\text{ kJ/kmol} \cdot \text{K } C_2H_4) = \mathbf{390{,}760\ kJ}\ (\text{per kmol } C_2H_4)$$

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15-90 Liquid octane is burned steadily with 50 percent excess air. The heat transfer rate from the combustion chamber, the entropy generation rate, and the reversible work and exergy destruction rate are to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis (*a*) The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol C_8H_{18}, the combustion equation can be written as

$$C_8H_{18}(\ell) + 1.5a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 0.5a_{th}O_2 + (1.5)(3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.5a_{th} = 8 + 4.5 + 0.5a_{th} \quad \longrightarrow \quad a_{th} = 12.5$$

Thus,

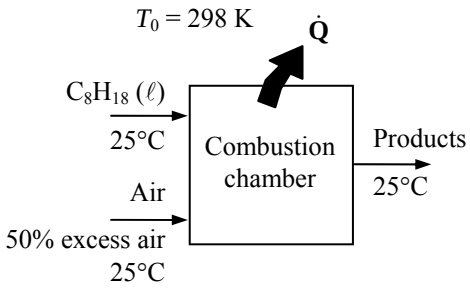
$$C_8H_{18}(\ell) + 18.75(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 6.25O_2 + 70.5N_2$$

Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P\bar{h}_{f,P}^\circ - \sum N_R\bar{h}_{f,R}^\circ$$

since all of the reactants are at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol
C_8H_{18} (ℓ)	-249,950
O_2	0
N_2	0
H_2O (l)	-285,830
CO_2	-393,520



Substituting,

$$-Q_{out} = (8)(-393{,}520) + (9)(-285{,}830) + 0 + 0 - (1)(-249{,}950) - 0 - 0 = -5{,}470{,}680 \text{ kJ/kmol of } C_8H_{18} \text{ or}$$

$$Q_{out} = 5{,}470{,}680 \text{ kJ/kmol of } C_8H_{18}$$

The C_8H_{18} is burned at a rate of 0.25 kg/min or

$$\dot{N} = \frac{\dot{m}}{M} = \frac{0.25 \text{ kg/min}}{[(8)(12) + (18)(1)] \text{ kg/kmol}} = 2.193 \times 10^{-3} \text{ kmol/min}$$

Thus,

$$\dot{Q}_{out} = \dot{N}Q_{out} = (2.193 \times 10^{-3} \text{ kmol/min})(5{,}470{,}680 \text{ kJ/kmol}) = \mathbf{11{,}997 \text{ kJ/min}}$$

The heat transfer for this process is also equivalent to the enthalpy of combustion of liquid C_8H_{18}, which could easily be de determined from Table A-27 to be $\bar{h}_C = 5{,}470{,}740$ kJ/kmol C_8H_{18}.

(*b*) The entropy generation during this process is determined from

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} \longrightarrow S_{gen} = \sum N_P\bar{s}_P - \sum N_R\bar{s}_R + \frac{Q_{out}}{T_{surr}}$$

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The C_8H_{18} is at 25°C and 1 atm, and thus its absolute entropy is $\bar{s}_{C_8H_{18}}$ = 360.79 kJ/kmol.K (Table A-26). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where y_i is the mole fraction of component *i*. Also,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^{\circ}(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

	$\mathbf{N_i}$	$\mathbf{y_i}$	$\bar{\mathbf{s}}_\mathbf{i}^{\circ}\mathbf{(T,1atm)}$	$\mathbf{R_u ln(y_i P_m)}$	$\mathbf{N_i \bar{s}_i}$
C_8H_{18}	1	1.00	360.79	---	360.79
O_2	18.75	0.21	205.14	-12.98	4089.75
N_2	70.50	0.79	191.61	-1.96	13646.69
					S_R = 18,097.23 kJ/K
CO_2	8	0.0944	213.80	-19.62	1867.3
H_2O (ℓ)	9	---	69.92	---	629.3
O_2	6.25	0.0737	205.04	-21.68	1417.6
N_2	70.50	0.8319	191.61	-1.53	13,616.3
					S_P = 17,531 kJ/K

Thus,

$$S_{gen} = S_P - S_R + \frac{Q_{surr}}{T_{surr}} = 17{,}531 - 18{,}097 + \frac{5{,}470{,}523 \text{ kJ}}{298 \text{ K}} = 17{,}798 \text{ kJ/kmol} \cdot \text{K}$$

and

$$\dot{S}_{gen} = \dot{N} S_{gen} = (2.193 \times 10^{-3} \text{ kmol/min})(17{,}798 \text{ kJ/kmol} \cdot \text{K}) = \mathbf{39.03 \text{ kJ/min} \cdot \text{K}}$$

(*c*) The exergy destruction rate associated with this process is determined from

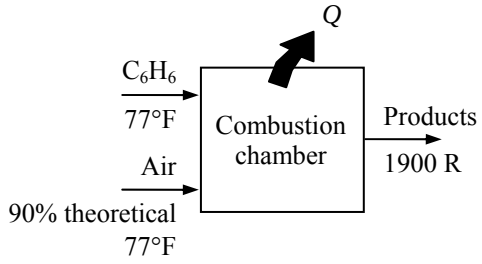
$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = (298 \text{ K})(39.03 \text{ kJ/min} \cdot \text{K}) = 11{,}632 \text{ kJ/min} = \mathbf{193.9 \text{ kW}}$$

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15-91E Benzene gas is burned steadily with 90 percent theoretical air. The heat transfer rate from the combustion chamber and the exergy destruction are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and the combustion gases are ideal gases. **3** Changes in kinetic and potential energies are negligible.

Analysis (*a*) The fuel is burned with insufficient amount of air, and thus the products will contain some CO as well as CO_2, H_2O, and N_2. The theoretical combustion equation of C_6H_6 is



$$C_6H_6 + a_{th}(O_2 + 3.76N_2) \longrightarrow 6CO_2 + 3H_2O + 3.76a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$a_{th} = 6 + 1.5 = 7.5$$

Then the actual combustion equation can be written as

$$C_6H_6 + (0.90)(7.5)(O_2 + 3.76N_2) \longrightarrow xCO_2 + (6-x)CO + 3H_2O + 25.38N_2$$

The value of x is determined from an O_2 balance,

$$(0.90)(7.5) = x + (6-x)/2 + 1.5 \longrightarrow x = 4.5$$

Thus,

$$C_6H_6 + 6.75(O_2 + 3.76N_2) \longrightarrow 4.5CO_2 + 1.5CO + 3H_2O + 25.38N_2$$

Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\bar{h}_{f,R}^\circ$$

since all of the reactants are at 77°F. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ Btu/lbmol	$\bar{h}_{537\ R}$ Btu/lbmol	$\bar{h}_{1900R}$ Btu/lbmol
C_6H_6 (*g*)	35,680	---	---
O_2	0	3725.1	14,322
N_2	0	3729.5	13,742
H_2O (*g*)	-104,040	4258.0	16,428
CO	-47,540	3725.1	13,850
CO_2	-169,300	4027.5	19,698

Thus,

$$\begin{aligned}-Q_{out} &= (4.5)(-169,300 + 19,698 - 4027.5) + (1.5)(-47,540 + 13,850 - 3725.1) \\ &\quad + (3)(-104,040 + 16,428 - 4258) + (25.38)(0 + 13,742 - 3729.5) - (1)(35,680) - 0 - 0 \\ &= \mathbf{-804,630\ Btu/lbmol\ of\ C_6H_6}\end{aligned}$$

(*b*) The entropy generation during this process is determined from

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = \sum N_P\bar{s}_P - \sum N_R\bar{s}_R + \frac{Q_{out}}{T_{surr}}$$

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The C_6H_6 is at 77°F and 1 atm, and thus its absolute entropy is $\bar{s}_{C_6H_6} = 64.34$ Btu/lbmol·R (Table A-26E). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where y_i is the mole fraction of component *i*. Also,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^{\circ}(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

	$\mathbf{N_i}$	$\mathbf{y_i}$	$\bar{\mathbf{s}}_\mathbf{i}^{\circ}\mathbf{(T,1atm)}$	$\mathbf{R_u ln(y_i P_m)}$	$\mathbf{N_i \bar{s}_i}$
C_6H_6	1	1.00	64.34	---	64.34
O_2	6.75	0.21	49.00	-3.10	351.68
N_2	25.38	0.79	45.77	-0.47	1173.57
					S_R = 1589.59 Btu/R
CO_2	4.5	0.1309	64.999	-4.038	310.67
CO	1.5	0.0436	56.509	-6.222	94.10
H_2O (*g*)	3	0.0873	56.097	-4.843	182.82
N_2	25.38	0.7382	54.896	-0.603	1408.56
					S_P = 1996.15 Btu/R

Thus,

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = 1996.15 - 1589.59 + \frac{804{,}630}{537} = 1904.9 \text{ Btu/R}$$

Then the exergy destroyed is determined from

$$X_{destroyed} = T_0 S_{gen} = (537 \text{ R})(1904.9 \text{ Btu/lbmol} \cdot \text{R}) = \mathbf{1{,}022{,}950\ Btu/R}\ (\text{per lbmol } C_6H_6)$$

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15-92 Liquid propane is burned steadily with 150 percent excess air. The mass flow rate of air, the heat transfer rate from the combustion chamber, and the rate of entropy generation are to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Properties The molar masses of C_3H_8 and air are 44 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis (*a*) The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol of C_3H_8, the combustion equation can be written as

$$C_3H_8(\ell) + 2.5a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 1.5a_{th}O_2 + (2.5)(3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$2.5a_{th} = 3 + 2 + 1.5a_{th} \quad \longrightarrow \quad a_{th} = 5$$

Substituting,

$$C_3H_8(\ell) + 12.5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 7.5O_2 + 47N_2$$

The air-fuel ratio for this combustion process is

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(12.5 \times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{(3 \text{ kmol})(12 \text{ kg/kmol}) + (4 \text{ kmol})(2 \text{ kg/kmol})} = 39.2 \text{ kg air/kg fuel}$$

Thus,

$$\dot{m}_{\text{air}} = (\text{AF})(\dot{m}_{\text{fuel}}) = (39.2 \text{ kg air/kg fuel})(0.4 \text{ kg fuel/min}) = \mathbf{15.7 \text{ kg air/min}}$$

(*b*) Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{\text{out}} = \sum N_P(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_P - \sum N_R(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables, (The $\bar{h}_f^\circ$ of liquid propane is obtained by adding the h_{fg} at 25°C to $\bar{h}_f^\circ$ of gaseous propane).

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{285\text{ K}}$ kJ/kmol	$\bar{h}_{298\text{ K}}$ kJ/kmol	$\bar{h}_{1200\text{ K}}$ kJ/kmol
C_3H_8 (ℓ)	-118,910	---	---	---
O_2	0	8296.5	8682	38,447
N_2	0	8286.5	8669	36,777
H_2O (*g*)	-241,820	---	9904	44,380
CO_2	-393,520	---	9364	53,848

Thus,

$$\begin{aligned} -Q_{\text{out}} &= (3)(-393{,}520 + 53{,}848 - 9364) + (4)(-241{,}820 + 44{,}380 - 9904) \\ &\quad + (7.5)(0 + 38{,}447 - 8682) + (47)(0 + 36{,}777 - 8669) - (1)(-118{,}910 + h_{298} - h_{298}) \\ &\quad - (12.5)(0 + 8296.5 - 8682) - (47)(0 + 8286.5 - 8669) \\ &= -190{,}464 \text{ kJ/kmol of } C_3H_8 \end{aligned}$$

Thus 190,464 kJ of heat is transferred from the combustion chamber for each kmol (44 kg) of propane. This corresponds to 190,464/44 = 4328.7 kJ of heat transfer per kg of propane. Then the rate of heat transfer for a mass flow rate of 0.4 kg/min for the propane becomes

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$$\dot{Q}_{\text{out}} = \dot{m} q_{\text{out}} = (0.4 \text{ kg/min})(4328.7 \text{ kJ/kg}) = \mathbf{1732 \text{ kJ/min}}$$

(*c*) The entropy generation during this process is determined from

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

The C_3H_8 is at 25°C and 1 atm, and thus its absolute entropy for the gas phase is $\bar{s}_{C_3H_8} = 269.91 \text{kJ/kmol·K}$ (Table A-26). Then the entropy of $C_3H_8(\ell)$ is obtained from

$$s_{C_3H_8}(\ell) \cong s_{C_3H_8}(\text{g}) - s_{fg} = s_{C_3H_8}(\text{g}) - \frac{\bar{h}_{fg}}{T} = 269.91 - \frac{15{,}060}{298.15} = 219.4 \text{ kJ/kmol} \cdot \text{K}$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\text{total}}$, where y_i is the mole fraction of component *i*. Then,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i \left(\bar{s}_i^{\circ}(T, P_0) - R_u \ln(y_i P_m) \right)$$

The entropy calculations can be presented in tabular form as

	$\mathbf{N_i}$	$\mathbf{y_i}$	$\bar{\mathbf{s}}_\mathbf{i}^{\circ}\mathbf{(T,1atm)}$	$\mathbf{R_u ln(y_i P_m)}$	$\mathbf{N_i \bar{s}_i}$
C_3H_8	1	---	219.40	---	219.40
O_2	12.5	0.21	203.70	-12.98	2708.50
N_2	47	0.79	190.18	-1.96	9030.58
				S_R =	11,958.48 kJ/K
CO_2	3	0.0488	279.307	-25.112	913.26
H_2O (*g*)	4	0.0650	240.333	-22.720	1052.21
O_2	7.5	0.1220	249.906	-17.494	2005.50
N_2	47	0.7642	234.115	-2.236	11108.50
				S_P =	15,079.47 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = 15{,}079.47 - 11{,}958.48 + \frac{190{,}464}{298} = 3760.1 \text{ kJ/K} \left(\text{per kmol } C_3H_8\right)$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = (\dot{N})(S_{\text{gen}}) = \left(\frac{0.4}{44} \text{ kmol/min} \right)(3760.1 \text{ kJ/kmol} \cdot \text{K}) = \mathbf{34.2 \text{ kJ/min} \cdot \text{K}}$$

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15-93 Problem 15-92 is reconsidered. The effect of the surroundings temperature on the rate of exergy destruction is to be studied.

Analysis The problem is solved using EES, and the solution is given below.

```
Fuel$ = 'Propane (C3H8)_liq'
T_fuel = (25 + 273.15) "[K]"
P_fuel = 101.3 [kPa]
m_dot_fuel = 0.4 [kg/min]*Convert(kg/min, kg/s)
Ex = 1.5 "Excess air"
P_air = 101.3 [kPa]
T_air = (12+273.15) "[K]"
T_prod = 1200 [K]
P_prod = 101.3 [kPa]
Mw_air = 28.97 "lbm/lbmol_air"
Mw_C3H8=(3*12+8*1) "kg/kmol_C3H8"
{TsurrC = 25 [C]}
T_surr = TsurrC+273.15 "[K]"

"For  theoretical dry air, the complete combustion equation is"
"C3H8 + A_th(O2+3.76 N2)=3 CO2+4 H2O +  A_th (3.76) N2 "

2*A_th=3*2+4*1"theoretical O balance"

"The balanced combustion equation with Ex%/100 excess moist air is"
"C3H8 + (1+EX)A_th(O2+3.76 N2)=3 CO2+ 4 H2O + (1+Ex) A_th (3.76) N2+ Ex( A_th) O2 "

"The air-fuel ratio on a mass basis is:"
AF = (1+Ex)*A_th*4.76*Mw_air/(1*Mw_C3H8) "kg_air/kg_fuel"

"The air mass flow rate is:"
m_dot_air = m_dot_fuel * AF

"Apply First Law SSSF to the combustion process per kilomole of fuel:"
E_in - E_out = DELTAE_cv
E_in =HR

"Since EES gives the enthalpy of gasesous components, we adjust the
EES calculated enthalpy to get the liquid enthalpy.  Subtracting the enthalpy
of vaporization from the gaseous enthalpy gives  the enthalpy of the liquid fuel.
h_fuel(liq) = h_fuel(gas) - h_fg_fuel"
h_fg_fuel = 15060 "kJ/kmol from Table A-27"
HR = 1*(enthalpy(C3H8, T=T_fuel) - h_fg_fuel)+ (1+Ex)*A_th*enthalpy(O2,T=T_air)+(1+Ex)*A_th*3.76
*enthalpy(N2,T=T_air)
E_out = HP + Q_out
HP=3*enthalpy(CO2,T=T_prod)+4*enthalpy(H2O,T=T_prod)+(1+Ex)*A_th*3.76*
enthalpy(N2,T=T_prod)+Ex*A_th*enthalpy(O2,T=T_prod)
DELTAE_cv = 0 "Steady-flow requirement"

"The heat transfer rate from the combustion chamber is:"
Q_dot_out=Q_out"kJ/kmol_fuel"/(Mw_C3H8 "kg/kmol_fuel")*m_dot_fuel"kg/s"  "kW"

"Entopy Generation due to the combustion process and heat rejection to the surroundings:"

"Entopy of the reactants per kilomole of fuel:"
P_O2_reac= 1/4.76*P_air  "Dalton's law of partial pressures for O2 in air"
s_O2_reac=entropy(O2,T=T_air,P=P_O2_reac)
P_N2_reac= 3.76/4.76*P_air "Dalton's law of partial pressures for N2 in air"
s_N2_reac=entropy(N2,T=T_air,P=P_N2_reac)
```

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```
s_C3H8_reac=entropy(C3H8, T=T_fuel,P=P_fuel) - s_fg_fuel "Adjust the EES gaseous value by s_fg"
"For phase change, s_fg is given by:"
s_fg_fuel = h_fg_fuel/T_fuel
SR = 1*s_C3H8_reac + (1+Ex)*A_th*s_O2_reac + (1+Ex)*A_th*3.76*s_N2_reac

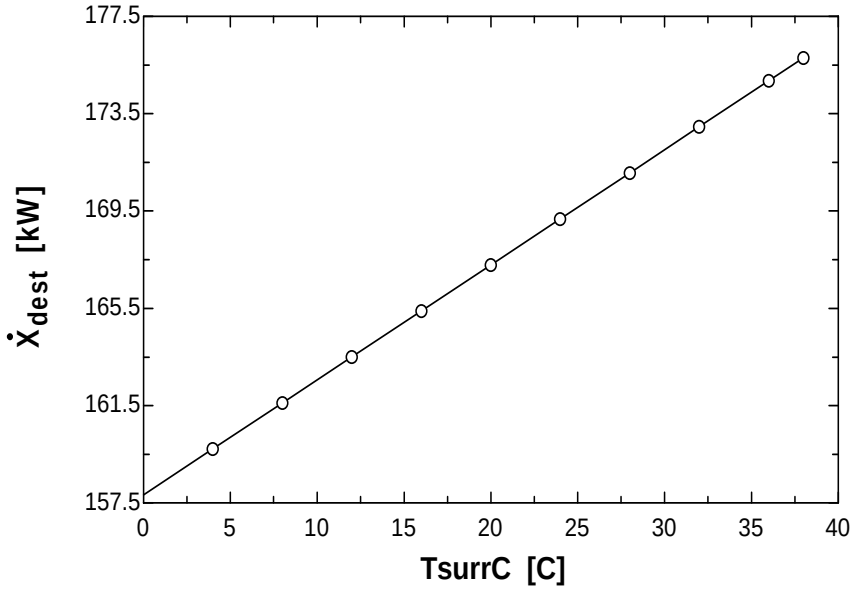
"Entopy of the products per kilomle of fuel:"
"By Dalton's law the partial pressures of the product gases is the product
of the mole fraction and P_prod"
N_prod = 3 + 4 + (1+Ex)*A_th*3.76 + Ex*A_th  "total kmol of products"
P_O2_prod = Ex*A_th/N_prod*P_prod "Patrial pressure O2 in products"
s_O2_prod=entropy(O2,T=T_prod,P=P_O2_prod)
P_N2_prod = (1+Ex)*A_th*3.76/N_prod*P_prod "Patrial pressure N2 in products"
s_N2_prod=entropy(N2,T=T_prod,P=P_N2_prod)
P_CO2_prod = 3/N_prod*P_prod       "Patrial pressure CO2 in products"
s_CO2_prod=entropy(CO2, T=T_prod,P=P_CO2_prod)
P_H2O_prod = 4/N_prod*P_prod      "Patrial pressure H2O in products"
s_H2O_prod=entropy(H2O, T=T_prod,P=P_H2O_prod)

SP = 3*s_CO2_prod + 4*s_H2O_prod + (1+Ex)*A_th*3.76*s_N2_prod + Ex*A_th*s_O2_prod

"Since Q_out is the heat rejected to the surroundings per kilomole fuel,
the entropy of the surroundings is:"
S_surr = Q_out/T_surr

"Rate of entropy generation:"
S_dot_gen = (SP - SR +S_surr)"kJ/kmol_fuel"/(Mw_C3H8 "kg/kmol_fuel")*m_dot_fuel"kg/s"  "kW/K"
X_dot_dest = T_surr*S_dot_gen"[kW]"
```

TsurrC [C]	X_{dest} [kW]
0	157.8
4	159.7
8	161.6
12	163.5
16	165.4
20	167.3
24	169.2
28	171.1
32	173
36	174.9
38	175.8



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15-94 Liquid octane is burned steadily with 70 percent excess air. The entropy generation and exergy destruction per unit mass of the fuel are to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

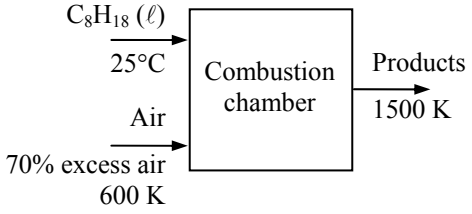
Properties The molar masses of C_8H_{18} and air are 114 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol C_8H_{18}, the combustion equation can be written as

$$C_8H_{18}(\ell) + 1.7a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 0.7a_{th}O_2 + (1.7)(3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.7a_{th} = 8 + 4.5 + 0.7a_{th} \longrightarrow a_{th} = 12.5$$



Thus,

$$C_8H_{18}(\ell) + 21.25(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 8.75O_2 + 79.9N_2$$

(*b*) Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol	$\bar{h}_{600K}$ kJ/kmol	$\bar{h}_{1500K}$ kJ/kmol
C_8H_{18} (ℓ)	-249,950	---	---	---
O_2	0	8682	17,929	49,292
N_2	0	8669	17,563	47,073
H_2O (*g*)	-241,820	9904	---	57,999
CO_2	-393,520	9364	---	71,078

Thus,

$$\begin{aligned}-Q_{out} &= (8)(-393{,}520 + 71{,}078 - 9364) + (9)(-241{,}820 + 57{,}999 - 9904)\\ &\quad + (8.75)(0 + 49{,}292 - 8682) + (79.9)(0 + 47{,}073 - 8669) - (1)(-249{,}950)\\ &\quad - (21.25)(0 + 17{,}929 - 8682) - (79.9)(0 + 17{,}563 - 8669)\\ &= -1{,}631{,}335 \text{ kJ/kmol of } C_8H_{18}\end{aligned}$$

The entropy generation during this process is determined from

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = \sum N_P\bar{s}_P - \sum N_R\bar{s}_R + \frac{Q_{out}}{T_{surr}}$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of P_m = 600 kPa (=600/101.325=5.92 atm), but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where y_i is the mole fraction of component *i*. Then,

$$S_i = N_i\bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

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The entropy calculations can be presented in tabular form as

	N_i	y_i	$\bar{s}_i^\circ(T,1\text{atm})$	$R_u \ln(y_i P_m)$	$N_i \bar{s}_i$
C_8H_{18}	1	---	466.73	14.79	451.94
O_2	21.25	0.21	226.35	1.81	4771.48
N_2	79.9	0.79	212.07	12.83	15,919.28
					S_R = 21,142.70 kJ/K
CO_2	8	0.0757	292.11	-6.673	2390.26
H_2O (g)	9	0.0852	250.45	-5.690	2305.26
O_2	8.75	0.0828	257.97	-5.928	2309.11
N_2	79.9	0.7563	241.77	12.46	18,321.87
					S_P = 25,326.50 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = 25{,}326.50 - 21{,}142.70 + \frac{1{,}631{,}335}{298} = 9658.1 \text{ kJ/K} \left(\text{per kmol } C_8H_{18}\right)$$

The exergy destruction is

$$X_{\text{dest}} = T_0 S_{\text{gen}} = (298)(9658.1 \text{ kJ/K}) = 2{,}878{,}114 \text{ kJ/K} \left(\text{per kmol } C_8H_{18}\right)$$

The entropy generation and exergy destruction per unit mass of the fuel are

$$S_{\text{gen}} = \frac{\bar{S}_{\text{gen}}}{M_{\text{fuel}}} = \frac{9658.1 \text{ kJ/K} \cdot \text{kmol}}{114 \text{ kg/kmol}} = \mathbf{84.72\ kJ/K \cdot kg\ C_8H_{18}}$$

$$X_{\text{dest}} = \frac{\bar{X}_{\text{dest}}}{M_{\text{fuel}}} = \frac{2{,}878{,}114 \text{ kJ/K} \cdot \text{kmol}}{114 \text{ kg/kmol}} = \mathbf{25{,}250\ kJ/kg\ C_8H_{18}}$$

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15-95 Methyl alcohol is burned steadily with 200 percent excess air in an automobile engine. The maximum amount of work that can be produced by this engine is to be determined.

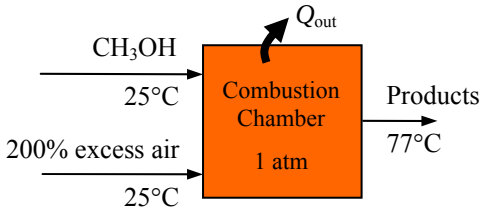
Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol CH_3OH the combustion equation can be written as

$$CH_3OH + 3a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 2a_{th}O_2 + 3a_{th} \times 3.76N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$0.5 + 3a_{th} = 1 + 1 + 2a_{th} \longrightarrow a_{th} = 1.5$$



Thus,

$$CH_3OH + 4.5(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3O_2 + 16.92N_2$$

Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol	$\bar{h}_{350K}$ kJ/kmol
CH_3OH	-200,670	---	---
O_2	0	8682	10,213
N_2	0	8669	10,180
H_2O (g)	-241,820	9904	11,652
CO_2	-393,520	9364	11,351

Thus,

$$\begin{aligned} -Q_{out} &= (1)(-393{,}520 + 11{,}351 - 9364) + (2)(-241{,}820 + 11{,}652 - 9904) \\ &\quad + (3)(0 + 10{,}213 - 8682) + (16.92)(0 + 10{,}180 - 8669) - (1)(-200{,}670) \\ &= -663{,}550 \text{ kJ/kmol of fuel} \end{aligned}$$

The entropy generation during this process is determined from

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{out}}{T_{surr}}$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where y_i is the mole fraction of component i. Then,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

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The entropy calculations can be presented in tabular form as

	N_i	y_i	$\bar{s}_i^\circ(T,1\text{atm})$	$R_u \ln(y_i P_m)$	$N_i \bar{s}_i$
CH_3OH	1	---	239.70	---	239.70
O_2	4.5	0.21	205.04	-12.98	981.09
N_2	16.92	0.79	191.61	-1.960	3275.20
					S_R = 4496 kJ/K
CO_2	1	0.0436	219.831	-26.05	245.88
H_2O (g)	2	0.0873	194.125	-20.27	428.79
O_2	3	0.1309	209.765	-16.91	680.03
N_2	16.92	0.7382	196.173	-2.52	3361.89
					S_P = 4717 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = 4717 - 4496 + \frac{663{,}550}{298} = 2448 \text{ kJ/K (per kmol fuel)}$$

The maximum work is equal to the exergy destruction

$$W_{\max} = X_{\text{dest}} = T_0 S_{\text{gen}} = (298)(2448 \text{ kJ/K}) = 729{,}400 \text{ kJ/K (per kmol fuel)}$$

Per unit mass basis,

$$W_{\max} = \frac{729{,}400 \text{ kJ/K} \cdot \text{kmol}}{32 \text{ kg/kmol}} = \mathbf{22{,}794 \text{ kJ/kg fuel}}$$

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Review Problems

15-96 A sample of a certain fluid is burned in a bomb calorimeter. The heating value of the fuel is to be determined.

Properties The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

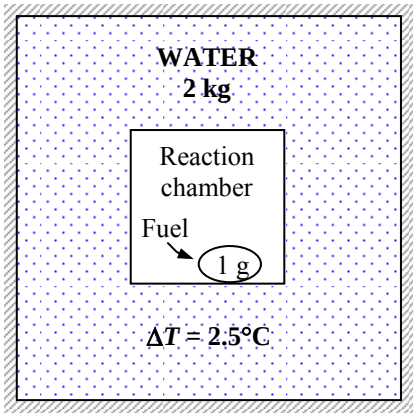
Analysis We take the water as the system, which is a closed system, for which the energy balance on the system $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ with $W = 0$ can be written as

$$Q_{\text{in}} = \Delta U$$

or

$$\begin{aligned} Q_{\text{in}} &= mc\Delta T \\ &= (2\ \text{kg})(4.18\ \text{kJ/kg}\cdot°\text{C})(2.5°\text{C}) \\ &= 20.90\ \text{kJ}\ (\text{per gram of fuel}) \end{aligned}$$

Therefore, heat transfer per kg of the fuel would be **20,900 kJ/kg fuel**. Disregarding the slight energy stored in the gases of the combustion chamber, this value corresponds to the heating value of the fuel.



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15-97E Hydrogen is burned with 100 percent excess air. The AF ratio and the volume flow rate of air are to be determined.

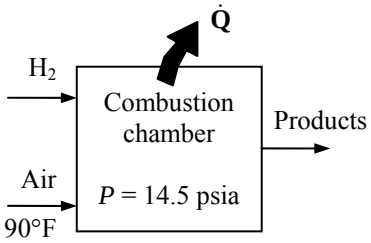
Assumptions **1** Combustion is complete. **2** Air and the combustion gases are ideal gases.

Properties The molar masses of H_2 and air are 2 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis (*a*) The combustion is complete, and thus products will contain only H_2O, O_2 and N_2. The moisture in the air does not react with anything; it simply shows up as additional H_2O in the products. Therefore, for simplicity, we will balance the combustion equation using dry air, and then add the moisture to both sides of the equation. The combustion equation in this case can be written as

$$H_2 + 2a_{th}(O_2 + 3.76N_2) \longrightarrow H_2O + a_{th}O_2 + (2)(3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient for air. It is determined from



O_2 balance: $2a_{th} = 0.5 + a_{th} \longrightarrow a_{th} = 0.5$

Substituting, $H_2 + (O_2 + 3.76N_2) \longrightarrow H_2O + 0.5O_2 + 3.76N_2$

Therefore, 4.76 lbmol of dry air will be used per kmol of the fuel.
The partial pressure of the water vapor present in the incoming air is

$$P_{v,\text{in}} = \phi_{\text{air}} P_{\text{sat@90°F}} = (0.60)(0.69904 \text{ psi}) = 0.419 \text{ psia}$$

The number of moles of the moisture that accompanies 4.76 lbmol of incoming dry air ($N_{v,\text{ in}}$) is determined to be

$$N_{v,\text{in}} = \left(\frac{P_{v,\text{in}}}{P_{\text{total}}}\right) N_{\text{total}} = \left(\frac{0.419 \text{ psia}}{14.5 \text{ psia}}\right)(4.76 + N_{v,\text{in}}) \longrightarrow N_{v,\text{in}} = 0.142 \text{ lbmol}$$

The balanced combustion equation is obtained by substituting the coefficients determined earlier and adding 0.142 lbmol of H_2O to both sides of the equation,

$$H_2 + (O_2 + 3.76N_2) + 0.142H_2O \longrightarrow 1.142H_2O + 0.5O_2 + 3.76N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(4.76 \text{ lbmol})(29 \text{ lbm/lbmol}) + (0.142 \text{ lbmol})(18 \text{ lbm/lbmol})}{(1 \text{ lbmol})(2 \text{ lbm/lbmol})} = \mathbf{70.3 \text{ lbm air/lbmfuel}}$$

(*b*) The mass flow rate of H_2 is given to be 10 lbm/h. Since we need 70.3 lbm air per lbm of H_2, the required mass flow rate of air is

$$\dot{m}_{\text{air}} = (\text{AF})(\dot{m}_{\text{fuel}}) = (70.3)(25 \text{ lbm/h}) = 1758 \text{ lbm/h}$$

The mole fractions of water vapor and the dry air in the incoming air are

$$y_{H_2O} = \frac{N_{H_2O}}{N_{\text{total}}} = \frac{0.142}{4.76 + 0.142} = 0.029 \text{ and } y_{\text{dryair}} = 1 - 0.029 = 0.971$$

Thus,

$$M = (yM)_{H_2O} + (yM)_{\text{dryair}} = (0.029)(18) + (0.971)(29) = 28.7 \text{ lbm/lbmol}$$

$$v = \frac{RT}{P} = \frac{(10.73/28.7 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(550 \text{ R})}{14.5 \text{ psia}} = 14.18 \text{ ft}^3/\text{lbm}$$

$$\dot{V} = \dot{m}v = (1758 \text{ lbm/h})(14.18 \text{ ft}^3/\text{lbm}) = \mathbf{24{,}928 \text{ ft}^3/\text{h}}$$

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15-98 A gaseous fuel with a known composition is burned with dry air, and the volumetric analysis of products gases is determined. The AF ratio, the percent theoretical air used, and the volume flow rate of air are to be determined.

Assumptions **1** Combustion is complete. **2** Air and the combustion gases are ideal gases.

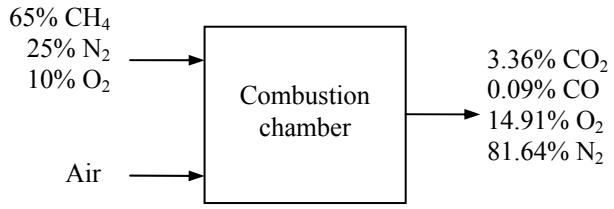
Properties The molar masses of C, H_2, N_2, O_2, and air are 12, 2, 28, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis Considering 100 kmol of dry products, the combustion equation can be written as

$$x(0.65\text{CH}_4 + 0.25\text{N}_2 + 0.10\text{O}_2) + a(\text{O}_2 + 3.76\text{N}_2) \longrightarrow 3.36\text{CO}_2 + 0.09\text{CO} + 14.91\text{O}_2 + 81.64\text{N}_2 + b\text{H}_2\text{O}$$

The unknown coefficients *x*, *a*, and *b* are determined from mass balances,

$$\text{C}: \ 0.65x = 3.36 + 0.09 \longrightarrow x = 5.31$$
$$\text{H}: \ (4)(0.65)x = 2b \longrightarrow b = 6.90$$
$$\text{N}_2: \ 0.25x + 3.76a = 81.64 \longrightarrow a = 21.36$$



$$\left[\text{Check } \text{O}_2: \ 0.10x + a = 3.36 + 0.045 + 14.91 + b/2 \longrightarrow a = 21.23\right]$$

The N_2 balance and O_2 balance gives two different *a* values. There must be a small error in the volumetric analysis of the products and the mass balance is not completely satisfied. Yet we solve the problem with an *a* value of 21.36 being aware of this situation. Then,

$$5.31(0.65\text{CH}_4 + 0.25\text{N}_2 + 0.10\text{O}_2) + 21.36(\text{O}_2 + 3.76\text{N}_2) \longrightarrow 3.36\text{CO}_2 + 0.09\text{CO} + 14.91\text{O}_2 + 81.64\text{N}_2 + 6.9\text{H}_2\text{O}$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 5.31,

$$(0.65\text{CH}_4 + 0.25\text{N}_2 + 0.10\text{O}_2) + 4.02(\text{O}_2 + 3.76\text{N}_2) \longrightarrow 0.633\text{CO}_2 + 0.017\text{CO} + 2.81\text{O}_2 + 15.37\text{N}_2 + 1.3\text{H}_2\text{O}$$

(*a*) The air-fuel ratio is determined from its definition,

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(4.02 \times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{0.65 \times 16 + 0.25 \times 28 + 0.10 \times 32} = \mathbf{26.9 \ kg \ air/kg \ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$(0.65\text{CH}_4 + 0.25\text{N}_2 + 0.10\text{O}_2) + a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2) \longrightarrow 0.65\text{CO}_2 + 1.3\text{H}_2\text{O} + (0.25 + 3.76a_{\text{th}})\text{N}_2$$

$$\text{O}_2: \ 0.10 + a_{\text{th}} = 0.65 + 0.65 \longrightarrow a_{\text{th}} = 1.2$$

Then, $$\text{Percent theoretical air} = \frac{m_{\text{air,act}}}{m_{\text{air,th}}} = \frac{N_{\text{air,act}}}{N_{\text{air,th}}} = \frac{(4.02)(4.76) \text{ kmol}}{(1.2)(4.76) \text{ kmol}} = 3.35 = \mathbf{335\%}$$

(*c*) The specific volume, mass flow rate, and the volume flow rate of air at the inlet conditions are

$$v = \frac{RT}{P} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})}{100 \text{ kPa}} = 0.855 \text{ m}^3/\text{kg}$$

$$\dot{m}_{\text{air}} = (\text{AF})\dot{m}_{\text{fuel}} = (26.9 \text{ kg air/kg fuel})(3.5 \text{ kg fuel/min}) = 94.15 \text{ m}^3/\text{min}$$

$$\dot{V}_{\text{air}} = (\dot{m}v)_{\text{air}} = (94.15 \text{ kg/min})(0.855 \text{ m}^3/\text{kg}) = \mathbf{80.5 \ m^3/min}$$

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15-99E Propane is burned with stoichiometric amount of air. The fraction of the water in the products that is vapor is to be determined.

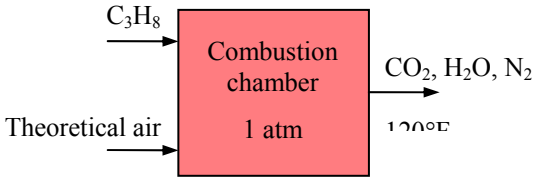
Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases.

Analysis The fuel is burned completely with the air, and thus the products will contain only CO_2, H_2O, and N_2. Considering 1 kmol C_3H_8, the combustion equation can be written as

$$C_3H_8 + 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$

The mole fraction of water in the products is

$$y = \frac{N_{H2O}}{N_{prod}} = \frac{4\,\text{kmol}}{(3+4+18.8)\text{kmol}} = 0.1550$$



The saturation pressure for the water vapor is

$$P_v = P_{sat@120°F} = 1.6951\,\text{psia}$$

When the combustion gases are saturated, the mole fraction of the water vapor will be

$$y_g = \frac{P_v}{P} = \frac{1.6951\,\text{kPa}}{14.696\,\text{kPa}} = 0.1153$$

Thus, the fraction of water vapor in the combustion products is

$$f_{vapor} = \frac{y_g}{y} = \frac{0.1153}{0.1550} = \mathbf{0.744}$$

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15-100 Coal whose mass percentages are specified is burned with 20% excess air. The dew-point temperature of the products is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, SO_2, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, H_2, O_2, S, and air are 12, 2, 32, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis We consider 100 kg of coal for simplicity. Noting that the mass percentages in this case correspond to the masses of the constituents, the mole numbers of the constituent of the coal are determined to be

$$N_{\text{C}} = \frac{m_{\text{C}}}{M_{\text{C}}} = \frac{61.40\text{ kg}}{12\text{ kg/kmol}} = 5.117\text{ kmol}$$

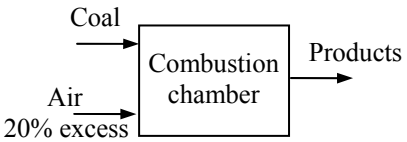
$$N_{\text{H2}} = \frac{m_{\text{H2}}}{M_{\text{H2}}} = \frac{5.79\text{ kg}}{2\text{ kg/kmol}} = 2.895\text{ kmol}$$

$$N_{\text{O2}} = \frac{m_{\text{O2}}}{M_{\text{O2}}} = \frac{25.31\text{ kg}}{32\text{ kg/kmol}} = 0.7909\text{ kmol}$$

$$N_{\text{N2}} = \frac{m_{\text{N2}}}{M_{\text{N2}}} = \frac{1.09\text{ kg}}{28\text{ kg/kmol}} = 0.03893\text{ kmol}$$

$$N_{\text{S}} = \frac{m_{\text{S}}}{M_{\text{S}}} = \frac{1.41\text{ kg}}{32\text{ kg/kmol}} = 0.04406\text{ kmol}$$

61.40% C
5.79% H_2
25.31% O_2
1.09% N_2
1.41% S
5.00% ash
(by mass)



The mole number of the mixture and the mole fractions are

$$N_m = 5.117 + 2.895 + 0.7909 + 0.03893 + 0.04406 = 8.886\text{ kmol}$$

$$y_{\text{C}} = \frac{N_{\text{C}}}{N_m} = \frac{5.117\text{ kmol}}{8.886\text{ kmol}} = 0.5758$$

$$y_{\text{H2}} = \frac{N_{\text{H2}}}{N_m} = \frac{2.895\text{ kmol}}{8.886\text{ kmol}} = 0.3258$$

$$y_{\text{O2}} = \frac{N_{\text{O2}}}{N_m} = \frac{0.7909\text{ kmol}}{8.886\text{ kmol}} = 0.0890$$

$$y_{\text{N2}} = \frac{N_{\text{N2}}}{N_m} = \frac{0.03893\text{ kmol}}{8.886\text{ kmol}} = 0.00438$$

$$y_{\text{S}} = \frac{N_{\text{S}}}{N_m} = \frac{0.04406\text{ kmol}}{8.886\text{ kmol}} = 0.00496$$

Ash consists of the non-combustible matter in coal. Therefore, the mass of ash content that enters the combustion chamber is equal to the mass content that leaves. Disregarding this non-reacting component for simplicity, the combustion equation may be written as

$$0.5758\text{C} + 0.3258\text{H}_2 + 0.0890\text{O}_2 + 0.00438\text{N}_2 + 0.00496\text{S} + 1.25a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.5758\text{CO}_2 + 0.3258\text{H}_2\text{O} + 0.00496\text{SO}_2 + 0.25a_{\text{th}}\text{O}_2 + 1.25a_{\text{th}} \times 3.76\text{N}_2$$

According to the oxygen balance,

$$\text{O}_2\text{ balance}: \quad 0.0890 + 1.25a_{\text{th}} = 0.5758 + 0.5 \times 0.3258 + 0.00496 + 0.25a_{\text{th}} \longrightarrow a_{\text{th}} = 0.6547$$

Substituting,

$$0.5758\text{C} + 0.3258\text{H}_2 + 0.0890\text{O}_2 + 0.00438\text{N}_2 + 0.00496\text{S} + 0.8184(\text{O}_2 + 3.76\text{N}_2)$$
$$\longrightarrow 0.5758\text{CO}_2 + 0.3258\text{H}_2\text{O} + 0.00496\text{SO}_2 + 0.1637\text{O}_2 + 3.077\text{N}_2$$

The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{\text{prod}}}\right)P_{\text{prod}} = \left(\frac{0.3258\text{ kmol}}{(0.5758 + 0.3258 + 0.00496 + 0.1637 + 3.077)\text{ kmol}}\right)(101.3\text{ kPa}) = 7.96\text{ kPa}$$

Thus, $T_{\text{dp}} = T_{\text{sat@7.96 kPa}} = \mathbf{41.3°C}$ (Table A-5)

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15-101 Methane is burned steadily with 50 percent excess air. The dew-point temperature of the water vapor in the products is to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases.

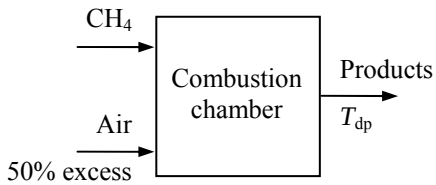
Properties The molar masses of CH_4 and air are 16 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol CH_4, the combustion equation can be written as

$$CH_4 + 1.5a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 0.5a_{th}O_2 + (1.5)(3.76)a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.5a_{th} = 1 + 1 + 0.5a_{th} \longrightarrow a_{th} = 2$$



Thus,

$$CH_4 + 3(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + O_2 + 11.28N_2$$

The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{prod}}\right)P_{prod} = \left(\frac{2\text{ kmol}}{(1+2+1+11.28)\text{ kmol}}\right)(101.325\text{ kPa}) = 13.26\text{ kPa}$$

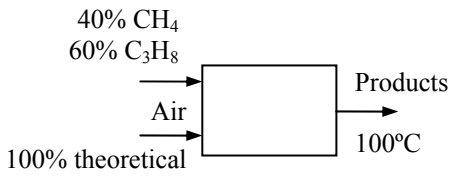
Thus,

$$T_{dp} = T_{sat@13.26\,kPa} = \mathbf{51.4°C} \qquad \text{(from EES)}$$

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15-102 A mixture of 40% by volume methane, CH4, and 60% by volume propane, C3H8, is burned completely with theoretical air. The amount of water formed during combustion process that will be condensed is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2 only.



Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion equation in this case can be written as

$$0.4\,CH_4 + 0.6\,C_3H_8 + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + F\,N_2$$

where a_{th} is the stoichiometric coefficient for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

Carbon balance: $B = 0.4 + 3\times 0.6 = 2.2$

Hydrogen balance: $2D = 4\times 0.4 + 8\times 0.6 = 2D \longrightarrow D = 3.2$

Oxygen balance: $2a_{th} = 2B + D \longrightarrow 2a_{th} = 2(2.2) + 3.2 \longrightarrow a_{th} = 3.8$

Nitrogen balance: $3.76a_{th} = F \longrightarrow 3.76(3.8) = F \longrightarrow F = 14.29$

Then, we write the balanced reaction equation as

$$0.4\,CH_4 + 0.6\,C_3H_8 + 3.8\left[O_2 + 3.76N_2\right] \longrightarrow 2.2\,CO_2 + 3.2\,H_2O + 14.29\,N_2$$

The vapor mole fraction in the products is

$$y_v = \frac{3.2}{2.2 + 3.2 + 14.29} = 0.1625$$

The partial pressure of water in the products is

$$P_{v,prod} = y_v P_{prod} = (0.1625)(100\,kPa) = 16.25\,kPa$$

The dew point temperature of the products is

$$T_{dp} = T_{sat@16.25\,kPa} = 55.64°C$$

The partial pressure of the water vapor remaining in the products at the product temperature is

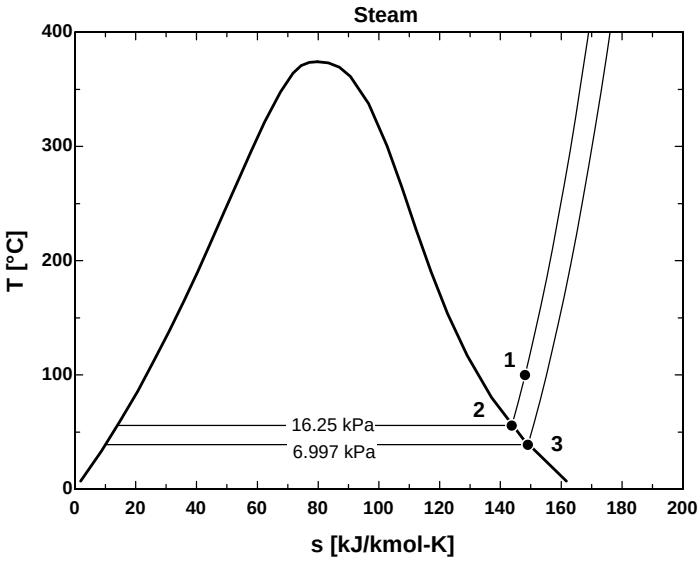
$$P_v = P_{sat@39°C} = 7.0\,kPa$$

The kmol of water vapor in the products at the product temperature is

$$P_v = \frac{N_v}{N_{total,product}} P_{prod}$$

$$7.0\,kPa = \frac{N_v}{2.2 + N_v + 14.29}$$

$$N_v = 1.241\,kmol$$



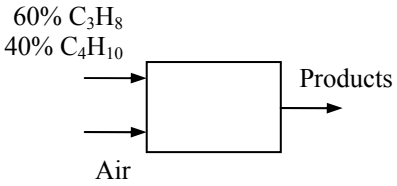
The kmol of water condensed is

$$N_w = 3.2 - 1.241 = \mathbf{1.96\ kmol\ water/kmol\ fuel}$$

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15-103 A gaseous fuel mixture of 60% propane, C_3H_8, and 40% butane, C_4H_{10}, on a volume basis is burned with an air-fuel ratio of 25. The moles of nitrogen in the air supplied to the combustion process, the moles of water formed in the combustion process, and the moles of oxygen in the product gases are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2 only.



Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The theoretical combustion equation in this case can be written as

$$0.6\,C_3H_8 + 0.4\,C_4H_{10} + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + F\,N_2$$

where a_{th} is the stoichiometric coefficient for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

Carbon balance: $B = 3 \times 0.6 + 4 \times 0.4 = 3.4$

Hydrogen balance: $8 \times 0.6 + 10 \times 0.4 = 2D \longrightarrow D = 4.4$

Oxygen balance: $2a_{th} = 2B + D \longrightarrow 2a_{th} = 2 \times 3.4 + 4.4 \longrightarrow a_{th} = 5.6$

Nitrogen balance: $3.76a_{th} = F \longrightarrow 3.76 \times 5.6 = F \longrightarrow F = 21.06$

Then, we write the balanced theoretical reaction equation as

$$0.6\,C_3H_8 + 0.4\,C_4H_{10} + 5.6\left[O_2 + 3.76N_2\right] \longrightarrow 3.4\,CO_2 + 4.4\,H_2O + 21.06\,N_2$$

The air-fuel ratio for the theoretical reaction is determined from

$$AF_{th} = \frac{m_{air}}{m_{fuel}} = \frac{(5.6 \times 4.76\ \text{kmol})(29\ \text{kg/kmol})}{(0.6 \times 44 + 0.4 \times 58)\ \text{kg}} = 15.59\ \text{kg air/kg fuel}$$

The percent theoretical air is

$$\text{Percent}TH_{air} = \frac{AF_{actual}}{AF_{th}} = \frac{25}{15.59} \times 100 = 160.4\%$$

The moles of nitrogen supplied is

$$N_{N2} = \frac{\text{Percent}TH_{air}}{100} \times a_{th} \times 3.76 = \frac{160.4}{100}(5.6)(3.76) = \mathbf{33.8\ kmol}\ \text{per kmol fuel}$$

The moles of water formed in the combustion process is

$$N_{H2O} = D = \mathbf{4.4\ kmol}\ \text{per kmol fuel}$$

The moles of oxygen in the product gases is

$$N_{O2} = \left(\frac{\text{Percent}TH_{air}}{100} - 1\right)a_{th} = \left(\frac{160.4}{100} - 1\right)(5.6) = \mathbf{3.38\ kmol}\ \text{per kmol fuel}$$

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15-104 Ethane is completely burned with air. Various parameters are to be determined for the given reaction.

Assumptions The water in the products is in the vapor phase.

Analysis (a) The reaction equation is given as

$$C_2H_6 + 4.788[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 3H_2O + 1.288O_2 + 18N_2$$

The partial pressure of water vapor is

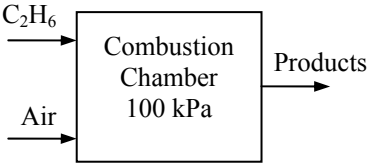
$$P_v = \frac{N_{H2O}}{N_{total}} P_{total} = \frac{3}{2+3+1.288+18}(100\text{ kPa}) = \frac{3\text{ kmol}}{24.288\text{ kmol}}(100\text{ kPa}) = 12.35\text{ kPa}$$

The dew point temperature of the product gases is the saturation temperature of water at this pressure:

$$T_{dp} = T_{sat@12.35\,kPa} = \mathbf{49.6°C} \quad \text{(Table A-5)}$$

(b) The partial pressure of oxygen is

$$P_{O2} = \frac{N_{O2}}{N_{total}} P_{total} = \frac{1.288\text{ kmol}}{24.288\text{ kmol}}(100\text{ kPa}) = 5.303\text{ kPa}$$



The specific volume of oxygen is then,

$$v_{O2} = \frac{R_{O2}T}{P_{O2}} = \frac{(0.2598\text{ kJ/kmol}\cdot\text{K})(373\text{ K})}{5.303\text{ kPa}} = \mathbf{18.3\ m^3/kg}$$

(c) The combustion reaction with stoichiometric air is

$$C_2H_6 + 3.5(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 3H_2O + 3.5\times 3.76N_2$$

Both the reactants and the products are taken to be at the standard reference state of 25°C and 1 atm for the calculation of heating values. The heat transfer for this process is equal to enthalpy of combustion. Note that N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then,

$$q = h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO2} + \left(N\bar{h}^\circ_f\right)_{H2O} - \left(N\bar{h}^\circ_f\right)_{C2H6}$$

For the LHV, the water in the products is taken to be vapor. Then,

$$h_C = (2\text{ kmol})(-393{,}520\text{ kJ/kmol}) + (3\text{ kmol})(-241{,}820\text{ kJ/kmol}) - (1\text{ kmol})(-84{,}680\text{ kJ/kmol})$$
$$= -1{,}427{,}820\text{ kJ/kmol ethane}$$

The LHV per unit kmol of the fuel is the negative of the enthalpy of combustion:

$$\text{LHV} = -h_C = \mathbf{1{,}427{,}820\ kJ/kmol\ C_2H_6}$$

(d) The average molar mass of the product gas is

$$M = \frac{N_{CO2}M_{CO2} + N_{H2O}M_{H2O} + N_{O2}M_{O2} + N_{N2}M_{N2}}{N_{total}}$$
$$= \frac{(2\text{ kmol})(44\text{ kg/kmol}) + (3\text{ kmol})(18\text{ kg/kmol}) + (1.288\text{ kmol})(32\text{ kg/kmol}) + (18\text{ kmol})(28\text{ kg/kmol})}{24.288\text{ kmol}}$$
$$= \frac{687.2\text{ kmol}}{24.288\text{ kmol}} = \mathbf{28.29\ kg/kmol}$$

(e) The average molar constant pressure specific heat of the product gas is

$$\bar{c}_p = \frac{N_{CO2}\bar{c}_{p,CO2} + N_{H2O}\bar{c}_{p,H2O} + N_{O2}\bar{c}_{p,O2} + N_{N2}\bar{c}_{p,N2}}{N_{total}}$$
$$= \frac{(2\times 41.16 + 3\times 34.28 + 1.288\times 30.14 + 18\times 29.27)\text{ kJ/K}}{24.288\text{ kmol}} = \frac{750.8\text{ kJ/K}}{24.288\text{ kmol}} = \mathbf{30.91\,kJ/kmol\cdot K}$$

(f) The air-fuel mass ratio is

$$\text{AF} = \frac{m_{air}}{m_{fuel}} = \frac{(4.788\times 4.76\times 29)\text{ kg}}{(1\times 30)\text{ kg}} = \frac{660.9\text{ kg}}{30\text{ kg}} = \mathbf{22.03\ kg\ air/kg\ fuel}$$

(g) For a molar fuel flow rate is 0.1 kmol/min, the mass flow rate of water in the product gases is

$$\dot{m}_{H2O} = \dot{N}_{C2H6}\frac{N_{H2O}}{N_{C2H6}}M_{H2O} = (0.1\text{ kmol/min})\frac{3\text{ kmol}}{1\text{ kmol}}(18\text{ kg/kmol}) = \mathbf{5.4\ kg/min}$$

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15-105 CO gas is burned with air during a steady-flow combustion process. The rate of heat transfer from the combustion chamber is to be determined.

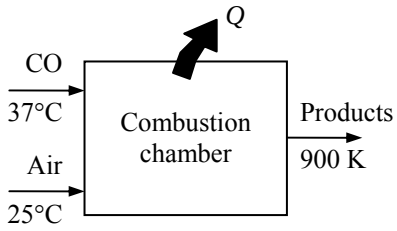
Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** Combustion is complete.

Properties The molar masses of CO and air are 28 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis We first need to calculate the amount of air used per kmol of CO before we can write the combustion equation,

$$\boldsymbol{\nu}_{\text{CO}} = \frac{RT}{P} = \frac{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(310\ \text{K})}{(110\ \text{kPa})} = 0.836\ \text{m}^3/\text{kg}$$

$$\dot{m}_{\text{CO}} = \frac{\dot{\boldsymbol{V}}_{\text{CO}}}{\boldsymbol{\nu}_{\text{CO}}} = \frac{0.4\ \text{m}^3/\text{min}}{0.836\ \text{m}^3/\text{kg}} = 0.478\ \text{kg/min}$$



Then the molar air-fuel ratio becomes

$$\overline{\text{AF}} = \frac{N_{\text{air}}}{N_{\text{fuel}}} = \frac{\dot{m}_{\text{air}}/M_{\text{air}}}{\dot{m}_{\text{fuel}}/M_{\text{fuel}}} = \frac{(1.5\ \text{kg/min})/(29\ \text{kg/kmol})}{(0.478\ \text{kg/min})/(28\ \text{kg/kmol})} = 3.03\ \text{kmol air/kmol fuel}$$

Thus the number of moles of O_2 used per mole of CO is 3.03/4.76 = 0.637. Then the combustion equation in this case can be written as

$$\text{CO} + 0.637(\text{O}_2 + 3.76\text{N}_2) \longrightarrow \text{CO}_2 + 0.137\text{O}_2 + 2.40\text{N}_2$$

Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{\text{out}} = \sum N_P(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_P - \sum N_R(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ \text{K}}$ kJ/kmol	$\bar{h}_{310\ \text{K}}$ kJ/kmol	$\bar{h}_{900\ \text{K}}$ kJ/kmol
CO	-110,530	8669	9014	27,066
O_2	0	8682	---	27,928
N_2	0	8669	---	26,890
CO_2	-393,520	9364	---	37,405

Thus,

$$\begin{aligned}-Q_{\text{out}} &= (1)(-393{,}520 + 37{,}405 - 9364) + (0.137)(0 + 27{,}928 - 8682) \\ &\quad + (2.4)(0 + 26{,}890 - 8669) - (1)(-110{,}530 + 9014 - 8669) - 0 - 0 \\ &= -208{,}927\ \text{kJ/kmol of CO}\end{aligned}$$

Then the rate of heat transfer for a mass flow rate of 0.956 kg/min for CO becomes

$$\dot{Q}_{\text{out}} = \dot{N}Q_{\text{out}} = \left(\frac{\dot{m}}{N}\right)Q_{\text{out}} = \left(\frac{0.478\ \text{kg/min}}{28\ \text{kg/kmol}}\right)(208{,}927\ \text{kJ/kmol}) = \mathbf{3567\ kJ/min}$$

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15-106 Ethanol gas is burned with 10% excess air. The combustion is incomplete. The theoretical kmols of oxygen in the reactants, the balanced chemical reaction, and the rate of heat transfer are to be determined.

Assumptions **1** Combustion is incomplete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

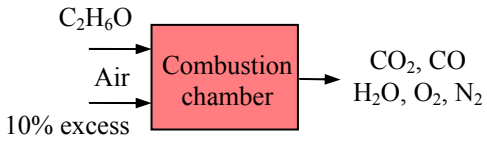
Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis (a) The balanced reaction equation for stoichiometric air is

$$C_2H_6O + a_{th}[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3\,H_2O + a_{th} \times 3.76\,N_2$$

The stoicihiometric coefficient a_{th} is determined from an O_2 balance:

$$0.5 + a_{th} = 2 + 1.5 \longrightarrow a_{th} = 3$$



Substituting,

$$C_2H_6O + 3[O_2 + 3.76N_2] \longrightarrow 2\,CO_2 + 3\,H_2O + 11.28\,N_2$$

Therefore, **3 kmol of oxygen** is required to burn 1 kmol of ethanol.

(b) The reaction with 10% excess air and incomplete combustion can be written as

$$C_2H_6O + 1.1 \times 3[O_2 + 3.76N_2] \longrightarrow 2(0.9\,CO_2 + 0.1\,CO) + 3\,H_2O + x\,O_2 + 1.1 \times 3 \times 3.76\,N_2$$

The coefficient for O_2 is determined from a mass balance,

O_2 balance: $$0.5 + 1.1 \times 3 = 0.9 \times 2 + 0.5 \times (0.1 \times 2) + 0.5 \times 3 + x \longrightarrow x = 0.4$$

Substituting,

$$C_2H_6O + 3.3[O_2 + 3.76N_2] \longrightarrow 1.8\,CO_2 + 0.2\,CO + 3\,H_2O + 0.4\,O_2 + 12.408\,N_2$$

(b) The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Both the reactants and products are at 25 °C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. Then, using the values given in the table,

$$\begin{aligned} -Q_{out} &= (1.8)(-393{,}520) + (0.2)(-110{,}530) + (3)(-241{,}820) - (1)(-235{,}310) \\ &= -1{,}220{,}590 \text{ kJ/kmol fuel} \end{aligned}$$

or

$$Q_{out} = 1{,}220{,}590 \text{ kJ/kmol fuel}$$

For a 3.5 kg/h of fuel burned, the rate of heat transfer is

$$\dot{Q}_{out} = \dot{N}Q_{out} = \left(\frac{\dot{m}}{M}\right)Q_{out} = \frac{3.5 \text{ kg/h}}{46 \text{ kg/kmol}}(1{,}220{,}590 \text{ kJ/kmol}) = 92{,}870 \text{ kJ/h} = \mathbf{25.80\ kW}$$

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15-107 Propane gas is burned with air during a steady-flow combustion process. The adiabatic flame temperature is to be determined for different cases.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

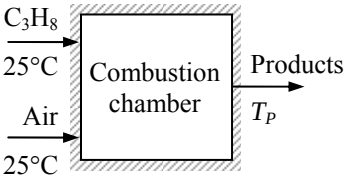
Analysis Adiabatic flame temperature is the temperature at which the products leave the combustion chamber under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$). Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P\left(\bar{h}_f^\circ + \bar{h}_T - \bar{h}^\circ\right)_P = \left(N\bar{h}_f^\circ\right)_{C_3H_8}$$

since all the reactants are at the standard reference temperature of 25°C, and $\bar{h}_f^\circ = 0$ for O_2 and N_2.

(*a*) The theoretical combustion equation of C_3H_8 with stoichiometric amount of air is

$$C_3H_8(g) + 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$



From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ \text{K}}$ kJ/kmol
C_3H_8 (*g*)	-103,850	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO	-110,530	8669
CO_2	-393,520	9364

Thus,

$$(3)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (4)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (18.8)(0 + \bar{h}_{N_2} - 8669) = (1)(-103{,}850)$$

It yields

$$3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.8\bar{h}_{N_2} = 2{,}274{,}675 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,274,675 / (3 + 4 + 18.8) = 88,165 kJ/kmol. This enthalpy value corresponds to about 2650 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2650 K, but somewhat under it because of the higher specific heats of CO_2 and H_2O.

At 2400 K: $3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.8\bar{h}_{N_2} = (3)(125{,}152) + (4)(103{,}508) + (18.8)(79{,}320)$
$= 2{,}280{,}704$ kJ (Higher than 2,274,675 kJ)

At 2350 K: $3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.8\bar{h}_{N_2} = (3)(122{,}091) + (4)(100{,}846) + (18.8)(77{,}496)$
$= 2{,}226{,}582$ kJ (Lower than 2,274,675 kJ)

By interpolation, T_P = **2394 K**

(*b*) The balanced combustion equation for complete combustion with 200% theoretical air is

$$C_3H_8(g) + 10(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 5O_2 + 37.6N_2$$

Substituting known numerical values,

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$$(3)(-393{,}520+\bar{h}_{CO_2}-9364)+(4)(-241{,}820+\bar{h}_{H_2O}-9904)$$
$$+(5)(0+\bar{h}_{O_2}-8682)+(37.6)(0+\bar{h}_{N_2}-8669)=(1)(-103{,}850)$$

which yields

$$3\bar{h}_{CO_2}+4\bar{h}_{H_2O}+5\bar{h}_{O_2}+37.6\bar{h}_{N_2}=2{,}481{,}060 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,481,060 / (3 + 4 + 5 + 37.6) = 50,021 kJ/kmol. This enthalpy value corresponds to about 1580 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 1580 K, but somewhat under it because of the higher specific heats of CO_2 and H_2O.

At 1540 K:
$$3\bar{h}_{CO_2}+4\bar{h}_{H_2O}+5\bar{h}_{O_2}+37.6\bar{h}_{N_2}=(3)(73{,}417)+(4)(59{,}888)+(5)(50{,}756)+(37.6)(48{,}470)$$
$$=2{,}536{,}055 \text{ kJ (Higher than 2,481,060 kJ)}$$

At 1500 K:
$$3\bar{h}_{CO_2}+4\bar{h}_{H_2O}+5\bar{h}_{O_2}+37.6\bar{h}_{N_2}=(3)(71{,}078)+(4)(57{,}999)+(5)(49{,}292)+(37.6)(47{,}073)$$
$$=2{,}461{,}630 \text{ kJ (Lower than 2,481,060 kJ)}$$

By interpolation, T_P = **1510 K**

(*c*) The balanced combustion equation for incomplete combustion with 95% theoretical air is

$$C_3H_8(g)+4.75(O_2+3.76N_2)\longrightarrow 2.5CO_2+0.5CO+4H_2O+17.86N_2$$

Substituting known numerical values,

$$(2.5)(-393{,}520+\bar{h}_{CO_2}-9364)+(0.5)(-110{,}530+\bar{h}_{CO}-8669)$$
$$+(4)(-241{,}820+\bar{h}_{H_2O}-9904)+(17.86)(0+\bar{h}_{N_2}-8669)=(1)(-103{,}850)$$

which yields

$$2.5\bar{h}_{CO_2}+0.5\bar{h}_{CO}+4\bar{h}_{H_2O}+17.86\bar{h}_{N_2}=2{,}124{,}684 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,124,684 / (2.5 + 4 + 0.5 + 17.86) = 85,466 kJ/kmol. This enthalpy value corresponds to about 2550 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2550 K, but somewhat under it because of the higher specific heats of CO_2 and H_2O.

At 2350 K:

$$2.5\bar{h}_{CO_2}+0.5\bar{h}_{CO}+4\bar{h}_{H_2O}+17.86\bar{h}_{N_2}=(2.5)(122{,}091)+(0.5)(78{,}178)+(4)(100{,}846)+(17.86)(77{,}496)$$
$$=2{,}131{,}779 \text{ kJ (Higher than 2,124,684 kJ)}$$

At 2300 K:

$$2.5\bar{h}_{CO_2}+0.5\bar{h}_{CO}+4\bar{h}_{H_2O}+17.86\bar{h}_{N_2}=(2.5)(119{,}035)+(0.5)(76{,}345)+(4)(98{,}199)+(17.86)(75{,}676)$$
$$=2{,}080{,}129 \text{ kJ (Lower than 2,124,684 kJ)}$$

By interpolation, T_P = **2343 K**

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15-108 The highest possible temperatures that can be obtained when liquid gasoline is burned steadily with air and with pure oxygen are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

Analysis The highest possible temperature that can be achieved during a combustion process is the temperature which occurs when a fuel is burned completely with stoichiometric amount of air in an adiabatic combustion chamber. It is determined from

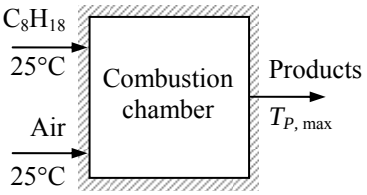
$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P\left(\bar{h}_f^\circ + \bar{h}_T - \bar{h}^\circ\right)_P = \left(N\bar{h}_f^\circ\right)_{C_8H_{18}}$$

since all the reactants are at the standard reference temperature of 25°C, and for O_2 and N_2. The theoretical combustion equation of C_8H_{18} air is

$$C_8H_{18} + 12.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 47N_2$$

From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol
C_8H_{18} (ℓ)	-249,950	---
O_2	0	8682
N_2	0	8669
H_2O (g)	-241,820	9904
CO_2	-393,520	9364



Thus,

$$(8)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (9)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (47)(0 + \bar{h}_{N_2} - 8669) = (1)(-249{,}950)$$

It yields $8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} + 47\bar{h}_{N_2} = 5{,}646{,}081$ kJ

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 5,646,081/(8 + 9 + 47) = 88,220 kJ/kmol. This enthalpy value corresponds to about 2650 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2650 K, but somewhat under it because of the higher specific heat of H_2O.

At 2400 K: $8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} + 47\bar{h}_{N_2} = (8)(125{,}152) + (9)(103{,}508) + (47)(79{,}320)$
$= 5{,}660{,}828$ kJ (Higher than 5,646,081 kJ)

At 2350 K: $8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} + 47\bar{h}_{N_2} = (8)(122{,}091) + (9)(100{,}846) + (47)(77{,}496)$
$= 5{,}526{,}654$ kJ (Lower than 5,646,081 kJ)

By interpolation, T_P = **2395 K**

If the fuel is burned with stoichiometric amount of pure O_2, the combustion equation would be

$$C_8H_{18} + 12.5O_2 \longrightarrow 8CO_2 + 9H_2O$$

Thus, $$(8)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (9)(-241{,}820 + \bar{h}_{H_2O} - 9904) = (1)(-249{,}950)$$

It yields $8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} = 5{,}238{,}638$ kJ

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 5,238,638/(8 + 9) = 308,155 kJ/kmol. This enthalpy value is higher than the highest enthalpy value listed for H_2O and CO_2. Thus an estimate of the adiabatic flame temperature can be obtained by extrapolation.

At 3200 K: $8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} = (8)(174{,}695) + (9)(147{,}457) = 2{,}724{,}673$ kJ

At 3250 K: $8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} = (8)(177{,}822) + (9)(150{,}272) = 2{,}775{,}024$ kJ

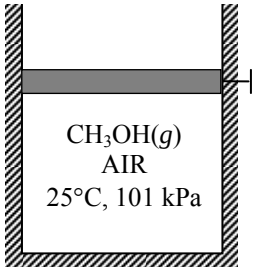
By extrapolation, we get T_P = **3597 K**. However, the solution of this problem using EES gives **5645 K**. The large difference between these two values is due to extrapolation.

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15-109 Methyl alcohol vapor is burned with the stoichiometric amount of air in a combustion chamber. The maximum pressure that can occur in the combustion chamber if the combustion takes place at constant volume and the maximum volume of the combustion chamber if the combustion occurs at constant pressure are to be determined.

Assumptions **1** Combustion is complete. **2** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis (*a*) The combustion equation of $CH_3OH(g)$ with stoichiometric amount of air is



$$CH_3OH + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1 + 2a_{th} = 2 + 2 \quad \longrightarrow \quad a_{th} = 1.5$$

Thus,

$$CH_3OH + 1.5(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 5.64N_2$$

The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$),

$$0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Assuming both the reactants and the products to behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by R_uT. It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298K} - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol
CH_3OH	-200,670	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO_2	-393,520	9364

Thus,

$$(1)(-393{,}520 + \bar{h}_{CO_2} - 9364 - 8.314 \times T_P) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314 \times T_P)$$
$$+ (5.64)(0 + \bar{h}_{N_2} - 8669 - 8.314 \times T_P) = (1)(-200{,}670 - 8.314 \times 298) + (1.5)(0 - 8.314 \times 298)$$
$$+ (5.64)(0 - 8.314 \times 298)$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} - 71.833 \times T_P = 734{,}388 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2850 K:

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} - 71.833 \times T_P = (1)(152{,}908) + (2)(127{,}952) + (5.64)(95{,}859) - (71.833)(2850)$$
$$= 744{,}733 \text{ kJ (Higher than 734,388 kJ)}$$

At 2800 K:

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} - 71.833 \times T_P = (1)(149{,}808) + (2)(125{,}198) + (5.64)(94{,}014) - (71.833)(2800)$$
$$= 729{,}311 \text{ kJ (Lower than 734,388 kJ)}$$

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By interpolation $T_P = 2816$ K

Since both the reactants and the products behave as ideal gases, the final (maximum) pressure that can occur in the combustion chamber is determined to be

$$\frac{P_1 \mathcal{V}}{P_2 \mathcal{V}} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow P_2 = \frac{N_2 T_2}{N_1 T_1} P_1 = \frac{(8.64 \text{ kmol})(2816 \text{ K})}{(8.14 \text{ kmol})(298 \text{ K})}(101 \text{ kPa}) = \mathbf{1013 \text{ kPa}}$$

(*b*) The combustion equation of $CH_3OH(g)$ remains the same in the case of constant pressure. Further, the boundary work in this case can be combined with the *u* terms so that the first law relation can be expressed in terms of enthalpies just like the steady-flow process,

$$Q = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Since both the reactants and the products behave as ideal gases, we have $h = h(T)$. Also noting that $Q = 0$ for an adiabatic combustion process, the 1st law relation reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298 \text{ K}}\right)_P = \sum N_R \left(\bar{h}_f^\circ\right)_R$$

since the reactants are at the standard reference temperature of 25°C. Then using data from the mini table above, we get

$$(1)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (5.64)(0 + \bar{h}_{N_2} - 8669)$$
$$= (1)(-200{,}670) + (1.5)(0) + (5.64)(0)$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} = 754{,}555 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2350 K: $\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} = (1)(122{,}091) + (2)(100{,}846) + (5.64)(77{,}496)$
$= 760{,}860$ kJ (Higher than 754,555 kJ)

At 2300 K: $\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} = (1)(119{,}035) + (2)(98{,}199) + (5.64)(75{,}676)$
$= 742{,}246$ kJ (Lower than 754,555 kJ)

By interpolation, $T_P = 2333$ K

Treating both the reactants and the products as ideal gases, the final (maximum) volume that the combustion chamber can have is determined to be

$$\frac{P \mathcal{V}_1}{P \mathcal{V}_2} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow \mathcal{V}_2 = \frac{N_2 T_2}{N_1 T_1} \mathcal{V}_1 = \frac{(8.64 \text{ kmol})(2333 \text{ K})}{(8.14 \text{ kmol})(298 \text{ K})}(1.5 \text{ L}) = \mathbf{12.5 \text{ L}}$$

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15-110 Problem 15–109 is reconsidered. The effect of the initial volume of the combustion chamber on the maximum pressure of the chamber for constant volume combustion or the maximum volume of the chamber for constant pressure combustion is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
V1=1.5 [L]
T1=(25+273) [K]
P1=101 [kPa]
T0=25+273 [K]

"Properties"
R_u=8.314 "[kJ/kmol-K]"

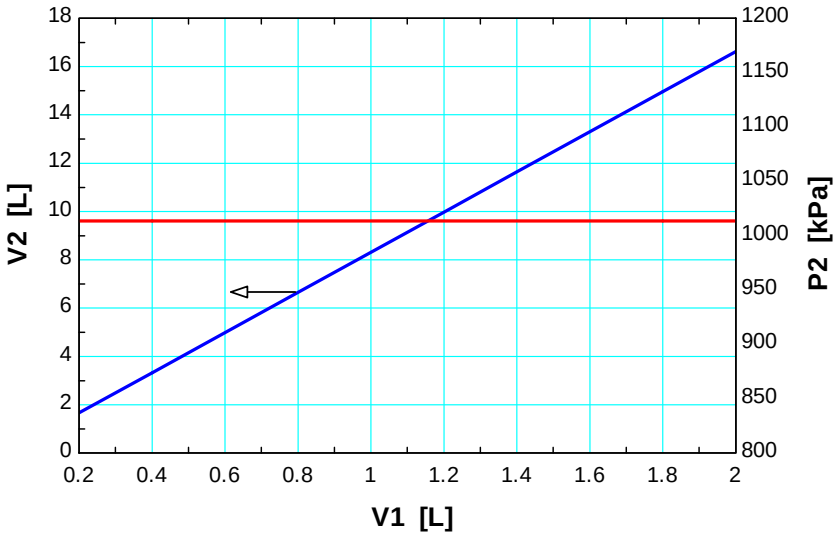
"Analysis"
"The stoichiometric combustion equation is:  CH3OH + a_th (O2+3.76N2) = CO2 + 2 H2O + 3.76*a_th N2"
1+2*a_th=2+2 "O balance"
"Mol numbers of reactants and products in kmol"
N_CH3OH=1
N_O2=a_th
N_N2=a_th*3.76
N_CO2=1
N_H2O=2
"Enthalpy of formation data from Table A-26 in kJ/kmol"
h_f_CH3OH=-200670
"Enthalpies of reactants in kJ/kmol"
h_O2=enthalpy(O2, T=T1)
h_N2_R=enthalpy(N2, T=T1)
"Enthalpies of products in kJ/kmol"
h_N2_P_a=enthalpy(N2, T=T2_a)
h_CO2_a=enthalpy(CO2, T=T2_a)
h_H2O_a=enthalpy(H2O, T=T2_a)

H_P_a=N_CO2*(h_CO2_a-R_u*T2_a)+N_H2O*(h_H2O_a-R_u*T2_a)+N_N2*(h_N2_P_a-R_u*T2_a)
H_R_a=N_CH3OH*(h_f_CH3OH-R_u*T1)+N_O2*(h_O2-R_u*T1)+N_N2*(h_N2_R-R_u*T1)
H_P_a=H_R_a
P2=(N_P/N_R)*(T2_a/T1)*P1 "Final pressure"
N_R=1+4.76*a_th
N_P=1+2+3.76*a_th
"(b)"
"Now ideal gas enthalpies of products are, in kJ/kmol"
h_N2_P_b=enthalpy(N2, T=T2_b)
h_CO2_b=enthalpy(CO2, T=T2_b)
h_H2O_b=enthalpy(H2O, T=T2_b)

H_P_b=N_CO2*h_CO2_b+N_H2O*h_H2O_b+N_N2*h_N2_P_b
H_R_b=N_CH3OH*h_f_CH3OH+N_O2*h_O2+N_N2*h_N2_R
H_P_b=H_R_b
V2=(N_P/N_R)*(T2_b/T1)*V1 "Final pressure"
```

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V1 [L]	V2 [L]	P2 [kPa]
0.2	1.663	1013
0.4	3.325	1013
0.6	4.988	1013
0.8	6.651	1013
1	8.313	1013
1.2	9.976	1013
1.4	11.64	1013
1.6	13.3	1013
1.8	14.96	1013
2	16.63	1013

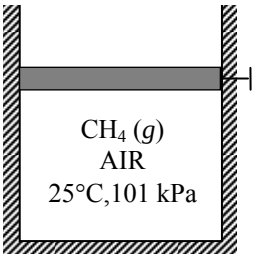


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15-111 Methane is burned with the stoichiometric amount of air in a combustion chamber. The maximum pressure that can occur in the combustion chamber if the combustion takes place at constant volume and the maximum volume of the combustion chamber if the combustion occurs at constant pressure are to be determined.

Assumptions **1** Combustion is complete. **2** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis (*a*) The combustion equation of $CH_4(g)$ with stoichiometric amount of air is



$$CH_4 + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$a_{th} = 1 + 1 \quad \longrightarrow \quad a_{th} = 2$$

Thus,

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 7.52N_2$$

The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$),

$$0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by R_uT. It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298K} - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol
CH_4	-74,850	---
O_2	0	8682
N_2	0	8669
H_2O (g)	-241,820	9904
CO_2	-393,520	9364

Thus,

$$(1)(-393{,}520 + \bar{h}_{CO_2} - 9364 - 8.314 \times T_P) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314 \times T_P)$$
$$+ (7.52)(0 + \bar{h}_{N_2} - 8669 - 8.314 \times T_P) = (1)(-74{,}850 - 8.314 \times 298) + (2)(0 - 8.314 \times 298)$$
$$+ (7.52)(0 - 8.314 \times 298)$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} - 87.463 \times T_P = 870{,}609 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2850 K:

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} - 87.463 \times T_P = (1)(152{,}908) + (2)(127{,}952) + (7.52)(95{,}859) - (87.463)(2850)$$
$$= 880{,}402 \text{ kJ (Higher than 870,609 kJ)}$$

At 2800 K:

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} - 87.463 \times T_P = (1)(149{,}808) + (2)(125{,}198) + (7.52)(94{,}014) - (87.463)(2800)$$
$$= 862{,}293 \text{ kJ (Lower than 870,609 kJ)}$$

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By interpolation, $T_P = 2823$ K

Treating both the reactants and the products as ideal gases, the final (maximum) pressure that can occur in the combustion chamber is determined to be

$$\frac{P_1 \mathcal{V}}{P_2 \mathcal{V}} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow P_2 = \frac{N_2 T_2}{N_1 T_1} P_1 = \frac{(10.52 \text{ kmol})(2823 \text{ K})}{(10.52 \text{ kmol})(298 \text{ K})}(101 \text{ kPa}) = \mathbf{957 \ kPa}$$

(*b*) The combustion equation of $CH_4(g)$ remains the same in the case of constant pressure. Further, the boundary work in this case can be combined with the *u* terms so that the first law relation can be expressed in terms of enthalpies just like the steady-flow process,

$$Q = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Again since both the reactants and the products behave as ideal gases, we have $h = h(T)$. Also noting that $Q = 0$ for an adiabatic combustion process, the energy balance relation reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298 \text{ K}}\right)_P = \sum N_R \left(\bar{h}_f^\circ\right)_R$$

since the reactants are at the standard reference temperature of 25°C. Then using data from the mini table above, we get

$$(1)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (7.52)(0 + \bar{h}_{N_2} - 8669)$$
$$= (1)(-74{,}850) + (2)(0) + (7.52)(0)$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} = 896{,}673 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2350 K: $\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} = (1)(122{,}091) + (2)(100{,}846) + (7.52)(77{,}496)$
$= 906{,}553$ kJ (Higher than 896,673 kJ)

At 2300 K: $\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} = (1)(119{,}035) + (2)(98{,}199) + (7.52)(75{,}676)$
$= 884{,}517$ kJ (Lower than 896,673 kJ)

By interpolation, $T_P = 2328$ K

Treating both the reactants and the products as ideal gases, the final (maximum) volume that the combustion chamber can have is determined to be

$$\frac{P \mathcal{V}_1}{P \mathcal{V}_2} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow \mathcal{V}_2 = \frac{N_2 T_2}{N_1 T_1} \mathcal{V}_1 = \frac{(10.52 \text{ kmol})(2328 \text{ K})}{(10.52 \text{ kmol})(298 \text{ K})}(1.5 \text{ L}) = \mathbf{11.7 \ L}$$

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15-112 n-Octane is burned with 100 percent excess air. The combustion is incomplete. The maximum work that can be produced is to be determined.

Assumptions **1** Combustion is incomplete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis The combustion equation with 100% excess air and 10% CO is

$$C_8H_{18} + 2\times 12.5(O_2 + 3.76N_2) \longrightarrow 8(0.90\ CO_2 + 0.10\ CO) + 9\ H_2O + xO_2 + 2\times 12.5\times 3.76\ N_2$$

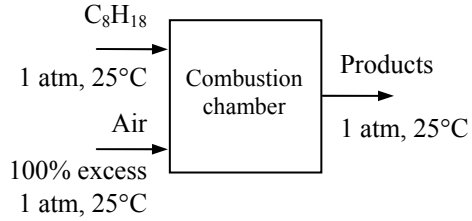
The coefficient for O_2 is determined from its mass balance as

$$25 = 7.2 + 0.4 + 4.5 + x \longrightarrow x = 12.9$$

Substituting,

$$C_8H_{18} + 25(O_2 + 3.76N_2) \longrightarrow 7.2\ CO_2 + 0.8\ CO + 9\ H_2O + 12.9\ O_2 + 94\ N_2$$

The reactants and products are at 25°C and 1 atm, which is the standard reference state and also the state of the surroundings. Therefore, the reversible work in this case is simply the difference between the Gibbs function of formation of the reactants and that of the products,



$$\begin{aligned} W_{rev} &= \sum N_R \bar{g}^\circ_{f,R} - \sum N_P \bar{g}^\circ_{f,P} \\ &= (1)(16{,}530) - (7.2)(-394{,}360) - (0.8)(-137{,}150) - (9)(-228{,}590) \\ &= 5{,}022{,}952\ \text{kJ} \quad \text{(per kmol of fuel)} \end{aligned}$$

since the g°_f of stable elements at 25°C and 1 atm is zero. Per unit mass basis,

$$W_{rev} = \frac{5{,}022{,}952\ \text{kJ/kmol}}{114\ \text{kg/kmol}} = \mathbf{44{,}060\ kJ/kg\ fuel}$$

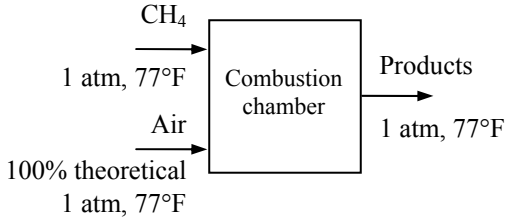
15-113E Methane is burned with stoichiometric air. The maximum work that can be produced is to be determined.

Assumptions **1** Combustion is incomplete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis The combustion equation is

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 7.52N_2$$

The reactants and products are at 77°F and 1 atm, which is the standard reference state and also the state of the surroundings. Therefore, the reversible work in this case is simply the difference between the Gibbs function of formation of the reactants and that of the products,



$$\begin{aligned} W_{rev} &= \sum N_R \bar{g}^\circ_{f,R} - \sum N_P \bar{g}^\circ_{f,P} \\ &= (1)(-21{,}860) - (1)(-169{,}680) - (2)(-98{,}350) \\ &= 344{,}520\ \text{Btu} \quad \text{(per lbmol of fuel)} \end{aligned}$$

since the g°_f of stable elements at 77°F and 1 atm is zero. Per unit mass basis,

$$W_{rev} = \frac{344{,}520\ \text{Btu/lbmol}}{16\ \text{lbm/lbmol}} = \mathbf{21{,}530\ Btu/lbm\ fuel}$$

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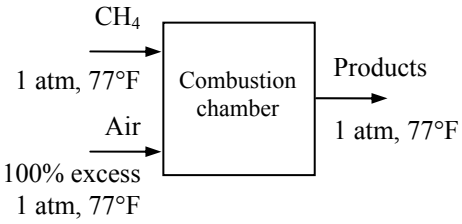
15-114E Methane is burned with 100% excess air. The maximum work that can be produced is to be determined and compared to when methane is burned with stoichiometric air.

Assumptions **1** Combustion is incomplete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis The combustion equation with 100% excess air is

$$CH_4 + 4(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 2O_2 + 15.04N_2$$

The reactants and products are at 77°F and 1 atm, which is the standard reference state and also the state of the surroundings. Therefore, the reversible work in this case is simply the difference between the Gibbs function of formation of the reactants and that of the products,



$$\begin{aligned} W_{\text{rev}} &= \sum N_R \bar{g}^\circ_{f,R} - \sum N_P \bar{g}^\circ_{f,P} \\ &= (1)(-21{,}860) - (1)(-169{,}680) - (2)(-98{,}350) \\ &= 344{,}520 \text{ Btu} \quad \text{(per lbmol of fuel)} \end{aligned}$$

since the g°_f of stable elements at 77°F and 1 atm is zero. Per unit mass basis,

$$W_{\text{rev}} = \frac{344{,}52 \text{ Btu/lbmol}}{16 \text{ lbm/lbmol}} = \mathbf{21{,}530 \text{ Btu/lbm fuel}}$$

The excess air only adds oxygen and nitrogen to the reactants and products. The excess air then does not change the maximum work.

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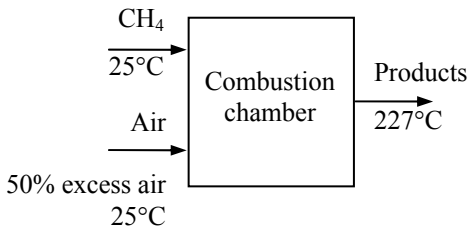
15-115 Methane is burned steadily with 50 percent excess air in a steam boiler. The amount of steam generated per unit of fuel mass burned, the change in the exergy of the combustion streams, the change in the exergy of the steam stream, and the lost work potential are to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Properties The molar masses of CH_4 and air are 16 kg/kmol and 29 kg/kmol, respectively (Table A-1).

Analysis (*a*) The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol CH_4, the combustion equation can be written as

$$CH_4 + 3(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + O_2 + 11.28N_2$$



Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol	$\bar{h}_{500K}$ kJ/kmol
CH_4	-74,850	---	---
O_2	0	8682	14,770
N_2	0	8669	14,581
H_2O (*g*)	-241,820	9904	16,828
CO_2	-393,520	9364	17,678

Thus,

$$\begin{aligned}-Q_{out} &= (1)(-393{,}520 + 17{,}678 - 9364) + (2)(-241{,}820 + 16{,}828 - 9904) \\ &\quad + (1)(0 + 14{,}770 - 8682) + (11.28)(0 + 14{,}581 - 8669) - (1)(-74{,}850) \\ &= -707{,}373 \text{ kJ/kmol of fuel}\end{aligned}$$

The heat loss per unit mass of the fuel is

$$Q_{out} = \frac{707{,}373 \text{ kJ/kmol of fuel}}{16 \text{ kg/kmol of fuel}} = 44{,}211 \text{ kJ/kg fuel}$$

The amount of steam generated per unit mass of fuel burned is determined from an energy balance to be (Enthalpies of steam are from tables A-4 and A-6)

$$\frac{m_s}{m_f} = \frac{Q_{out}}{\Delta h_s} = \frac{44{,}211 \text{ kJ/kg fuel}}{(3214.5 - 852.26) \text{ kJ/kg steam}} = \mathbf{18.72 \text{ kg steam/kg fuel}}$$

(*b*) The entropy generation during this process is determined from

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{out}}{T_{surr}}$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where y_i is the mole fraction of component *i*. Then,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

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The entropy calculations can be presented in tabular form as

	N_i	y_i	$\bar{s}_i^\circ(T,1\text{atm})$	$R_u \ln(y_i P_m)$	$N_i \bar{s}_i$
CH_4	1	---	186.16	0	186.16
O_2	3	0.21	205.04	-12.98	654.06
N_2	11.28	0.79	191.61	-1.960	2183.47
					S_R = 3023.69 kJ/K
CO_2	1	0.0654	234.814	-22.67	257.48
H_2O (g)	2	0.1309	206.413	-16.91	446.65
O_2	1	0.0654	220.589	-22.67	243.26
N_2	11.28	0.7382	206.630	-2.524	2359.26
					S_P = 3306.65 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = 3306.65 - 3023.69 + \frac{707{,}373}{298} = 2657 \text{ kJ/K (per kmol fuel)}$$

The exergy change of the combustion streams is equal to the exergy destruction since there is no actual work output. That is,

$$\Delta X_{\text{gases}} = -X_{\text{dest}} = -T_0 S_{\text{gen}} = -(298 \text{ K})(2657 \text{ kJ/K}) = -791{,}786 \text{ kJ/kmol fuel}$$

Per unit mass basis,

$$\Delta X_{\text{gases}} = \frac{-791{,}786 \text{ kJ/kmol fuel}}{16 \text{ kg/kmol}} = \mathbf{-49{,}490 \text{ kJ/kg fuel}}$$

Note that the exergy change is negative since the exergy of combustion gases decreases.

(*c*) The exergy change of the steam stream is

$$\Delta X_{\text{steam}} = \Delta h - T_0 \Delta s = (3214.5 - 852.26) - (298)(6.7714 - 2.3305) = \mathbf{1039 \text{ kJ/kg steam}}$$

(*d*) The lost work potential is the negative of the net exergy change of both streams:

$$\begin{aligned} X_{\text{dest}} &= -\left(\frac{m_s}{m_f} \Delta X_{\text{steam}} + \Delta X_{\text{gases}} \right) \\ &= -\left[(18.72 \text{ kg steam/kg fuel})(1039 \text{ kJ/kg steam}) + (-49{,}490 \text{ kJ/kg fuel}) \right] \\ &= \mathbf{30{,}040 \text{ kJ/kg fuel}} \end{aligned}$$

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15-116 A coal from Utah is burned steadily with 50 percent excess air in a steam boiler. The amount of steam generated per unit of fuel mass burned, the change in the exergy of the combustion streams, the change in the exergy of the steam stream, and the lost work potential are to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible. **5** The effect of sulfur on the energy and entropy balances is negligible.

Properties The molar masses of C, H_2, N_2, O_2, S, and air are 12, 2, 28, 32, 32, and 29 kg/kmol, respectively (Table A-1).

Analysis (*a*) We consider 100 kg of coal for simplicity. Noting that the mass percentages in this case correspond to the masses of the constituents, the mole numbers of the constituent of the coal are determined to be

$$N_{\mathrm{C}} = \frac{m_{\mathrm{C}}}{M_{\mathrm{C}}} = \frac{61.40\ \mathrm{kg}}{12\ \mathrm{kg/kmol}} = 5.117\ \mathrm{kmol}$$

$$N_{\mathrm{H2}} = \frac{m_{\mathrm{H2}}}{M_{\mathrm{H2}}} = \frac{5.79\ \mathrm{kg}}{2\ \mathrm{kg/kmol}} = 2.895\ \mathrm{kmol}$$

$$N_{\mathrm{O2}} = \frac{m_{\mathrm{O2}}}{M_{\mathrm{O2}}} = \frac{25.31\ \mathrm{kg}}{32\ \mathrm{kg/kmol}} = 0.7909\ \mathrm{kmol}$$

$$N_{\mathrm{N2}} = \frac{m_{\mathrm{N2}}}{M_{\mathrm{N2}}} = \frac{1.09\ \mathrm{kg}}{28\ \mathrm{kg/kmol}} = 0.03893\ \mathrm{kmol}$$

$$N_{\mathrm{S}} = \frac{m_{\mathrm{S}}}{M_{\mathrm{S}}} = \frac{1.41\ \mathrm{kg}}{32\ \mathrm{kg/kmol}} = 0.04406\ \mathrm{kmol}$$

61.40% C
5.79% H_2
25.31% O_2
1.09% N_2
1.41% S
5.00% ash
(by mass)

The mole number of the mixture and the mole fractions are

$$N_m = 5.117 + 2.895 + 0.7909 + 0.03893 + 0.04406 = 8.886\ \mathrm{kmol}$$

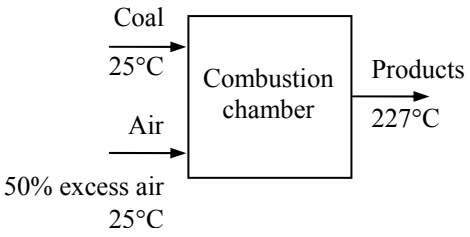
$$y_{\mathrm{C}} = \frac{N_{\mathrm{C}}}{N_m} = \frac{5.117\ \mathrm{kmol}}{8.886\ \mathrm{kmol}} = 0.5758$$

$$y_{\mathrm{H2}} = \frac{N_{\mathrm{H2}}}{N_m} = \frac{2.895\ \mathrm{kmol}}{8.886\ \mathrm{kmol}} = 0.3258$$

$$y_{\mathrm{O2}} = \frac{N_{\mathrm{O2}}}{N_m} = \frac{0.7909\ \mathrm{kmol}}{8.886\ \mathrm{kmol}} = 0.0890$$

$$y_{\mathrm{N2}} = \frac{N_{\mathrm{N2}}}{N_m} = \frac{0.03893\ \mathrm{kmol}}{8.886\ \mathrm{kmol}} = 0.00438$$

$$y_{\mathrm{S}} = \frac{N_{\mathrm{S}}}{N_m} = \frac{0.04406\ \mathrm{kmol}}{8.886\ \mathrm{kmol}} = 0.00496$$



Ash consists of the non-combustible matter in coal. Therefore, the mass of ash content that enters the combustion chamber is equal to the mass content that leaves. Disregarding this non-reacting component for simplicity, the combustion equation may be written as

$$0.5758\mathrm{C} + 0.3258\mathrm{H}_2 + 0.0890\mathrm{O}_2 + 0.00438\mathrm{N}_2 + 0.00496\mathrm{S} + 1.5a_{\mathrm{th}}(\mathrm{O}_2 + 3.76\mathrm{N}_2)$$
$$\longrightarrow 0.5758\mathrm{CO}_2 + 0.3258\mathrm{H}_2\mathrm{O} + 0.00496\mathrm{SO}_2 + 0.5a_{\mathrm{th}}\mathrm{O}_2 + 1.5a_{\mathrm{th}} \times 3.76\mathrm{N}_2$$

According to the oxygen balance,

$$\mathrm{O}_2\ \text{balance}: \quad 0.0890 + 1.5a_{\mathrm{th}} = 0.5758 + 0.5 \times 0.3258 + 0.00496 + 0.5a_{\mathrm{th}} \longrightarrow a_{\mathrm{th}} = 0.6547$$

Substituting,

$$0.5758\mathrm{C} + 0.3258\mathrm{H}_2 + 0.0890\mathrm{O}_2 + 0.00438\mathrm{N}_2 + 0.00496\mathrm{S} + 0.9821(\mathrm{O}_2 + 3.76\mathrm{N}_2)$$
$$\longrightarrow 0.5758\mathrm{CO}_2 + 0.3258\mathrm{H}_2\mathrm{O} + 0.00496\mathrm{SO}_2 + 0.3274\mathrm{O}_2 + 3.693\mathrm{N}_2$$

The apparent molecular weight of the coal is

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$$M_m = \frac{m_m}{N_m} = \frac{(0.5758\times 12 + 0.3258\times 2 + 0.0890\times 32 + 0.00438\times 28 + 0.00496\times 32)\ \text{kg}}{(0.5758 + 0.3258 + 0.0890 + 0.00438 + 0.00496)\ \text{kmol}}$$
$$= \frac{10.69\ \text{kg}}{1.0\ \text{kmol}} = 10.69\ \text{kg/kmol coal}$$

Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol	$\bar{h}_{500K}$ kJ/kmol
O_2	0	8682	14,770
N_2	0	8669	14,581
H_2O (g)	-241,820	9904	16,828
CO_2	-393,520	9364	17,678

Thus,

$$-Q_{\text{out}} = (0.5758)(-393{,}520 + 17{,}678 - 9364) + (0.3258)(-241{,}820 + 16{,}828 - 9904)$$
$$+ (0.3274)(0 + 14{,}770 - 8682) + (3.693)(0 + 14{,}581 - 8669) - 0$$
$$= -274{,}505\ \text{kJ/kmol of fuel}$$

The heat loss per unit mass of the fuel is

$$Q_{\text{out}} = \frac{274{,}505\ \text{kJ/kmol of fuel}}{10.69\ \text{kg/kmol of fuel}} = 25{,}679\ \text{kJ/kg fuel}$$

The amount of steam generated per unit mass of fuel burned is determined from an energy balance to be (Enthalpies of steam are from tables A-4 and A-6)

$$\frac{m_s}{m_f} = \frac{Q_{\text{out}}}{\Delta h_s} = \frac{25{,}679\ \text{kJ/kg fuel}}{(3214.5 - 852.26)\ \text{kJ/kg steam}} = \mathbf{10.87\ kg\ steam/kg\ fuel}$$

(*b*) The entropy generation during this process is determined from

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\text{total}}$, where y_i is the mole fraction of component i. Then,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

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The entropy calculations can be presented in tabular form as

	$\mathbf{N_i}$	$\mathbf{y_i}$	$\bar{\mathbf{s}}_\mathbf{i}^\circ\mathbf{(T,1atm)}$	$\mathbf{R_u ln(y_i P_m)}$	$\mathbf{N_i \bar{s}_i}$
C	0.5758	0.5758	5.74	-4.589	5.95
H_2	0.3258	0.3258	130.68	-9.324	45.61
O_2	0.0890	0.0890	205.04	-20.11	20.04
N_2	0.00438	0.00438	191.61	-45.15	1.04
O_2	0.9821	0.21	205.04	-12.98	214.12
N_2	3.693	0.79	191.61	-1.960	714.85
					S_R = 1001.61 kJ/K
CO_2	0.5758	0.1170	234.814	-17.84	145.48
H_2O (*g*)	0.3258	0.0662	206.413	-22.57	74.60
O_2	0.3274	0.0665	220.589	-22.54	79.60
N_2	3.693	0.7503	206.630	-2.388	771.90
					S_P = 1071.58 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = 1071.58 - 1001.61 + \frac{274{,}505}{298} = 991.1 \text{ kJ/K (per kmol fuel)}$$

The exergy change of the combustion streams is equal to the exergy destruction since there is no actual work output. That is,

$$\Delta X_{\text{gases}} = -X_{\text{dest}} = -T_0 S_{\text{gen}} = -(298 \text{ K})(991.1 \text{ kJ/K}) = -295{,}348 \text{ kJ/kmol fuel}$$

Per unit mass basis,

$$\Delta X_{\text{gases}} = \frac{-295{,}348 \text{ kJ/K}}{10.69 \text{ kg/kmol}} = \mathbf{-27{,}630 \text{ kJ/kg fuel}}$$

Note that the exergy change is negative since the exergy of combustion gases decreases.

(*c*) The exergy change of the steam stream is

$$\Delta X_{\text{steam}} = \Delta h - T_0 \Delta s = (3214.5 - 852.26) - (298)(6.7714 - 2.3305) = \mathbf{1039 \text{ kJ/kg steam}}$$

(*d*) The lost work potential is the negative of the net exergy change of both streams:

$$\begin{aligned} X_{\text{dest}} &= -\left(\frac{m_s}{m_f} \Delta X_{\text{steam}} + \Delta X_{\text{gases}} \right) \\ &= -\left[(10.87 \text{ kg steam/kg fuel})(1039 \text{ kJ/kg steam}) + (-27{,}630 \text{ kJ/kg fuel}) \right] \\ &= \mathbf{16{,}340 \text{ kJ/kg fuel}} \end{aligned}$$

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15-117 An expression for the HHV of a gaseous alkane C_nH_{2n+2} in terms of n is to be developed.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2. **3** Combustion gases are ideal gases.

Analysis The complete reaction balance for 1 kmol of fuel is

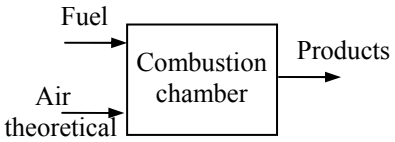
$$C_nH_{2n+2} + \frac{3n+1}{2}(O_2 + 3.76N_2) \longrightarrow nCO_2 + (n+1)H_2O + \frac{3n+1}{2}(3.76)N_2$$

Both the reactants and the products are taken to be at the standard reference state of 25°C and 1 atm for the calculation of heating values. The heat transfer for this process is equal to enthalpy of combustion. Note that N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then,

$$q = h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO2} + \left(N\bar{h}^\circ_f\right)_{H2O} - \left(N\bar{h}^\circ_f\right)_{fuel}$$

For the HHV, the water in the products is taken to be liquid. Then,

$$h_C = n(-393{,}520) + (n+1)(-285{,}830) - \left(\bar{h}^\circ_f\right)_{fuel}$$



The HHV of the fuel is

$$\text{HHV} = \frac{-h_C}{M_{fuel}} = \frac{n(-393{,}520) + (n+1)(-285{,}830) - \left(\bar{h}^\circ_f\right)_{fuel}}{M_{fuel}}$$

For the LHV, the water in the products is taken to be vapor. Then,

$$\text{LHV} = \frac{n(-393{,}520) + (n+1)(-241{,}820) - \left(\bar{h}^\circ_f\right)_{fuel}}{M_{fuel}}$$

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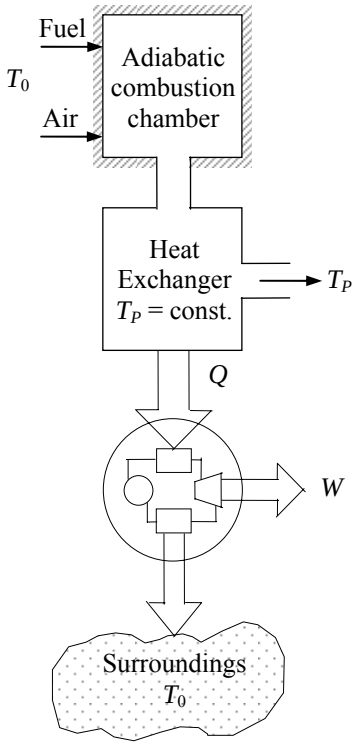
15-118 It is to be shown that the work output of the Carnot engine will be maximum when $T_p = \sqrt{T_0 T_{af}}$. It is also to be shown that the maximum work output of the Carnot engine in this case becomes $w = CT_{\text{af}}\left(1 - \frac{\sqrt{T_0}}{\sqrt{T_{\text{af}}}}\right)^2$.

Analysis The combustion gases will leave the combustion chamber and enter the heat exchanger at the adiabatic flame temperature T_{af} since the chamber is adiabatic and the fuel is burned completely. The combustion gases experience no change in their chemical composition as they flow through the heat exchanger. Therefore, we can treat the combustion gases as a gas stream with a constant specific heat c_p. Noting that the heat exchanger involves no work interactions, the energy balance equation for this single-stream steady-flow device can be written as

$$\dot{Q} = \dot{m}(h_e - h_i) = \dot{m}C(T_p - T_{\text{af}})$$

where $\dot{Q}$ is the negative of the heat supplied to the heat engine. That is,

$$\dot{Q}_H = -\dot{Q} = \dot{m}C(T_{\text{af}} - T_p)$$



Then the work output of the Carnot heat engine can be expressed as

$$\dot{W} = \dot{Q}_H\left(1 - \frac{T_0}{T_p}\right) = \dot{m}C(T_{\text{af}} - T_p)\left(1 - \frac{T_0}{T_p}\right) \qquad (1)$$

Taking the partial derivative of $\dot{W}$ with respect to T_p while holding T_{af} and T_0 constant gives

$$\frac{\partial W}{\partial T_p} = 0 \longrightarrow -\dot{m}C\left(1 - \frac{T_0}{T_p}\right) + \dot{m}C(T_p - T_{\text{af}})\frac{T_0}{T_p^2} = 0$$

Solving for T_p we obtain

$$T_p = \sqrt{T_0 T_{\text{af}}}$$

which the temperature at which the work output of the Carnot engine will be a maximum. The maximum work output is determined by substituting the relation above into Eq. (1),

$$\dot{W} = \dot{m}C(T_{\text{af}} - T_p)\left(1 - \frac{T_0}{T_p}\right) = \dot{m}C\left(T_{\text{af}} - \sqrt{T_0 T_{\text{af}}}\right)\left(1 - \frac{T_0}{\sqrt{T_0 T_{\text{af}}}}\right)$$

It simplifies to

$$\dot{W} = \dot{m}CT_{\text{af}}\left(1 - \frac{\sqrt{T_0}}{\sqrt{T_{\text{af}}}}\right)^2$$

or

$$w = CT_{\text{af}}\left(1 - \frac{\sqrt{T_0}}{\sqrt{T_{\text{af}}}}\right)^2$$

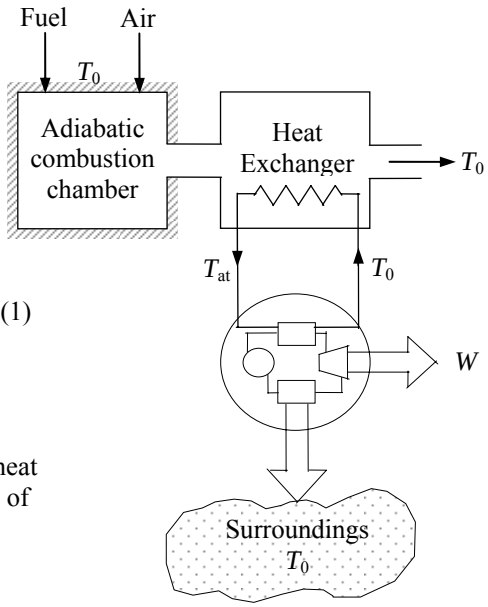
which is the desired relation.

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15-119 It is to be shown that the work output of the reversible heat engine operating at the specified conditions is $\dot{W}_{\text{rev}} = \dot{m}CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right)$. It is also to be shown that the effective flame temperature T_e of the furnace considered is $T_e = \frac{T_{\text{af}} - T_0}{\ln(T_{\text{af}}/T_0)}$.

Analysis The combustion gases will leave the combustion chamber and enter the heat exchanger at the adiabatic flame temperature T_{af} since the chamber is adiabatic and the fuel is burned completely. The combustion gases experience no change in their chemical composition as they flow through the heat exchanger. Therefore, we can treat the combustion gases as a gas stream with a constant specific heat c_p. Also, the work output of the reversible heat engine is equal to the reversible work W_{rev} of the heat exchanger as the combustion gases are cooled from T_{af} to T_0. That is,

$$\begin{aligned}\dot{W}_{\text{rev}} &= \dot{m}(h_i - h_e - T_0(s_i - s_e)) \\ &= \dot{m}C\left(T_{\text{af}} - T_0 - T_0\left(C\ln\frac{T_{\text{af}}}{T_0} - R\ln\frac{P_{\text{af}}^{\nearrow 0}}{P_0}\right)\right) \\ &= \dot{m}C\left(T_{\text{af}} - T_0 - T_0 C\ln\frac{T_{\text{af}}}{T_0}\right)\end{aligned}$$



which can be rearranged as

$$\dot{W}_{\text{rev}} = \dot{m}CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right) \quad \text{or} \quad w_{\text{rev}} = CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right) \quad (1)$$

which is the desired result.

The effective flame temperature T_e can be determined from the requirement that a Carnot heat engine which receives the same amount of heat from a heat reservoir at constant temperature T_e produces the same amount of work. The amount of heat delivered to the heat engine above is

$$\dot{Q}_H = \dot{m}(h_i - h_e) = \dot{m}C(T_{\text{af}} - T_0)$$

A Carnot heat engine which receives this much heat at a constant temperature T_e will produce work in the amount of

$$\dot{W} = \dot{Q}_H\eta_{\text{th,Carnot}} = \dot{m}C(T_{\text{af}} - T_0)\left(1 - \frac{T_0}{T_e}\right) \quad (2)$$

Setting equations (1) and (2) equal to each other yields

$$\begin{aligned}\dot{m}CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right) &= \dot{m}C(T_{\text{af}} - T_0)\left(1 - \frac{T_0}{T_e}\right) \\ T_{\text{af}} - T_0 - T_0\ln\frac{T_{\text{af}}}{T_0} &= T_{\text{af}} - T_{\text{af}}\frac{T_0}{T_e} - T_0 + T_0\frac{T_0}{T_e}\end{aligned}$$

Simplifying and solving for T_e, we obtain

$$T_e = \frac{T_{\text{af}} - T_0}{\ln(T_{\text{af}}/T_0)}$$

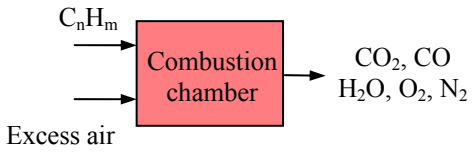
which is the desired relation.

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15-120 The combustion of a hydrocarbon fuel C_nH_m with excess air and incomplete combustion is considered. The coefficients of the reactants and products are to be written in terms of other parameters.

Analysis The balanced reaction equation for stoichiometric air is

$$C_nH_m + A_{th}[O_2 + 3.76N_2] \longrightarrow n\,CO_2 + (m/2)\,H_2O + 3.76A_{th}\,N_2$$



The stoichiometric coefficient A_{th} is determined from an O_2 balance:

$$A_{th} = n + m/4$$

The reaction with excess air and incomplete combustion is

$$C_nH_m + (1+B)A_{th}[O_2 + 3.76N_2] \longrightarrow an\,CO_2 + bn\,CO + (m/2)\,H_2O + G\,O_2 + 3.76(1+B)A_{th}\,N_2$$

The given reaction is

$$C_nH_m + (1+B)A_{th}[O_2 + 3.76N_2] \longrightarrow D\,CO_2 + E\,CO + F\,H_2O + G\,O_2 + J\,N_2$$

Thus,

$$D = an$$
$$E = bn$$
$$F = m/2$$
$$J = 3.76(1+B)A_{th}$$

The coefficient G for O_2 is determined from a mass balance,

O_2 balance:

$$(1+B)A_{th} = an + \frac{bn}{2} + \frac{m}{4} + G$$

$$(1+B)\left(n + \frac{m}{4}\right) = an + \frac{bn}{2} + \frac{m}{4} + G$$

$$\left(n + \frac{m}{4}\right) + BA_{th} = an + \frac{bn}{2} + \frac{m}{4} + G$$

$$G = n + BA_{th} - an - \frac{bn}{2}$$

$$= n(1-a) + BA_{th} - \frac{bn}{2}$$

$$= nb - \frac{bn}{2} + BA_{th}$$

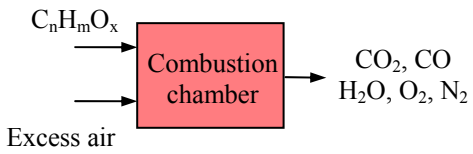
$$= \frac{bn}{2} + BA_{th}$$

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15-121 The combustion of an alcohol fuel ($C_nH_mO_x$) with excess air and incomplete combustion is considered. The coefficients of the reactants and products are to be written in terms of other parameters.

Analysis The balanced reaction equation for stoichiometric air is

$$C_nH_mO_x + A_{th}[O_2 + 3.76N_2] \longrightarrow n\,CO_2 + (m/2)\,H_2O + 3.76A_{th}\,N_2$$



The stoichiometric coefficient A_{th} is determined from an O_2 balance:

$$x/2 + A_{th} = n + m/4 \longrightarrow A_{th} = n + m/4 - x/2$$

The reaction with excess air and incomplete combustion is

$$C_nH_mO_x + (1+B)A_{th}[O_2 + 3.76N_2] \longrightarrow an\,CO_2 + bn\,CO + (m/2)\,H_2O + G\,O_2 + 3.76(1+B)A_{th}\,N_2$$

The given reaction is

$$C_nH_mO_x + (1+B)A_{th}[O_2 + 3.76N_2] \longrightarrow D\,CO_2 + E\,CO + F\,H_2O + G\,O_2 + J\,N_2$$

Thus,

$$D = an$$
$$E = bn$$
$$F = m/2$$
$$J = 3.76(1+B)A_{th}$$

The coefficient G for O_2 is determined from a mass balance,

O_2 balance:

$$\frac{x}{2} + (1+B)A_{th} = an + \frac{bn}{2} + \frac{m}{4} + G$$

$$\frac{x}{2} + (1+B)\left(n + \frac{m}{4} - \frac{x}{2}\right) = an + \frac{bn}{2} + \frac{m}{4} + G$$

$$\frac{x}{2} + \left(n + \frac{m}{4} - \frac{x}{2}\right) + B\left(n + \frac{m}{4} - \frac{x}{2}\right) = an + \frac{bn}{2} + \frac{m}{4} + G$$

$$\frac{x}{2} + \left(n + \frac{m}{4} - \frac{x}{2}\right) + BA_{th} = an + \frac{bn}{2} + \frac{m}{4} + G$$

$$G = n - an + BA_{th} - \frac{bn}{2}$$

$$= n(1-a) - \frac{bn}{2} + BA_{th}$$

$$= nb - \frac{bn}{2} + BA_{th}$$

$$= \frac{bn}{2} + BA_{th}$$

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15-122 The combustion of a mixture of an alcohol fuel ($C_nH_mO_x$) and a hydrocarbon fuel (C_wH_z) with excess air and incomplete combustion is considered. The coefficients of the reactants and products are to be written in terms of other parameters.

Analysis The balanced reaction equation for stoichiometric air is

$$y_1\ C_nH_mO_x + y_2\ C_wH_z + A_{th}\left[O_2 + 3.76N_2\right] \longrightarrow (y_1n + y_2w)\,CO_2 + 0.5(y_1m + y_2z)\,H_2O + 3.76A_{th}\ N_2$$

The stoichiometric coefficient A_{th} is determined from an O_2 balance:

$$y_1x/2 + A_{th} = (y_1n + y_2w) + (y_1m + y_2z)/4 \longrightarrow A_{th} = (y_1n + y_2w) + (y_1m + y_2z)/4 - y_1x/2$$

The reaction with excess air and incomplete combustion is

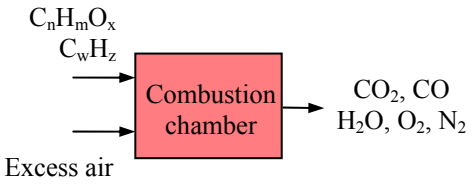
$$y_1\ C_nH_mO_x + y_2\ C_wH_z + (1+B)A_{th}\left[O_2 + 3.76N_2\right]$$
$$\longrightarrow a(y_1n + y_2w)\,CO_2 + b(y_1n + y_2w)\,CO + 0.5(y_1m + y_2z)\,H_2O + G\,O_2 + 3.76(1+B)A_{th}\ N_2$$

The given reaction is

$$y_1\ C_nH_mO_x + y_2\ C_wH_z + (1+B)A_{th}\left[O_2 + 3.76N_2\right] \longrightarrow D\,CO_2 + E\,CO + F\,H_2O + G\,O_2 + J\,N_2$$

Thus,

$$D = a(y_1n + y_2w)$$
$$E = b(y_1n + y_2w)$$
$$F = 0.5(y_1m + y_2z)$$
$$J = 3.76(1+B)A_{th}$$



The coefficient G for O_2 is determined from a mass balance,

O_2 balance:

$$0.5y_1x + (1+B)A_{th} = a(y_1n + y_2w) + 0.5b(y_1n + y_2w) + 0.25(y_1m + y_2z) + G$$
$$0.5y_1x + (y_1n + y_2w) + 0.25(y_1m + y_2z) - 0.5y_1x + BA_{th} = a(y_1n + y_2w) + 0.5b(y_1n + y_2w) + 0.25(y_1m + y_2z) + G$$
$$G = (y_1n + y_2w) - a(y_1n + y_2w) - 0.5b(y_1n + y_2w) + BA_{th}$$
$$= (y_1n + y_2w)(1-a) - 0.5b(y_1n + y_2w) + BA_{th}$$
$$= b(y_1n + y_2w) - 0.5b(y_1n + y_2w) + BA_{th}$$
$$= 0.5b(y_1n + y_2w) + BA_{th}$$

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15-123 The effect of the amount of air on the adiabatic flame temperature of liquid octane (C_8H_{18}) is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2) (Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Adiabatic, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  ((y/4 + x-z/2)
(Theo_air/100 - 1) +w/2)O2"

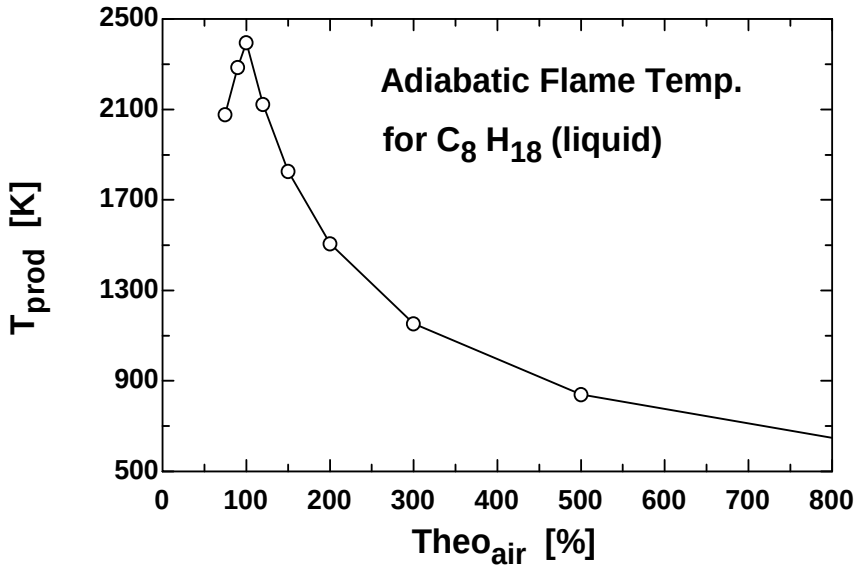
"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
        x=2;y=2; z=0
          Name$='Acetylene'
                                        h_fuel = 226730
else
If fuel$='C3H8(l)' then
        x=3; y=8; z=0
          Name$='Propane(liq)'
                                        h_fuel = -103850-15060
else
If fuel$='C8H18(l)' then
        x=8; y=18; z=0
          Name$='Octane(liq)'
                                        h_fuel = -249950
else
if fuel$='CH4(g)' then
        x=1; y=4; z=0
          Name$='Methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
        x=1; y=4; z=1
          Name$='Methyl alcohol'
                                        h_fuel = -200670
endif; endif; endif; endif; endif
end
Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
{MolCO = 0
MolCO2 = x}
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
```

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```
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3 %',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END
{"Input data from the diagram window"
T_air = 298    [K]
Theo_air = 200  "%"
Fuel$='CH4(g)'}
T_fuel = 298   [K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=(x-w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x+y/4-
z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

$Theo_{air}$ [%]	T_{prod} [K]
75	2077
90	2287
100	2396
120	2122
150	1827
200	1506
300	1153
500	840.1
800	648.4



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15-124 A general program is to be written to determine the adiabatic flame temperature during the complete combustion of a hydrocarbon fuel C_nH_m at 25°C in a steady-flow combustion chamber when the percent of excess air and its temperature are specified.

Analysis The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2) (Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Adiabatic, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  ((y/4 + x-z/2)
(Theo_air/100 - 1) +w/2)O2"

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
        x=2;y=2; z=0
          Name$='acetylene'
                                        h_fuel = 226730
else
If fuel$='C3H8(l)' then
        x=3; y=8; z=0
          Name$='propane(liq)'
                                        h_fuel = -103850-15060
else
If fuel$='C8H18(l)' then
        x=8; y=18; z=0
          Name$='octane(liq)'
                                        h_fuel = -249950
else
if fuel$='CH4(g)' then
        x=1; y=4; z=0
          Name$='methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
        x=1; y=4; z=1
          Name$='methyl alcohol'
                                        h_fuel = -200670

endif; endif; endif; endif; endif
end

Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)

ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100

IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
{MolCO = 0
MolCO2 = x}
```

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```
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3 %',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END

{"Input data from the diagram window"
T_air = 298 [K]
Theo_air = 120 [%]
Fuel$='CH4(g)'}
T_fuel = 298 [K]

Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=(x-w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x+y/4-
z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

SOLUTION for the sample calculation

A_th=5	fuel$='C3H8(l)'
HP=-119035 [kJ/kg]	HR=-119035 [kJ/kg]
h_fuel=-118910	Moles_CO=0.000
Moles_CO2=3.000	Moles_H2O=4
Moles_N2=22.560	Moles_O2=1.000
MolO2=1	Name$='propane(liq)'
SolMeth$='>= 100%, the solution assumes complete combustion.'	
Theo_air=120 [%]	Th_air=1.200
T_air=298 [K]	T_fuel=298 [K]
T_prod=2112 [K]	w=0
x=3	y=8
z=0	

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15-125 The minimum percent of excess air that needs to be used for the fuels $CH_4(g)$, $C_2H_2(g)$, $CH_3OH(g)$, $C_3H_8(g)$, and $C_8H_{18}(l)$ if the adiabatic flame temperature is not to exceed 1500 K is to be determined.

Analysis The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2) (Theo_air/100 - 1) O2"
{"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
1+ 2*A_th=1*2+2*1"theoretical O balance"}

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
        x=2;y=2; z=0
          Name$='acetylene'
                                        h_fuel = 226730
else
If fuel$='C3H8(g)' then
        x=3; y=8; z=0
          Name$='propane'
                                        h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
        x=8; y=18; z=0
          Name$='octane'
                                        h_fuel = -249950
else
if fuel$='CH4(g)' then
        x=1; y=4; z=0
          Name$='methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
        x=1; y=4; z=1
          Name$='methyl alcohol'
                                        h_fuel = -200670

endif; endif; endif; endif; endif
end

{"Input data from the diagram window"
T_air = 298 [K]
Fuel$='CH4(g)'}
T_fuel = 298 [K]

Excess_air=Theo_air - 100 "[%]"
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th = y/4 + x-z/2
Th_air = Theo_air/100
HR=h_fuel+ (y/4 + x-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(y/4 + x-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
```

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HP=x*enthalpy(CO2,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(y/4 + x-z/2)*
(Theo_air/100)*enthalpy(N2,T=T_prod)+(y/4 + x-z/2) *(Theo_air/100 - 1)*enthalpy(O2,T=T_prod)

Moles_O2=(y/4 + x-z/2) *(Theo_air/100 - 1)
Moles_N2=3.76*(y/4 + x-z/2)* (Theo_air/100)
Moles_CO2=x
Moles_H2O=y/2
T[1]=T_prod; xa[1]=Theo_air

SOLUTION for a sample calculation

A_th=2.5	Excess_air=156.251 [%]
fuel$='C2H2(g)'	HP=226596 [kJ/kg]
HR=226596 [kJ/kg]	h_fuel=226730
Moles_CO2=2	Moles_H2O=1
Moles_N2=24.09	Moles_O2=3.906
Name$='acetylene'	Theo_air=256.3 [%]
Th_air=2.563	T[1]=1500 [K]
T_air=298 [K]	T_fuel=298 [K]
T_prod=1500 [K]	x=2
xa[1]=256.3	y=2
z=0	

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15-126 The minimum percentages of excess air that need to be used for the fuels $CH_4(g)$, $C_2H_2(g)$, $CH_3OH(g)$, $C_3H_8(g)$, and C8H18(l) AFOR adiabatic flame temperatures of 1200 K, 1750 K, and 2000 K are to be determined.

Analysis The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2) (Theo_air/100 - 1) O2"
{"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
1+ 2*A_th=1*2+2*1"theoretical O balance"}

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
        x=2;y=2; z=0
          Name$='acetylene'
                                        h_fuel = 226730
else
If fuel$='C3H8(g)' then
        x=3; y=8; z=0
          Name$='propane'
                                        h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
        x=8; y=18; z=0
          Name$='octane'
                                        h_fuel = -249950
else
if fuel$='CH4(g)' then
        x=1; y=4; z=0
          Name$='methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
        x=1; y=4; z=1
          Name$='methyl alcohol'
                                        h_fuel = -200670

endif; endif; endif; endif; endif
end

{"Input data from the diagram window"
T_air = 298 [K]
Fuel$='CH4(g)'}
T_fuel = 298 [K]

Excess_air=Theo_air - 100 "[%]"
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th = y/4 + x-z/2
Th_air = Theo_air/100
HR=h_fuel+ (y/4 + x-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(y/4 + x-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
```

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HP=x*enthalpy(CO2,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(y/4 + x-z/2)*
(Theo_air/100)*enthalpy(N2,T=T_prod)+(y/4 + x-z/2) *(Theo_air/100 - 1)*enthalpy(O2,T=T_prod)

Moles_O2=(y/4 + x-z/2) *(Theo_air/100 - 1)
Moles_N2=3.76*(y/4 + x-z/2)* (Theo_air/100)
Moles_CO2=x
Moles_H2O=y/2
T[1]=T_prod; xa[1]=Theo_air

SOLUTION for a sample calculation

A_th=5	Excess_air=31.395 [%]
fuel$='C3H8(g)'	HP=-103995 [kJ/kg]
HR=-103995 [kJ/kg]	h_fuel=-103858
Moles_CO2=3	Moles_H2O=4
Moles_N2=24.7	Moles_O2=1.570
Name$='propane'	Theo_air=131.4 [%]
Th_air=1.314	T[1]=2000 [K]
T_air=298 [K]	T_fuel=298 [K]
T_prod=2000 [K]	x=3
xa[1]=131.4	y=8
z=0	

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15-127 The adiabatic flame temperature of $CH_4(g)$ is to be determined when both the fuel and the air enter the combustion chamber at 25°C for the cases of 0, 20, 40, 60, 80, 100, 200, 500, and 1000 percent excess air.

Analysis The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2) (Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Adiabatic, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
          <--> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  ((y/4 + x-z/2)
(Theo_air/100 - 1) +w/2)O2"
"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
          x=2;y=2; z=0
            Name$='acetylene'
                                        h_fuel = 226730
else
If fuel$='C3H8(g)' then
          x=3; y=8; z=0
            Name$='propane'
                                        h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
          x=8; y=18; z=0
            Name$='octane'
                                        h_fuel = -249950
else
if fuel$='CH4(g)' then
          x=1; y=4; z=0
            Name$='methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
          x=1; y=4; z=1
            Name$='methyl alcohol'
                                        h_fuel = -200670

endif; endif; endif; endif; endif
end

Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)

ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100

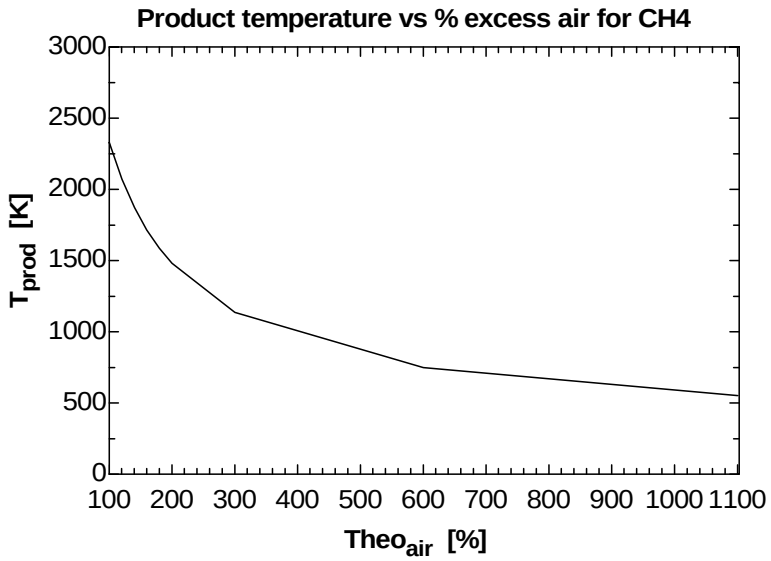
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
{MolCO = 0
MolCO2 = x}
w=0
MolO2 = A_th*(Th_air - 1)
```

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```
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3 %',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END

{"Input data from the diagram window"
T_air = 298 [K]
Theo_air = 200 [%]
Fuel$='CH4(g)'}
T_fuel = 298 [K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=(x-w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x+y/4-
z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

$Theo_{air}$ [%]	T_{prod} [K]
100	2329
120	2071
140	1872
160	1715
180	1587
200	1480
300	1137
600	749.5
1100	553



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15-128 The fuel among $CH_4(g)$, $C_2H_2(g)$, $C_2H_6(g)$, $C_3H_8(g)$, and $C_8H_{18}(l)$ that gives the highest temperature when burned completely in an adiabatic constant-volume chamber with the theoretical amount of air is to be determined.

Analysis The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm  with Stoichiometric Air at T_fuel =T_air=T_reac in a constant volume, closed system:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
        --> xCO2  +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  (x+y/4-z/2) (Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"

"Adiabatic, Incomplete Combustion of fuel CnHm with Stoichiometric Air at T_fuel =T_air=T_reac in a constant volume, closed system:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
   --> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  ((x+y/4-z/2) (Theo_air/100 - 1) +w/2)O2"

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "

"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
        x=2;y=2; z=0
          Name$='acetylene'
                                        h_fuel = 226730"Table A.26"
else
If fuel$='C3H8(g)' then
        x=3; y=8; z=0
          Name$='propane'
                                        h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
        x=8; y=18; z=0
          Name$='octane'
                                        h_fuel = -249950"Table A.26"
else
if fuel$='CH4(g)' then
        x=1; y=4; z=0
          Name$='methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
        x=1; y=4; z=1
          Name$='methyl alcohol'
                                        h_fuel = -200670"Table A.26"
endif; endif; endif; endif; endif
end

Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
w=0
MolO2 = A_th*(Th_air - 1)
```

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```
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3 %',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END

{"Input data from the diagram window"
Theo_air = 200 [%]
Fuel$='CH4(g)'}

T_reac = 298  [K]
T_air = T_reac
T_fuel = T_reac
R_u = 8.314 [kJ/kmol-K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
UR=(h_fuel-R_u*T_fuel)+ (x+y/4-z/2) *(Theo_air/100) *(enthalpy(O2,T=T_air)-R_u*T_air)+3.76*(x+y/4-z/2)
*(Theo_air/100) *(enthalpy(N2,T=T_air)-R_u*T_air)
UP=(x-w)*(enthalpy(CO2,T=T_prod)-R_u*T_prod)+w*(enthalpy(CO,T=T_prod)-
R_u*T_prod)+(y/2)*(enthalpy(H2O,T=T_prod)-R_u*T_prod)+3.76*(x+y/4-z/2)*
(Theo_air/100)*(enthalpy(N2,T=T_prod)-R_u*T_prod)+MolO2*(enthalpy(O2,T=T_prod)-R_u*T_prod)

UR =UP "Adiabatic, constant volume conservation of energy"
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

SOLUTION for CH4

A_th=2	fuel$='CH4(g)'	h_fuel=-74875
Moles_CO=0.000	Moles_CO2=1.000	Moles_H2O=2
Moles_N2=7.520	Moles_O2=0.000	MolO2=0
Name$='methane'	R_u=8.314 [kJ/kmol-K]	
SolMeth$='>= 100%, the solution assumes complete combustion.'		
Theo_air=100 [%]	Th_air=1.000	T_air=298 [K]
T_fuel=298 [K]	**T_prod=2824 [K]**	T_reac=298 [K]
UP=-100981	UR=-100981	w=0
x=1	y=4	z=0

SOLUTION for C2H2

A_th=2.5	fuel$='C2H2(g)'	h_fuel=226730
Moles_CO=0.000	Moles_CO2=2.000	Moles_H2O=1
Moles_N2=9.400	Moles_O2=0.000	MolO2=0
Name$='acetylene'	R_u=8.314 [kJ/kmol-K]	
SolMeth$='>= 100%, the solution assumes complete combustion.'		
Theo_air=100 [%]	Th_air=1.000	T_air=298 [K]
T_fuel=298 [K]	**T_prod=3535 [K]**	T_reac=298 [K]
UP=194717	UR=194717	w=0
x=2	y=2	z=0

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SOLUTION for CH3OH

A_th=1.5	fuel$='CH3OH(g)'	h_fuel=-200670
Moles_CO=0.000	Moles_CO2=1.000	Moles_H2O=2
Moles_N2=5.640	Moles_O2=0.000	MolO2=0
Name$='methyl alcohol'	R_u=8.314 [kJ/kmol-K]	
SolMeth$='>= 100%, the solution assumes complete combustion.'		
Theo_air=100 [%]	Th_air=1.000	T_air=298 [K]
T_fuel=298 [K]	**T_prod=2817 [K]**	T_reac=298 [K]
UP=-220869	UR=-220869	w=0
x=1	y=4	z=1

SOLUTION for C3H8

A_th=5	fuel$='C3H8(g)'	h_fuel=-103858
Moles_CO=0.000	Moles_CO2=3.000	Moles_H2O=4
Moles_N2=18.800	Moles_O2=0.000	MolO2=0
Name$='propane'	R_u=8.314 [kJ/kmol-K]	
SolMeth$='>= 100%, the solution assumes complete combustion.'		
Theo_air=100 [%]	Th_air=1.000	T_air=298 [K]
T_fuel=298 [K]	**T_prod=2909 [K]**	T_reac=298 [K]
UP=-165406	UR=-165406	w=0
x=3	y=8	z=0

SOLUTION for C8H18

A_th=12.5	fuel$='C8H18(l)'	h_fuel=-249950
Moles_CO=0.000	Moles_CO2=8.000	Moles_H2O=9
Moles_N2=47.000	Moles_O2=0.000	MolO2=0
Name$='octane'	R_u=8.314 [kJ/kmol-K]	
SolMeth$='>= 100%, the solution assumes complete combustion.'		
Theo_air=100 [%]	Th_air=1.000	T_air=298 [K]
T_fuel=298 [K]	**T_prod=2911 [K]**	T_reac=298 [K]
UP=-400104	UR=-400104	w=0
x=8	y=18	z=0

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Fundamentals of Engineering (FE) Exam Problems

15-129 A fuel is burned with 70 percent theoretical air. This is equivalent to

(a) 30% excess air (b) 70% excess air (c) 30% deficiency of air
(d) 70% deficiency of air (e) stoichiometric amount of air

Answer (c) 30% deficiency ofair

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
air_th=0.7
"air_th=air_access+1"
air_th=1-air_deficiency
```

15-130 Propane C_3H_8 is burned with 150 percent theoretical air. The air-fuel mass ratio for this combustion process is

(a) 5.3 (b) 10.5 (c) 15.7 (d) 23.4 (e) 39.3

Answer (d) 23.4

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_C=3
n_H=8
m_fuel=n_H*1+n_C*12
a_th=n_C+n_H/4
coeff=1.5  "coeff=1 for theoretical combustion, 1.5 for 50% excess air"
n_O2=coeff*a_th
n_N2=3.76*n_O2
m_air=n_O2*32+n_N2*28
AF=m_air/m_fuel
```

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15-131 One kmol of methane (CH_4) is burned with an unknown amount of air during a combustion process. If the combustion is complete and there are 1 kmol of free O_2 in the products, the air-fuel mass ratio is

(a) 34.6 (b) 25.7 (c) 17.2 (d) 14.3 (e) 11.9

Answer (b) 25.7

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_C=1
n_H=4
m_fuel=n_H*1+n_C*12
a_th=n_C+n_H/4
(coeff-1)*a_th=1  "O2 balance: Coeff=1 for theoretical combustion, 1.5 for 50% excess air"
n_O2=coeff*a_th
n_N2=3.76*n_O2
m_air=n_O2*32+n_N2*28
AF=m_air/m_fuel

"Some Wrong Solutions with Common Mistakes:"
W1_AF=1/AF "Taking the inverse of AF"
W2_AF=n_O2+n_N2 "Finding air-fuel mole ratio"
W3_AF=AF/coeff "Ignoring excess air"
```

15-132 A fuel is burned steadily in a combustion chamber. The combustion temperature will be the highest except when

(a) the fuel is preheated.
(b) the fuel is burned with a deficiency of air.
(c) the air is dry.
(d) the combustion chamber is well insulated.
(e) the combustion is complete.

Answer (b) the fuel is burned with a deficiency of air.

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15-133 An equimolar mixture of carbon dioxide and water vapor at 1 atm and 60°C enter a dehumidifying section where the entire water vapor is condensed and removed from the mixture, and the carbon dioxide leaves at 1 atm and 60°C. The entropy change of carbon dioxide in the dehumidifying section is

(a) –2.8 kJ/kg·K (b) –0.13 kJ/kg·K (c) 0 (d) 0.13 kJ/kg·K (e) 2.8 kJ/kg·K

Answer (b) –0.13 kJ/kg·K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp_CO2=0.846
R_CO2=0.1889
T1=60+273 "K"
T2=T1
P1= 1 "atm"
P2=1 "atm"
y1_CO2=0.5; P1_CO2=y1_CO2*P1
y2_CO2=1; P2_CO2=y2_CO2*P2
Ds_CO2=Cp_CO2*ln(T2/T1)-R_CO2*ln(P2_CO2/P1_CO2)

"Some Wrong Solutions with Common Mistakes:"
W1_Ds=0 "Assuming no entropy change"
W2_Ds=Cp_CO2*ln(T2/T1)-R_CO2*ln(P1_CO2/P2_CO2) "Using pressure fractions backwards"
```

15-134 Methane (CH_4) is burned completely with 80% excess air during a steady-flow combustion process. If both the reactants and the products are maintained at 25°C and 1 atm and the water in the products exists in the liquid form, the heat transfer from the combustion chamber per unit mass of methane is

(a) 890 MJ/kg (b) 802 MJ/kg (c) 75 MJ/kg (d) 56 MJ/kg (e) 50 MJ/kg

Answer (d) 56 MJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T= 25 "C"
P=1 "atm"
EXCESS=0.8
"Heat transfer in this case is the HHV at room temperature,"
HHV_CH4 =55.53 "MJ/kg"
LHV_CH4 =50.05 "MJ/kg"

"Some Wrong Solutions with Common Mistakes:"
W1_Q=LHV_CH4 "Assuming lower heating value"
W2_Q=EXCESS*hHV_CH4 "Assuming Q to be proportional to excess air"
```

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15-135 The higher heating value of a hydrocarbon fuel C_nH_m with $m = 8$ is given to be 1560 MJ/kmol of fuel. Then its lower heating value is

(a) 1384 MJ/kmol (b) 1208 MJ/kmol (c) 1402 MJ/kmol (d) 1540 MJ/kmol (e) 1550 MJ/kmol

Answer (a) 1384 MJ/kmol

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
HHV=1560 "MJ/kmol fuel"
h_fg=2.4423 "MJ/kg, Enthalpy of vaporization of water at 25C"
n_H=8
n_water=n_H/2
m_water=n_water*18
LHV=HHV-h_fg*m_water

"Some Wrong Solutions with Common Mistakes:"
W1_LHV=HHV - h_fg*n_water  "Using mole numbers instead of mass"
W2_LHV= HHV - h_fg*m_water*2 "Taking mole numbers of H2O to be m instead of m/2"
W3_LHV= HHV - h_fg*n_water*2 "Taking mole numbers of H2O to be m instead of m/2, and using mole numbers"
```

15-136 Acetylene gas (C_2H_2) is burned completely during a steady-flow combustion process. The fuel and the air enter the combustion chamber at 25°C, and the products leave at 1500 K. If the enthalpy of the products relative to the standard reference state is –404 MJ/kmol of fuel, the heat transfer from the combustion chamber is

(a) 177 MJ/kmol (b) 227 MJ/kmol (c) 404 MJ/kmol (d) 631 MJ/kmol (e) 751 MJ/kmol

Answer (d) 631 MJ/kmol

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
hf_fuel=226730/1000 "MJ/kmol fuel"
H_prod=-404 "MJ/kmol fuel"
H_react=hf_fuel
Q_out=H_react-H_prod

"Some Wrong Solutions with Common Mistakes:"
W1_Qout= -H_prod "Taking Qout to be H_prod"
W2_Qout= H_react+H_prod "Adding enthalpies instead of subtracting them"
```

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15-137 Benzene gas (C_6H_6) is burned with 95 percent theoretical air during a steady-flow combustion process. The mole fraction of the CO in the products is

(a) 8.3% (b) 4.7% (c) 2.1% (d) 1.9% (e) 14.3%

Answer (c) 2.1%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_C=6
n_H=6
a_th=n_C+n_H/4
coeff=0.95  "coeff=1 for theoretical combustion, 1.5 for 50% excess air"
"Assuming all the H burns to H2O, the combustion equation is
   C6H6+coeff*a_th(O2+3.76N2)----- (n_CO2) CO2+(n_CO)CO+(n_H2O) H2O+(n_N2) N2"
n_O2=coeff*a_th
n_N2=3.76*n_O2
n_H2O=n_H/2
n_CO2+n_CO=n_C
2*n_CO2+n_CO+n_H2O=2*n_O2  "Oxygen balance"
n_prod=n_CO2+n_CO+n_H2O+n_N2  "Total mole numbers of product gases"
y_CO=n_CO/n_prod "mole fraction of CO in product gases"

"Some Wrong Solutions with Common Mistakes:"
W1_yCO=n_CO/n1_prod; n1_prod=n_CO2+n_CO+n_H2O "Not including N2 in n_prod"
W2_yCO=(n_CO2+n_CO)/n_prod "Using both CO and CO2 in calculations"
```

15-138 A fuel is burned during a steady-flow combustion process. Heat is lost to the surroundings at 300 K at a rate of 1120 kW. The entropy of the reactants entering per unit time is 17 kW/K and that of the products is 15 kW/K. The total rate of exergy destruction during this combustion process is

(a) 520 kW (b) 600 kW (c) 1120 kW (d) 340 kW (e) 739 kW

Answer (a) 520 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
To=300 "K"
Q_out=1120 "kW"
S_react=17 "kW'K"
S_prod= 15 "kW/K"
S_react-S_prod-Q_out/To+S_gen=0 "Entropy balance for steady state operation, Sin-Sout+Sgen=0"
X_dest=To*S_gen

"Some Wrong Solutions with Common Mistakes:"
W1_Xdest=S_gen "Taking Sgen as exergy destruction"
W2_Xdest=To*S_gen1; S_react-S_prod-S_gen1=0 "Ignoring Q_out/To"
```

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15-139 ··· 15-144 Design and Essay Problems

15-139 A certain industrial process generates a liquid solution of ethanol and water as the waste product. The solution is to be burned using methane. A combustion process is to be developed to accomplish this incineration process with minimum amount of methane.

Analysis The mass flow rate of the liquid ethanol-water solution is given to be 10 kg/s. Considering that the mass fraction of ethanol in the solution is 0.2,

$$\dot{m}_{\text{ethanol}} = (0.2)(10 \text{ kg/s}) = 2 \text{ kg/s}$$
$$\dot{m}_{\text{water}} = (0.8)(10 \text{ kg/s}) = 8 \text{ kg/s}$$

Noting that the molar masses $M_{\text{ethanol}} = 46$ and $M_{\text{water}} = 18$ kg/kmol and that mole numbers $N = m/M$, the mole flow rates become

$$\dot{N}_{\text{ethanol}} = \frac{\dot{m}_{\text{ethanol}}}{M_{\text{ethanol}}} = \frac{2 \text{ kg/s}}{46 \text{ kg/kmol}} = 0.04348 \text{ kmol/s}$$
$$\dot{N}_{\text{water}} = \frac{\dot{m}_{\text{water}}}{M_{\text{water}}} = \frac{8 \text{ kg/s}}{18 \text{ kg/kmol}} = 0.44444 \text{ kmol/s}$$

Note that

$$\frac{\dot{N}_{\text{water}}}{\dot{N}_{\text{ethanol}}} = \frac{0.44444}{0.04348} = 10.222 \text{ kmol } H_2O/\text{kmol } C_2H_5OH$$

That is, 10.222 moles of liquid water is present in the solution for each mole of ethanol.

Assuming complete combustion, the combustion equation of C_2H_5OH (ℓ) with stoichiometric amount of air is

$$C_2H_5OH(\ell) + a_{\text{th}}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 3H_2O + 3.76a_{\text{th}}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1 + 2a_{\text{th}} = 4 + 3 \longrightarrow a_{\text{th}} = 3$$

Thus,

$$C_2H_5OH(\ell) + 3(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 3H_2O + 11.28N_2$$

Noting that 10.222 kmol of liquid water accompanies each kmol of ethanol, the actual combustion equation can be written as

$$C_2H_5OH(\ell) + 3(O_2 + 3.76N_2) + 10.222H_2O(\ell) \longrightarrow 2CO_2 + 3H_2O(g) + 11.28N_2 + 10.222H_2O(l)$$

The heat transfer for this combustion process is determined from the steady-flow energy balance equation with $W = 0$,

$$Q = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. We assume all the reactants to enter the combustion chamber at the standard reference temperature of 25°C. Furthermore, we assume the products to leave the combustion chamber at 1400 K which is a little over the required temperature of 1100°C. From the tables,

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Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol	$\bar{h}_{1400\ K}$ kJ/kmol
C_2H_5OH (ℓ)	-277,690	---	---
CH_4	-74,850	---	---
O_2	0	8682	45,648
N_2	0	8669	43,605
H_2O (g)	-241,820	9904	53,351
H_2O (ℓ)	-285,830	---	---
CO_2	-393,520	9364	65,271

Thus,

$$\begin{aligned} Q &= (2)(-393{,}520 + 65{,}271 - 9364) + (3)(-241{,}820 + 53{,}351 - 9904) \\ &\quad + (11.28)(0 + 43{,}605 - 8669) - (1)(-277{,}690) - 0 - 0 \\ &\quad + (10.222)(-241{,}820 + 53{,}351 - 9904) - (10.222)(-285{,}830) \\ &= 295{,}409 \text{ kJ/kmol of } C_2H_5OH \end{aligned}$$

The positive sign indicates that 295,409 kJ of heat must be supplied to the combustion chamber from another source (such as burning methane) to ensure that the combustion products will leave at the desired temperature of 1400 K. Then the rate of heat transfer required for a mole flow rate of 0.04348 kmol C_2H_5OH/s CO becomes

$$\dot{Q} = \dot{N}Q = (0.04348 \text{ kmol/s})(295{,}409 \text{ kJ/kmol}) = 12{,}844 \text{ kJ/s}$$

Assuming complete combustion, the combustion equation of $CH_4(g)$ with stoichiometric amount of air is

$$CH_4 + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

Thus,

$$a_{th} = 1 + 1 \longrightarrow a_{th} = 2$$

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 7.52N_2$$

The heat transfer for this combustion process is determined from the steady-flow energy balance $E_{in} - E_{out} = \Delta E_{system}$ equation as shown above under the same assumptions and using the same mini table:

$$\begin{aligned} Q &= (1)(-393{,}520 + 65{,}271 - 9364) + (2)(-241{,}820 + 53{,}351 - 9904) \\ &\quad + (7.52)(0 + 43{,}605 - 8669) - (1)(-74{,}850) - 0 - 0 \\ &= -396{,}790 \text{ kJ/kmol of } CH_4 \end{aligned}$$

That is, 396,790 kJ of heat is supplied to the combustion chamber for each kmol of methane burned. To supply heat at the required rate of 12,844 kJ/s, we must burn methane at a rate of

$$\dot{N}_{CH_4} = \frac{\dot{Q}}{Q} = \frac{12{,}844 \text{ kJ/s}}{396{,}790 \text{ kJ/kmol}} = 0.03237 \text{ kmol}CH_4\text{/s}$$

or,

$$\dot{m}_{CH_4} = M_{CH_4}\dot{N}_{CH_4} = (16 \text{ kg/kmol})(0.03237 \text{ kmol}CH_4\text{/s}) = \mathbf{0.5179\ kg/s}$$

Therefore, we must supply methane to the combustion chamber at a minimum rate 0.5179 kg/s in order to maintain the temperature of the combustion chamber above 1400 K.



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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 16
CHEMICAL AND PHASE EQUILIBRIUM

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K_p and Equilibrium Composition of Ideal Gases

16-1C No, the wooden table is NOT in chemical equilibrium with the air. With proper catalyst, it will reach with the oxygen in the air and burn.

16-2C They are

$$K_p = \frac{P_C^{\nu_C} P_D^{\nu_D}}{P_A^{\nu_A} P_B^{\nu_B}}, \quad K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{and} \quad K_p = \frac{N_C^{\nu_C} N_D^{\nu_D}}{N_A^{\nu_A} N_B^{\nu_B}} \left(\frac{P}{N_{\text{total}}} \right)^{\Delta \nu}$$

where $\Delta \nu = \nu_C + \nu_D - \nu_A - \nu_B$. The first relation is useful in partial pressure calculations, the second in determining the K_p from gibbs functions, and the last one in equilibrium composition calculations.

16-3C (*a*) No, because K_p depends on temperature only.

(*b*) In general, the total mixture pressure affects the mixture composition. The equilibrium constant for the reaction $N_2 + O_2 \Leftrightarrow 2NO$ can be expressed as

$$K_p = \frac{N_{\text{NO}}^{\nu_{\text{NO}}}}{N_{\text{N}_2}^{\nu_{N_2}} N_{\text{O}_2}^{\nu_{\text{O}_2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_{\text{NO}} - \nu_{\text{N}_2} - \nu_{\text{O}_2})}$$

The value of the exponent in this case is 2-1-1 = 0. Therefore, changing the total mixture pressure will have no effect on the number of moles of N_2, O_2 and NO.

16-4C (*a*) The equilibrium constant for the reaction $CO + \frac{1}{2} O_2 \Leftrightarrow CO_2$ can be expressed as

$$K_p = \frac{N_{\text{CO}_2}^{\nu_{\text{CO}_2}}}{N_{\text{CO}}^{\nu_{\text{CO}}} N_{\text{O}_2}^{\nu_{\text{O}_2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_{\text{CO}_2} - \nu_{\text{CO}} - \nu_{\text{O}_2})}$$

Judging from the values in Table A-28, the K_p value for this reaction decreases as temperature increases. That is, the indicated reaction will be less complete at higher temperatures. Therefore, the number of moles of CO_2 will decrease and the number moles of CO and O_2 will increase as the temperature increases.

(*b*) The value of the exponent in this case is 1-1-0.5=-0.5, which is negative. Thus as the pressure increases, the term in the brackets will decrease. The value of K_p depends on temperature only, and therefore it will not change with pressure. Then to keep the equation balanced, the number of moles of the products (CO_2) must increase, and the number of moles of the reactants (CO, O_2) must decrease.

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16-5C (*a*) The equilibrium constant for the reaction $N_2 \Leftrightarrow 2N$ can be expressed as

$$K_p = \frac{N_N^{\nu_N}}{N_{N_2}^{\nu_{N_2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_N - \nu_{N_2})}$$

Judging from the values in Table A-28, the K_p value for this reaction increases as the temperature increases. That is, the indicated reaction will be more complete at higher temperatures. Therefore, the number of moles of N will increase and the number moles of N_2 will decrease as the temperature increases.

(*b*) The value of the exponent in this case is 2-1 = 1, which is positive. Thus as the pressure increases, the term in the brackets also increases. The value of K_p depends on temperature only, and therefore it will not change with pressure. Then to keep the equation balanced, the number of moles of the products (N) must decrease, and the number of moles of the reactants (N_2) must increase.

16-6C The equilibrium constant for the reaction $CO + \frac{1}{2} O_2 \Leftrightarrow CO_2$ can be expressed as

$$K_p = \frac{N_{CO_2}^{\nu_{CO_2}}}{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_{CO_2} - \nu_{CO} - \nu_{O_2})}$$

Adding more N_2 (an inert gas) at constant temperature and pressure will increase N_{total} but will have no direct effect on other terms. Then to keep the equation balanced, the number of moles of the products (CO_2) must increase, and the number of moles of the reactants (CO, O_2) must decrease.

16-7C The values of the equilibrium constants for each dissociation reaction at 3000 K are, from Table A-28,

$N_2 \Leftrightarrow 2N \Leftrightarrow \ln K_p = -22.359$

$H_2 \Leftrightarrow 2H \Leftrightarrow \ln K_p = -3.685$ (greater than - 22.359)

Thus H_2 is more likely to dissociate than N_2.

16-8C (*a*) This reaction is the reverse of the known CO reaction. The equilibrium constant is then

$1/K_P$

(*b*) This reaction is the reverse of the known CO reaction at a different pressure. Since pressure has no effect on the equilibrium constant,

$1/K_P$

(*c*) This reaction is the same as the known CO reaction multiplied by 2. The quilibirium constant is then

K_P^2

(*d*) This is the same as reaction (*c*) occurring at a different pressure. Since pressure has no effect on the equilibrium constant,

K_P^2

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16-9C (*a*) This reaction is the reverse of the known H_2O reaction. The equilibrium constant is then

$$1/K_P$$

(*b*) This reaction is the reverse of the known H_2O reaction at a different pressure. Since pressure has no effect on the equilibrium constant,

$$1/K_P$$

(*c*) This reaction is the same as the known H_2O reaction multiplied by 3. The quilibirium constant is then

$$K_P^3$$

(*d*) This is the same as reaction (*c*) occurring at a different pressure. Since pressure has no effect on the equilibrium constant,

$$K_P^3$$

16-10 The partial pressures of the constituents of an ideal gas mixture is given. The Gibbs function of the nitrogen in this mixture at the given mixture pressure and temperature is to be determined.

Analysis The partial pressure of nitrogen is

$$P_{N2} = 110\ \text{kPa} = (110/101.325) = 1.086\ \text{atm}$$

The Gibbs function of nitrogen at 293 K and 1.086 atm is

$$\begin{aligned}\bar{g}(293\ \text{K}, 1.086\ \text{atm}) &= \bar{g}^*(293\ \text{K}, 1\ \text{atm}) + R_u T \ln P_{N2} \\ &= 0 + (8.314\ \text{kJ/kmol.K})(293\ \text{K})\ln(1.086\ \text{atm}) \\ &= \mathbf{200\ kJ/kmol}\end{aligned}$$

N_2 ,CO_2, NO

P_{N2} = 110 kPa
293 K

16-11 The mole fractions of the constituents of an ideal gas mixture is given. The Gibbs function of the N_2 in this mixture at the given mixture pressure and temperature is to be determined.

Analysis From Tables A-18 and A-26, at 1 atm pressure,

$$\begin{aligned}\bar{g}^*(600\ \text{K}, 1\ \text{atm}) &= \bar{g}_f^{\circ} + \Delta\left[\bar{h}(T) - T\bar{s}^{\circ}(T)\right] \\ &= 0 + (17{,}563 - 600 \times 212.066) - (8669 - 298 \times 191.502) \\ &= -61{,}278\ \text{kJ/kmol}\end{aligned}$$

30% N_2
30% O_2
40% H_2O
5 atm
600 K

The partial pressure of N_2 is

$$P_{CO} = y_{N2} P = (0.30)(5\ \text{atm}) = 1.5\ \text{atm}$$

The Gibbs function of N_2 at 600 K and 1.5 atm is

$$\begin{aligned}\bar{g}(600\ \text{K}, 1.5\ \text{atm}) &= \bar{g}^*(600\ \text{K}, 1\ \text{atm}) + R_u T \ln P_{CO} \\ &= -61{,}278\ \text{kJ/kmol} + (8.314\ \text{kJ/kmol})(600\ \text{K})\ln(1.5\ \text{atm}) \\ &= \mathbf{-59{,}260\ kJ/kmol}\end{aligned}$$

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16-12 The temperature at which 0.2 percent of diatomic oxygen dissociates into monatomic oxygen at two pressures is to be determined.

Assumptions **1** The equilibrium composition consists of N_2 and N. **2** The constituents of the mixture are ideal gases.

Analysis (*a*) The stoichiometric and actual reactions can be written as

Stoichiometric: $N_2 \Leftrightarrow 2N$ (thus $\nu_{N2} = 1$ and $\nu_N = 2$)

Actual: $N_2 \Leftrightarrow \underbrace{0.998N_2}_{\text{react.}} + \underbrace{0.004N}_{\text{prod.}}$

$N_2 \leftrightarrow 2N$
0.2 %
1 kPa

The equilibrium constant K_p can be determined from

$$K_p = \frac{N_N^{\nu_N}}{N_{N2}^{\nu_{N2}}}\left(\frac{P}{N_{\text{total}}}\right)^{\nu_N - \nu_{N2}} = \frac{0.004^2}{0.998}\left(\frac{1/101.325}{0.998+0.004}\right)^{2-1} = 1.579 \times 10^{-7}$$

and

$$\ln K_p = -15.66$$

From Table A-28, the temperature corresponding to this $\ln K_p$ value is

T = **3628 K**

(*b*) At 10 kPa,

$$K_p = \frac{N_N^{\nu_N}}{N_{N2}^{\nu_{N2}}}\left(\frac{P}{N_{\text{total}}}\right)^{\nu_N - \nu_{N2}} = \frac{0.004^2}{0.998}\left(\frac{10/101.325}{0.998+0.004}\right)^{2-1} = 1.579 \times 10^{-6}$$

$$\ln K_p = -13.36$$

From Table A-28, the temperature corresponding to this $\ln K_p$ value is

T = **3909 K**

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16-13 The equilibrium constant of the reaction $H_2O \Leftrightarrow H_2 + \frac{1}{2}O_2$ is to be determined using Gibbs function.

Analysis (*a*) The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

$H_2O \leftrightarrow H_2 + ½O_2$

500 K

where

$$\Delta G^*(T) = \nu_{H2}\bar{g}^*_{H2}(T) + \nu_{O2}\bar{g}^*_{O2}(T) - \nu_{H2O}\bar{g}^*_{H2O}(T)$$

At 500 K,

$$\begin{aligned}
\Delta G^*(T) &= \nu_{H2}\bar{g}^*_{H2}(T) + \nu_{O2}\bar{g}^*_{O2}(T) - \nu_{H2O}\bar{g}^*_{H2O}(T) \\
&= \nu_{H2}(\bar{h} - T\bar{s})_{H2} + \nu_{O2}(\bar{h} - T\bar{s})_{O2} - \nu_{H2O}(\bar{h} - T\bar{s})_{H2O} \\
&= \nu_{H2}[(\bar{h}_f + \bar{h}_{500} - \bar{h}_{298}) - T\bar{s}]_{H2} \\
&\quad + \nu_{O2}[(\bar{h}_f + \bar{h}_{500} - \bar{h}_{298}) - T\bar{s}]_{O2} \\
&\quad - \nu_{H2O}[(\bar{h}_f + \bar{h}_{500} - \bar{h}_{298}) - T\bar{s}]_{H2O} \\
&= 1\times(0 + 14{,}350 - 8468 - 500\times 145.628) \\
&\quad + 0.5\times(0 + 14{,}770 - 8682 - 500\times 220.589) \\
&\quad - 1\times(-241{,}820 + 16{,}828 - 9904 - 500\times 206.413) \\
&= 219{,}067 \text{ kJ/kmol}
\end{aligned}$$

Substituting,

$$\ln K_p = -(219{,}067 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(500 \text{ K})] = \mathbf{-52.70}$$

or

$$K_p = \mathbf{1.30\times 10^{-23}} \quad (\text{Table A - 28}: \ln K_p = -52.70)$$

At 2000 K,

$$\begin{aligned}
\Delta G^*(T) &= \nu_{H2}\bar{g}^*_{H2}(T) + \nu_{O2}\bar{g}^*_{O2}(T) - \nu_{H2O}\bar{g}^*_{H2O}(T) \\
&= \nu_{H2}(\bar{h} - T\bar{s})_{H2} + \nu_{O2}(\bar{h} - T\bar{s})_{O2} - \nu_{H2O}(\bar{h} - T\bar{s})_{H2O} \\
&= \nu_{H2}[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{H2} \\
&\quad + \nu_{O2}[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{O2} \\
&\quad - \nu_{H2O}[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{H2O} \\
&= 1\times(0 + 61{,}400 - 8468 - 2000\times 188.297) \\
&\quad + 0.5\times(0 + 67{,}881 - 8682 - 2000\times 268.655) \\
&\quad - 1\times(-241{,}820 + 82{,}593 - 9904 - 2000\times 264.571) \\
&= 135{,}556 \text{ kJ/kmol}
\end{aligned}$$

Substituting,

$$\ln K_p = -(135{,}556 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(2000 \text{ K})] = \mathbf{-8.15}$$

or

$$K_p = \mathbf{2.88\times 10^{-4}} \quad (\text{Table A - 28}: \ln K_p = -8.15)$$

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16-14 The reaction $C + O_2 \Leftrightarrow CO_2$ is considered. The mole fraction of the carbon dioxide produced when this reaction occurs at a1 atm and 3800 K are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, C and O_2. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

$C + O_2 \Leftrightarrow CO_2$
3800 K
1 atm

Stoichiometric: $C + O_2 \Leftrightarrow CO_2$ (thus $\nu_C = 1,\ \nu_{O2} = 1,$ and $\nu_{CO2} = 1$)

Actual: $C + O_2 \longrightarrow \underbrace{xC + yO_2}_{\text{react.}} + \underbrace{zCO_2}_{\text{products}}$

C balance: $1 = x + z \longrightarrow z = 1 - x$

O balance: $2 = 2y + 2z \longrightarrow y = 1 - z = 1 - (1 - x) = x$

Total number of moles: $N_{\text{total}} = x + y + z = 1 + x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO2}^{\nu_{CO2}}}{N_C^{\nu_C} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_{CO2} - \nu_C - \nu_{O2})}$$

From the problem statement at 3800 K, $\ln K_p = -0.461$. Then,

$$K_p = \exp(-0.461) = 0.6307$$

Substituting,

$$0.6307 = \frac{(1 - x)}{(x)(x)} \left(\frac{1}{1 + x} \right)^{1-1-1}$$

Solving for x,

$$x = 0.7831$$

Then,

$$y = x = 0.7831$$

$$z = 1 - x = 0.2169$$

Therefore, the equilibrium composition of the mixture at 3800 K and 1 atm is

$$0.7831\,C + 0.7831\,O_2 + 0.2169\,CO_2$$

The mole fraction of carbon dioxide is

$$y_{CO2} = \frac{N_{CO2}}{N_{\text{total}}} = \frac{0.2169}{1 + 0.7831} = \mathbf{0.1216}$$

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16-15 The reaction $C + O_2 \Leftrightarrow CO_2$ is considered. The mole fraction of the carbon dioxide produced when this reaction occurs at a1 atm and 3800 K and 700 kPa and 3800 K are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, C and O_2. **2** The constituents of the mixture are ideal gases.

Analysis We first solve the problem for 1 atm pressure:

The stoichiometric and actual reactions in this case are

$C + O_2 \Leftrightarrow CO_2$
3800 K
1 atm

Stoichiometric: $C + O_2 \Leftrightarrow CO_2$ (thus $\nu_C = 1,\ \nu_{O2} = 1,$ and $\nu_{CO2} = 1$)

Actual: $C + O_2 \longrightarrow \underbrace{xC + yO_2}_{\text{react.}} + \underbrace{zCO_2}_{\text{products}}$

C balance: $1 = x + z \longrightarrow z = 1 - x$

O balance: $2 = 2y + 2z \longrightarrow y = 1 - z = 1 - (1 - x) = x$

Total number of moles: $N_{\text{total}} = x + y + z = 1 + x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO2}^{\nu_{CO2}}}{N_C^{\nu_C} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_{CO2} - \nu_C - \nu_{O2})}$$

From the problem statement at 3800 K, $\ln K_p = -0.461$. Then,

$$K_p = \exp(-0.461) = 0.6307$$

Substituting,

$$0.6307 = \frac{(1-x)}{(x)(x)} \left(\frac{1}{1+x} \right)^{1-1-1}$$

Solving for x,

$$x = 0.7831$$

Then,

$$y = x = 0.7831$$

$$z = 1 - x = 0.2169$$

Therefore, the equilibrium composition of the mixture at 3800 K and 1 atm is

$$0.7831\,C + 0.7831\,O_2 + 0.2169\,CO_2$$

The mole fraction of carbon dioxide is

$$y_{CO2} = \frac{N_{CO2}}{N_{\text{total}}} = \frac{0.2169}{1 + 0.7831} = \mathbf{0.1216}$$

We repeat the calculations at 700 kPa pressure:

The pressure in this case is 700 kPa/(101.325 kPa/atm) = 6.908 atm. Then,

$$K_p = \frac{N_{CO2}^{\nu_{CO2}}}{N_C^{\nu_C} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_{CO2} - \nu_C - \nu_{O2})}$$

$$0.6307 = \frac{(1-x)}{(x)(x)} \left(\frac{6.908}{1+x} \right)^{1-1-1}$$

$C + O_2 \Leftrightarrow CO_2$
3800 K
700 kPa

$$x = 0.4320$$

$$y = x = 0.4320$$

$$z = 1 - x = 0.5680$$

Therefore, the equilibrium composition of the mixture at 3800 K and 700 kPa is

$$0.4320\,C + 0.4320\,O_2 + 0.5680\,CO_2$$

The mole fraction of carbon dioxide is

$$y_{CO2} = \frac{N_{CO2}}{N_{\text{total}}} = \frac{0.5680}{1 + 0.4320} = \mathbf{0.3966}$$

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16-16 The reaction $C + O_2 \Leftrightarrow CO_2$ is considered. The mole fraction of the carbon dioxide produced when this reaction occurs at a1 atm and 3800 K and 700 kPa and 3800 K are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, C and O_2. **2** The constituents of the mixture are ideal gases.

Analysis We first solve the problem for the reaction $C + O_2 \Leftrightarrow CO_2$:

The stoichiometric and actual reactions in this case are

$C + O_2 \Leftrightarrow CO_2$
3800 K
1 atm

Stoichiometric: $C + O_2 \Leftrightarrow CO_2$ (thus $\nu_C = 1,\ \nu_{O2} = 1,$ and $\nu_{CO2} = 1$)

Actual: $C + O_2 \longrightarrow \underbrace{xC + yO_2}_{\text{react.}} + \underbrace{zCO_2}_{\text{products}}$

C balance: $1 = x + z \longrightarrow z = 1 - x$

O balance: $2 = 2y + 2z \longrightarrow y = 1 - z = 1 - (1 - x) = x$

Total number of moles: $N_{\text{total}} = x + y + z = 1 + x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO2}^{\nu_{CO2}}}{N_C^{\nu_C} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}} \right)^{(\nu_{CO2} - \nu_C - \nu_{O2})}$$

From the problem statement at 3800 K, $\ln K_p = -0.461$. Then,

$$K_p = \exp(-0.461) = 0.6307$$

Substituting,

$$0.6307 = \frac{(1 - x)}{(x)(x)} \left(\frac{1}{1 + x} \right)^{1-1-1}$$

Solving for x,

$$x = 0.7831$$

Then,

$$y = x = 0.7831$$

$$z = 1 - x = 0.2169$$

Therefore, the equilibrium composition of the mixture at 3800 K and 1 atm is

$$0.7831\,C + 0.7831\,O_2 + 0.2169\,CO_2$$

The mole fraction of carbon dioxide is

$$y_{CO2} = \frac{N_{CO2}}{N_{\text{total}}} = \frac{0.2169}{1 + 0.7831} = \mathbf{0.1216}$$

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If the reaction is: C + (O_2 + 3.76 N_2) ⇔ CO_2 + 3.76 N_2

The stoichiometric and actual reactions in this case are

Stoichiometric:

$$\mathrm{C} + (\mathrm{O_2} + 3.76\,\mathrm{N_2}) \Leftrightarrow \mathrm{CO_2} + 3.76\,\mathrm{N_2} \quad (\text{thus } \nu_{\mathrm{C}} = 1,\ \nu_{\mathrm{O2}} = 1, \nu_{\mathrm{N2}} = 3.76, \nu_{\mathrm{CO2}} = 1, \text{and } \nu_{\mathrm{N2}} = 3.76)$$

Actual: $$\mathrm{C} + (\mathrm{O_2} + 3.76\,\mathrm{N_2}) \longrightarrow \underbrace{x\mathrm{C} + y\mathrm{O_2}}_{\text{react.}} + \underbrace{z\mathrm{CO_2} + 3.76\mathrm{N_2}}_{\text{products}}$$

$C+(O_2+3.76N_2)$ ⇔$CO_2+3.76N_2$
3800 K
1 atm

C balance: $1 = x + z \longrightarrow z = 1 - x$

O balance: $2 = 2y + 2z \longrightarrow y = 1 - z = 1 - (1 - x) = x$

Total number of moles: $N_{\text{total}} = x + y + z + 3.76 = 4.76 + x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{\mathrm{CO2}}^{\nu_{\mathrm{CO2}}} N_{\mathrm{N2}}^{\nu_{\mathrm{N2}}}}{N_{\mathrm{C}}^{\nu_{\mathrm{C}}} N_{\mathrm{O2}}^{\nu_{\mathrm{O2}}} N_{\mathrm{N2}}^{\nu_{\mathrm{N2}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{CO2}} + \nu_{\mathrm{N2}} - \nu_{\mathrm{C}} - \nu_{\mathrm{O2}} - \nu_{\mathrm{N2}})}$$

or $$K_p = \frac{N_{\mathrm{CO2}}^{\nu_{\mathrm{CO2}}}}{N_{\mathrm{C}}^{\nu_{\mathrm{C}}} N_{\mathrm{O2}}^{\nu_{\mathrm{O2}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{CO2}} - \nu_{\mathrm{C}} - \nu_{\mathrm{O2}}}$$

From the problem statement at 3800 K, $\ln K_p = 12.49$. Then,

$$K_p = \exp(12.49) = 265{,}670$$

Substituting,

$$265{,}670 = \frac{(1-x)}{(x)(x)} \left(\frac{1}{4.76 + x}\right)^{-1}$$

Solving for x,

$$x = 0.004226$$

Then,

$$y = x = 0.004226$$

$$z = 1 - x = 0.9958$$

Therefore, the equilibrium composition of the mixture at 3800 K and 1 atm is

$$0.004226\,\mathrm{C} + 0.004226\,\mathrm{O_2} + 3.76\,\mathrm{N_2} + 0.9958\,\mathrm{CO_2} + 3.76\,\mathrm{N_2}$$

The mole fraction of carbon dioxide is

$$y_{\mathrm{CO2}} = \frac{N_{\mathrm{CO2}}}{N_{\text{total}}} = \frac{0.9958}{4.76 + 0.004226} = \mathbf{0.2090}$$

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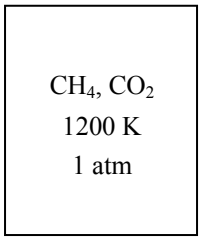
16-17 A gaseous mixture consisting of methane and carbon dioxide is heated. The equilibrium composition (by mole fraction) of the resulting mixture is to be determined.

Assumptions **1** The equilibrium composition consists of CH_4, C, H_2, and CO_2. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

Stoichiometric: $\mathrm{CH_4 \Leftrightarrow C + 2H_2}$ (thus $\nu_{CH4} = 1$, $\nu_C = 1$, and $\nu_{H2} = 2$)

Actual: $0.3\mathrm{CH_4} + 0.7\mathrm{CO_2} \longrightarrow \underbrace{x\mathrm{CH_4}}_{\text{react.}} + \underbrace{y\mathrm{C} + z\mathrm{H_2}}_{\text{products}} + \underbrace{0.7\mathrm{CO_2}}_{\text{inert}}$



C balance: $0.3 = x + y \longrightarrow y = 0.3 - x$

H balance: $1.2 = 4x + 2z \longrightarrow z = 0.6 - 2x$

Total number of moles: $N_{\text{total}} = x + y + z + 1 = 1.6 - 2x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CH4}^{\nu_{CH4}}}{N_C^{\nu_C} N_{H2}^{\nu_{H2}}} \left(\frac{P}{N_{\text{total}}} \right)^{\nu_{CH4} - \nu_C - \nu_{H2}}$$

From the problem statement at 1200 K, $\ln K_p = 4.147$. Then,

$$K_p = \exp(4.147) = 63.244$$

For the reverse reaction that we consider,

$$K_p = 1/63.244 = 0.01581$$

Substituting,

$$0.01581 = \frac{x}{(0.3 - x)(0.6 - 2x)^2} \left(\frac{1}{1.6 - 2x} \right)^{1-1-2}$$

Solving for x,

$$x = 0.0006637$$

Then,

$$y = 0.3 - x = 0.2993$$

$$z = 0.6 - 2x = 0.5987$$

Therefore, the equilibrium composition of the mixture at 1200 K and 1 atm is

$$0.0006637\,\mathrm{CH_4} + 0.2993\,\mathrm{C} + 0.5987\,\mathrm{H_2} + 0.7\,\mathrm{CO_2}$$

The mole fractions are

$$y_{CH4} = \frac{N_{CH4}}{N_{\text{total}}} = \frac{0.0006637}{1.6 - 2 \times 0.0006637} = \frac{0.0006637}{1.599} = \mathbf{0.000415}$$

$$y_C = \frac{N_C}{N_{\text{total}}} = \frac{0.2993}{1.599} = \mathbf{0.1872}$$

$$y_{H2} = \frac{N_{H2}}{N_{\text{total}}} = \frac{0.5987}{1.599} = \mathbf{0.3745}$$

$$y_{CO2} = \frac{N_{CO2}}{N_{\text{total}}} = \frac{0.7}{1.599} = \mathbf{0.4379}$$

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16-18 The dissociation reaction $CO_2 \Leftrightarrow CO + O$ is considered. The composition of the products at given pressure and temperature is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, and O. **2** The constituents of the mixture are ideal gases.

Analysis For the stoichiometric reaction $CO_2 \Leftrightarrow CO + \frac{1}{2}O_2$, from Table A-28, at 2500 K

$$\ln K_p = -3.331$$

CO_2
2500 K
1 atm

For the oxygen dissociation reaction $0.5O_2 \Leftrightarrow O$, from Table A-28, at 2500 K,

$$\ln K_p = -8.509/2 = -4.255$$

For the desired stoichiometric reaction $CO_2 \Leftrightarrow CO + O$ (thus $\nu_{CO2} = 1$, $\nu_{CO} = 1$ and $\nu_O = 1$),

$$\ln K_p = -3.331 - 4.255 = -7.586$$

and

$$K_p = \exp(-7.586) = 0.0005075$$

Actual: $CO_2 \longrightarrow \underbrace{xCO_2}_{\text{react.}} + \underbrace{yCO + zO}_{\text{products}}$

C balance: $1 = x + y \longrightarrow y = 1 - x$

O balance: $2 = 2x + y + z \longrightarrow z = 1 - x$

Total number of moles: $N_{\text{total}} = x + y + z = 2 - x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO}^{\nu_{CO}} N_O^{\nu_O}}{N_{CO2}^{\nu_{CO2}}} \left(\frac{P}{N_{\text{total}}}\right)^{\nu_{CO}+\nu_O-\nu_{CO2}}$$

Substituting,

$$0.0005075 = \frac{(1-x)(1-x)}{x}\left(\frac{1}{2-x}\right)^{1+1-1}$$

Solving for x,

$$x = 0.9775$$

Then,

$$y = 1 - x = 0.0225$$

$$z = 1 - x = 0.0225$$

Therefore, the equilibrium composition of the mixture at 2500 K and 1 atm is

$$\mathbf{0.9775\ CO_2 + 0.0225\ CO + 0.0225\ O}$$

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16-19 The dissociation reaction $CO_2 \Leftrightarrow CO + O$ is considered. The composition of the products at given pressure and temperature is to be determined when nitrogen is added to carbon dioxide.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis For the stoichiometric reaction $CO_2 \Leftrightarrow CO + \frac{1}{2}O_2$, from Table A-28, at 2500 K

$$\ln K_p = -3.331$$

For the oxygen dissociation reaction $0.5O_2 \Leftrightarrow O$, from Table A-28, at 2500 K,

$$\ln K_p = -8.509/2 = -4.255$$

For the desired stoichiometric reaction $CO_2 \Leftrightarrow CO + O$ (thus $\nu_{CO2} = 1$, $\nu_{CO} = 1$ and $\nu_O = 1$),

$$\ln K_p = -3.331 - 4.255 = -7.586$$

and

$$K_p = \exp(-7.586) = 0.0005075$$

Actual: $CO_2 + 3N_2 \longrightarrow \underbrace{xCO_2}_{\text{react.}} + \underbrace{yCO + zO}_{\text{products}} + \underbrace{3N_2}_{\text{inert}}$

C balance: $1 = x + y \longrightarrow y = 1 - x$

O balance: $2 = 2x + y + z \longrightarrow z = 1 - x$

Total number of moles: $N_{\text{total}} = x + y + z + 3 = 5 - x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO}^{\nu_{CO}} N_O^{\nu_O}}{N_{CO2}^{\nu_{CO2}}} \left(\frac{P}{N_{\text{total}}}\right)^{\nu_{CO}+\nu_O-\nu_{CO2}}$$

Substituting,

$$0.0005075 = \frac{(1-x)(1-x)}{x}\left(\frac{1}{5-x}\right)^{1+1-1}$$

Solving for x,

$$x = 0.9557$$

Then,

$$y = 1 - x = 0.0443$$

$$z = 1 - x = 0.0443$$

Therefore, the equilibrium composition of the mixture at 2500 K and 1 atm is

$$\mathbf{0.9557\,CO_2 + 0.0443\,CO + 0.0443\,O + 3N_2}$$

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16-20 The reaction $N_2 + O_2 \Leftrightarrow 2NO$ is considered. The equilibrium mole fraction of NO 1600 K and 1 atm is to be determined.

Assumptions **1** The equilibrium composition consists of N_2, O_2, and NO. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

Stoichiometric: $\mathrm{N_2 + O_2 \Leftrightarrow 2NO}$ (thus $\nu_{N2} = 1,\ \nu_{O2} = 1,$ and $\nu_{NO} = 2$)

Actual: $\mathrm{N_2 + O_2 \longrightarrow \underbrace{\mathit{x}N_2 + \mathit{y}O_2}_{\text{react.}} + \underbrace{\mathit{z}NO}_{\text{products}}}$

N_2, O_2
1600 K
1 atm

N balance: $2 = 2x + z \longrightarrow z = 2 - 2x$

O balance: $2 = 2y + z \longrightarrow y = x$

Total number of moles: $N_{\text{total}} = x + y + z = 2$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N2}^{\nu_{N2}} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{NO} - \nu_{N2} - \nu_{O2})}$$

From Table A-28, at 1600 K, $\ln K_p = -5.294$. Since the stoichiometric reaction being considered is double this reaction,

$$K_p = \exp(-2 \times 5.294) = 2.522 \times 10^{-5}$$

Substituting,

$$2.522 \times 10^{-5} = \frac{(2 - 2x)^2}{x^2} \left(\frac{1}{2}\right)^{2-1-1}$$

Solving for x,

$$x = 0.9975$$

Then,

$$y = x = 0.9975$$

$$z = 2 - 2x = 0.005009$$

Therefore, the equilibrium composition of the mixture at 1000 K and 1 atm is

$$0.9975\,\mathrm{N_2} + 0.9975\,\mathrm{O_2} + 0.005009\,\mathrm{NO}$$

The mole fraction of NO is then

$$y_{NO} = \frac{N_{NO}}{N_{\text{total}}} = \frac{0.005009}{2} = \mathbf{0.002505}$$

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16-21E The equilibrium constant of the reaction $H_2 + 1/2O_2 \leftrightarrow H_2O$ is listed in Table A-28 at different temperatures. The data are to be verified at two temperatures using Gibbs function data.

Analysis (*a*) The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T) / R_u T$$

where

$$\Delta G^*(T) = \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{H_2}\bar{g}^*_{H_2}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T)$$

$H_2 + ½O_2 \leftrightarrow H_2O$

537 R

At 537 R,

$$\Delta G^*(T) = 1(-98{,}350) - 1(0) - 0.5(0) = -98{,}350 \text{ Btu / lbmol}$$

Substituting,

$$\ln K_p = -(-98{,}350 \text{ Btu / lbmol}) / [(1.986 \text{ Btu / lbmol} \cdot \text{R})(537 \text{ R})] = 92.22$$

or

$$K_p = \mathbf{1.12 \times 10^{40}} \quad (\text{Table A - 28: } \ln K_p = 92.21)$$

(*b*) At 4320 R,

$$\begin{aligned}
\Delta G^*(T) &= \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{H_2}\bar{g}^*_{H_2}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T) \\
&= \nu_{H_2O}(\bar{h} - T\bar{s})_{H_2O} - \nu_{H_2}(\bar{h} - T\bar{s})_{H_2} - \nu_{O_2}(\bar{h} - T\bar{s})_{O_2} \\
&= \nu_{H_2O}[(\bar{h}_f + \bar{h}_{4320} - \bar{h}_{537}) - T\bar{s}]_{H_2O} \\
&\quad - \nu_{H_2}[(\bar{h}_f + \bar{h}_{4320} - \bar{h}_{298}) - T\bar{s}]_{H_2} \\
&\quad - \nu_{O_2}[(\bar{h}_f + \bar{h}_{4320} - \bar{h}_{298}) - T\bar{s}]_{O_2} \\
&= 1 \times (-104{,}040 + 44{,}533 - 4258 - 4320 \times 65.504) \\
&\quad - 1 \times (0 + 32{,}647.2 - 3640.3 - 4320 \times 46.554) \\
&\quad - 0.5 \times (0 + 35{,}746 - 3725.1 - 4320 \times 65.831) \\
&= -48{,}451 \text{ Btu/lbmol}
\end{aligned}$$

Substituting,

$$\ln K_p = -(-48{,}451 \text{ Btu/lbmol})/[(1.986 \text{ Btu/lbmol.R})(4320 \text{ R})] = 5.647$$

or

$$K_p = \mathbf{283} \quad (\text{Table A - 28: } \ln K_p = 5.619$$

Discussion Solving this problem using EES with the built-in ideal gas properties give $K_p = 1.04 \times 10^{40}$ for part (*a*) and K_p = 278 for part (*b*).

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16-22 The equilibrium constant of the reaction $CO + 1/2O_2 \leftrightarrow CO_2$ at 298 K and 2000 K are to be determined, and compared with the values listed in Table A-28.

Analysis (*a*) The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

$$CO + \tfrac{1}{2}O_2 \Leftrightarrow CO_2$$
298 K

where

$$\Delta G^*(T) = \nu_{CO2}\bar{g}^*_{CO2}(T) - \nu_{CO}\bar{g}^*_{CO}(T) - \nu_{O2}\bar{g}^*_{O2}(T)$$

At 298 K,

$$\Delta G^*(T) = 1(-394{,}360) - 1(-137{,}150) - 0.5(0) = -257{,}210 \text{ kJ/kmol}$$

where the Gibbs functions are obtained from Table A-26. Substituting,

$$\ln K_p = -\frac{(-257{,}210 \text{ kJ/kmol})}{(8.314 \text{ kJ/kmol}\cdot\text{K})(298 \text{ K})} = \mathbf{103.81}$$

From Table A-28: $\ln K_p = \mathbf{103.76}$

(*b*) At 2000 K,

$$\begin{aligned}\Delta G^*(T) &= \nu_{CO2}\bar{g}^*_{CO2}(T) - \nu_{CO}\bar{g}^*_{CO}(T) - \nu_{O2}\bar{g}^*_{O2}(T)\\ &= \nu_{CO2}(\bar{h} - T\bar{s})_{CO2} - \nu_{CO}(\bar{h} - T\bar{s})_{CO} - \nu_{O2}(\bar{h} - T\bar{s})_{O2}\\ &= 1\left[(-302{,}128) - (2000)(309.00)\right] - 1\left[(-53{,}826) - (2000)(258.48)\right] - 0.5\left[(59{,}193) - (2000)(268.53)\right]\\ &= -110{,}409 \text{ kJ/kmol}\end{aligned}$$

The enthalpies at 2000 K and entropies at 2000 K and 101.3 kPa (1 atm) are obtained from EES. Substituting,

$$\ln K_p = -\frac{(-110{,}409 \text{ kJ/kmol})}{(8.314 \text{ kJ/kmol}\cdot\text{K})(2000 \text{ K})} = \mathbf{6.64}$$

From Table A-28:

$$\ln K_p = \mathbf{6.635}$$

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16-23 The effect of varying the percent excess air during the steady-flow combustion of hydrogen is to be studied.

Analysis The combustion equation of hydrogen with stoichiometric amount of air is

$$H_2 + 0.5[O_2 + 3.76N_2] \longrightarrow H_2O + 0.5(3.76)\,N_2$$

For the incomplete combustion with 100% excess air, the combustion equation is

$$H_2 + (1+Ex)(0.5)[O_2 + 3.76N_2] \longrightarrow 0.97\,H_2O + a\,H_2 + b\,O_2 + c\,N_2$$

The coefficients are to be determined from the mass balances

Hydrogen balance: $2 = 0.97 \times 2 + a \times 2 \longrightarrow a = 0.03$

Oxygen balance: $(1+Ex) \times 0.5 \times 2 = 0.97 + b \times 2$

Nitrogen balance: $(1+Ex) \times 0.5 \times 3.76 \times 2 = c \times 2$

Solving the above equations, we find the coefficients (Ex = 1, a = 0.03 b = 0.515, c = 3.76) and write the balanced reaction equation as

$$H_2 + [O_2 + 3.76N_2] \longrightarrow 0.97\,H_2O + 0.03\,H_2 + 0.515\,O_2 + 3.76\,N_2$$

Total moles of products at equilibrium are

$$N_{tot} = 0.97 + 0.03 + 0.515 + 3.76 = 5.275$$

The assumed equilibrium reaction is

$$H_2O \longleftrightarrow H_2 + 0.5O_2$$

The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{H2}\,\bar{g}^*_{H2}(T_{prod}) + \nu_{O2}\,\bar{g}^*_{O2}(T_{prod}) - \nu_{H2O}\,\bar{g}^*_{H2O}(T_{prod})$$

and the Gibbs functions are defined as

$$\bar{g}^*_{H2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{H2}$$
$$\bar{g}^*_{O2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{O2}$$
$$\bar{g}^*_{H2O}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{H2O}$$

The equilibrium constant is also given by

$$K_p = \left(\frac{P}{N_{tot}}\right)^{1+0.5-1} \frac{ab^{0.5}}{0.97^1} = \left(\frac{1}{5.275}\right)^{0.5} \frac{(0.03)(0.515)^{0.5}}{0.97} = 0.009664$$

and $\ln K_p = \ln(0.009664) = -4.647$

The corresponding temperature is obtained solving the above equations using EES to be

$$T_{prod} = \mathbf{2600\,K}$$

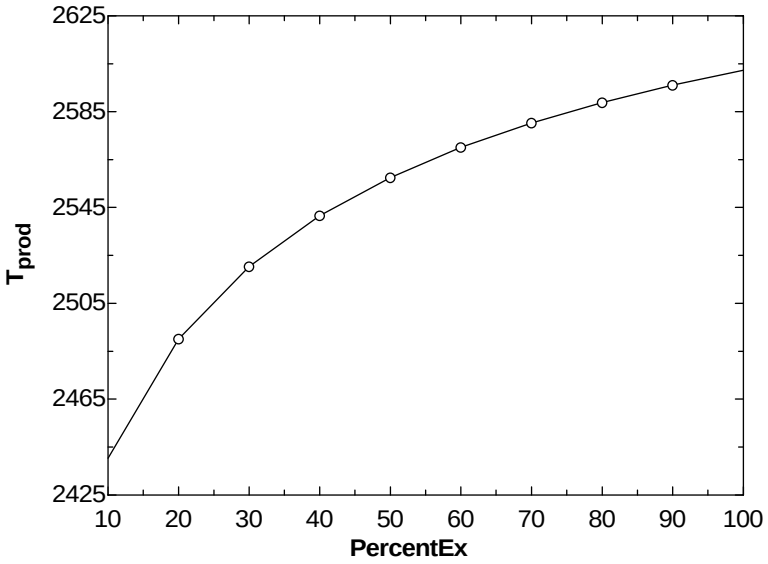
This is the temperature at which 97 percent of H_2 will burn into H_2O. The copy of EES solution is given next.

```
"Input Data from parametric table:"
{PercentEx = 10}
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3"[kPa]"
```

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R_u=8.314 "[kJ/kmol-K]"
"The combustion equation of H2 with stoichiometric amount of air is
H2 + 0.5(O2 + 3.76N2)=H2O +0.5(3.76)N2"
"For the incomplete combustion with 100% excess air, the combustion equation is
H2 + (1+EX)(0.5)(O2 + 3.76N2)=0.97 H2O +aH2 + bO2+cN2"
"Specie balance equations give the values of a, b, and c."
"H, hydrogen"
2 = 0.97*2 + a*2
"O, oxygen"
(1+Ex)*0.5*2=0.97 + b*2
"N, nitrogen"
(1+Ex)*0.5*3.76 *2 = c*2
N_tot =0.97+a +b +c "Total kilomoles of products at equilibrium"
"The assumed equilibrium reaction is
H2O=H2+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each H2mponent in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_H2O=Enthalpy(H2O,T=T_prod)-T_prod *Entropy(H2O,T=T_prod ,P=101.3)
g_H2=Enthalpy(H2,T=T_prod)-T_prod *Entropy(H2,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod)-T_prod *Entropy(O2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG =1*g_H2+0.5*g_O2-1*g_H2O
"The equilibrium constant is given by Eq. 15-14."
K_P = exp(-DELTAG /(R_u*T_prod))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(a^1*b^0.5)/(0.97^1)"
sqrt(P/N_tot)*a *sqrt(b)=K_P *0.97
lnK_p = ln(k_P)

ln K_p	PercentEx [%]	T_{prod} [K]
-5.414	10	2440
-5.165	20	2490
-5.019	30	2520
-4.918	40	2542
-4.844	50	2557
-4.786	60	2570
-4.739	70	2580
-4.7	80	2589
-4.667	90	2596
-4.639	100	2602



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16-24 The equilibrium constant of the reaction $CH_4 + 2O_2 \leftrightarrow CO_2 + 2H_2O$ at 25°C is to be determined.

Analysis The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{CO_2}\bar{g}^*_{CO_2}(T) + \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{CH_4}\bar{g}^*_{CH_4}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T)$$

$CH_4 + 2O_2 \leftrightarrow CO_2 + 2H_2O$
25°C

At 25°C,

$$\Delta G^*(T) = 1(-394{,}360) + 2(-228{,}590) - 1(-50{,}790) - 2(0) = -800{,}750 \text{ kJ/kmol}$$

Substituting,

$$\ln K_p = -(-800{,}750 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol} \cdot \text{K})(298 \text{ K})] = 323.04$$

or $$K_p = \mathbf{1.96 \times 10^{140}}$$

16-25 The equilibrium constant of the reaction $CO_2 \leftrightarrow CO + 1/2O_2$ is listed in Table A-28 at different temperatures. It is to be verified using Gibbs function data.

Analysis (*a*) The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where $$\Delta G^*(T) = \nu_{CO}\bar{g}^*_{CO}(T) + \nu_{O_2}\bar{g}^*_{O_2}(T) - \nu_{CO_2}\bar{g}^*_{CO_2}(T)$$

$CO_2 \leftrightarrow CO + ½O_2$
298 K

At 298 K,

$$\Delta G^*(T) = 1(-137{,}150) + 0.5(0) - 1(-394{,}360) = 257{,}210 \text{ kJ/kmol}$$

Substituting,

$$\ln K_p = -(257{,}210 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol} \cdot \text{K})(298 \text{ K})] = -103.81$$

or $$K_p = \mathbf{8.24 \times 10^{-46}} \quad (\text{Table A-28}: \ln K_p = -103.76)$$

(*b*) At 1800 K,

$$\begin{aligned}
\Delta G^*(T) &= \nu_{CO}\bar{g}^*_{CO}(T) + \nu_{O_2}\bar{g}^*_{O_2}(T) - \nu_{CO_2}\bar{g}^*_{CO_2}(T) \\
&= \nu_{CO}(\bar{h} - T\bar{s})_{CO} + \nu_{O_2}(\bar{h} - T\bar{s})_{O_2} - \nu_{CO_2}(\bar{h} - T\bar{s})_{CO_2} \\
&= \nu_{CO}[(\bar{h}_f + \bar{h}_{1800} - \bar{h}_{298}) - T\bar{s}]_{CO} \\
&\quad + \nu_{O_2}[(\bar{h}_f + \bar{h}_{1800} - \bar{h}_{298}) - T\bar{s}]_{O_2} \\
&\quad - \nu_{CO_2}[(\bar{h}_f + \bar{h}_{1800} - \bar{h}_{298}) - T\bar{s}]_{CO_2} \\
&= 1 \times (-110{,}530 + 58{,}191 - 8669 - 1800 \times 254.797) \\
&\quad + 0.5 \times (0 + 60{,}371 - 8682 - 1800 \times 264.701) \\
&\quad - 1 \times (-393{,}520 + 88{,}806 - 9364 - 1800 \times 302.884) \\
&= 127{,}240.2 \text{ kJ/kmol}
\end{aligned}$$

Substituting, $$\ln K_p = -(127{,}240.2 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol} \cdot \text{K})(1800 \text{ K})] = -8.502$$

or $$K_p = \mathbf{2.03 \times 10^{-4}} \quad (\text{Table A-28}: \ln K_p = -8.497)$$

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16-26 Carbon monoxide is burned with 100 percent excess air. The temperature at which 93 percent of CO burn to CO_2 is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O_2, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis Assuming N_2 to remain as an inert gas, the stoichiometric and actual reactions can be written as

Stoichiometric: $\mathrm{CO} + \frac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{CO_2}$ (thus $\nu_{\mathrm{CO_2}} = 1,\ \nu_{\mathrm{CO}} = 1,$ and $\nu_{\mathrm{O_2}} = \frac{1}{2}$)

Actual: $\mathrm{CO} + 1(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \underbrace{0.93\mathrm{CO_2}}_{\text{product}} + \underbrace{0.07\mathrm{CO} + 0.535\mathrm{O_2}}_{\text{reactants}} + \underbrace{3.76\mathrm{N_2}}_{\text{inert}}$

The equilibrium constant K_p can be determined from

$$K_p = \frac{N_{\mathrm{CO_2}}^{\nu_{\mathrm{CO_2}}}}{N_{\mathrm{CO}}^{\nu_{\mathrm{CO}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{CO_2}} - \nu_{\mathrm{CO}} - \nu_{\mathrm{O_2}})}$$

$$= \frac{0.93}{0.07 \times 0.535^{0.5}} \left(\frac{1}{0.93 + 0.07 + 0.535 + 3.76}\right)^{1-1.5}$$

$$= 41.80$$

$\mathrm{CO} + ½\mathrm{O_2} \leftrightarrow \mathrm{CO_2}$
93 %
1 atm

and

$$\ln K_p = 3.733$$

From Table A-28, the temperature corresponding to this K_p value is $T =$ **2424 K**

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16-27 Problem 16-26 is reconsidered. The effect of varying the percent excess air during the steady-flow process from 0 to 200 percent on the temperature at which 93 percent of CO burn into CO2 is to be studied.

Analysis The problem is solved using EES, and the solution is given below.

```
"To solve this problem, we need to give EES a guess value for T_prop other than
the default value of 1.  Set the guess value of T_prod to 1000 K by selecting  Variable
Information in the Options menu.  Then press F2 or click the Calculator icon."

"Input Data from the diagram window:"
{PercentEx = 100}
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3 [kPa]
R_u=8.314 [kJ/kmol-K]
f=0.93

"The combustion equation of CO with stoichiometric amount of air is
CO + 0.5(O2 + 3.76N2)=CO2 +0.5(3.76)N2"

"For the incomplete combustion with 100% excess air, the combustion equation is
CO + (1+EX)(0.5)(O2 + 3.76N2)=0.97 CO2 +aCO + bO2+cN2"

"Specie balance equations give the values of a, b, and c."
"C, Carbon"
1 = f + a
"O, oxygen"
1 +(1+Ex)*0.5*2=f*2 + a *1 + b*2
"N, nitrogen"
(1+Ex)*0.5*3.76 *2 = c*2

N_tot =f+a +b +c  "Total kilomoles of products at equilibrium"

"The assumed equilibrium reaction is CO2=CO+0.5O2"

"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"

g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)

"The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2

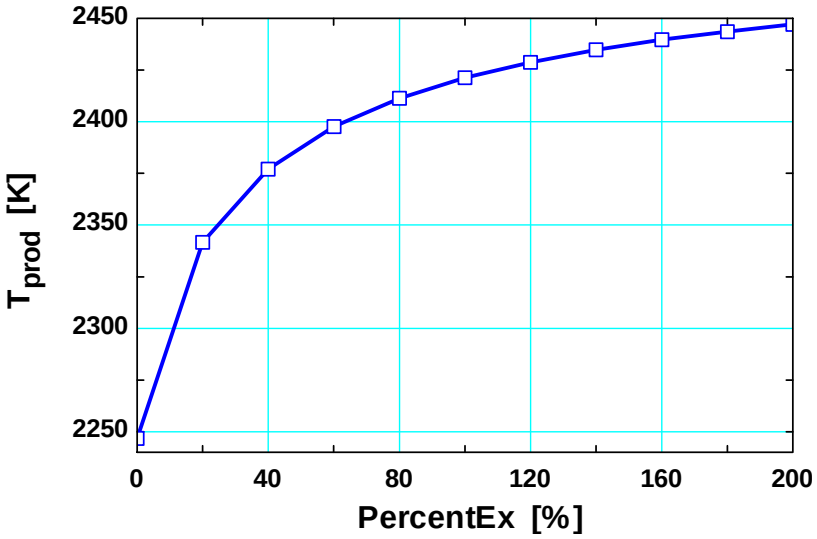
"The equilibrium constant is given by Eq. 15-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(a^1*b^0.5)/(0.97^1)"

sqrt(P/N_tot )*a *sqrt(b )=K_P *f
lnK_p = ln(k_P)

"Compare the value of lnK_p calculated by EES with the value of lnK_p from table A-28 in the text."
```

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PercentEx [%]	T_{prod} [K]
0	2247
20	2342
40	2377
60	2398
80	2411
100	2421
120	2429
140	2435
160	2440
180	2444
200	2447



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16-28E Carbon monoxide is burned with 100 percent excess air. The temperature at which 93 percent of CO burn to CO_2 is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O_2, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis Assuming N_2 to remain as an inert gas, the stoichiometric and actual reactions can be written as

Stoichiometric: $\mathrm{CO} + \frac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{CO_2}$ (thus $\nu_{\mathrm{CO_2}} = 1,\ \nu_{\mathrm{CO}} = 1,$ and $\nu_{\mathrm{O_2}} = \frac{1}{2}$)

Actual: $\mathrm{CO} + 1(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \underbrace{0.93\mathrm{CO_2}}_{\text{product}} + \underbrace{0.07\mathrm{CO} + 0.535\mathrm{O_2}}_{\text{reactants}} + \underbrace{3.76\mathrm{N_2}}_{\text{inert}}$

The equilibrium constant K_p can be determined from

$$K_p = \frac{N_{\mathrm{CO_2}}^{\nu_{\mathrm{CO_2}}}}{N_{\mathrm{CO}}^{\nu_{\mathrm{CO}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{CO_2}} - \nu_{\mathrm{CO}} - \nu_{\mathrm{O_2}})}$$

$$= \frac{0.93}{0.07 \times 0.535^{0.5}} \left(\frac{1}{0.93 + 0.07 + 0.535 + 3.76}\right)^{1-1.5}$$

$$= 41.80$$

$\mathrm{CO} + \frac{1}{2}\mathrm{O_2} \leftrightarrow \mathrm{CO_2}$
93 %
1 atm

and

$$\ln K_p = 3.733$$

From Table A-28, the temperature corresponding to this K_p value is T = **2424 K = 4363 R**

16-29 Hydrogen is burned with 150 percent theoretical air. The temperature at which 98 percent of H_2 will burn to H_2O is to be determined.

Assumptions **1** The equilibrium composition consists of H_2O, H_2, O_2, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis Assuming N_2 to remain as an inert gas, the stoichiometric and actual reactions can be written as

Stoichiometric: $\mathrm{H_2} + \frac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{H_2O}$ (thus $\nu_{\mathrm{H_2O}} = 1, \nu_{\mathrm{H_2}} = 1,$ and $\nu_{\mathrm{O_2}} = \frac{1}{2}$)

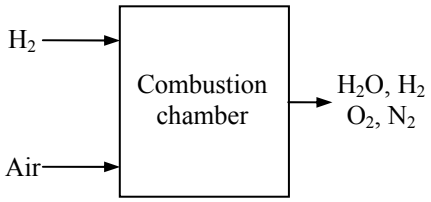
Actual: $\mathrm{H_2} + 0.75(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \underbrace{0.98\mathrm{H_2O}}_{\text{product}} + \underbrace{0.02\mathrm{H_2} + 0.26\mathrm{O_2}}_{\text{reactants}} + \underbrace{2.82\mathrm{N_2}}_{\text{inert}}$

The equilibrium constant K_p can be determined from

$$K_p = \frac{N_{\mathrm{H_2O}}^{\nu_{\mathrm{H_2O}}}}{N_{\mathrm{H_2}}^{\nu_{\mathrm{H_2}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{H_2O}} - \nu_{\mathrm{H_2}} - \nu_{\mathrm{O_2}})}$$

$$= \frac{0.98}{0.02 \times 0.26^{0.5}} \left(\frac{1}{0.98 + 0.02 + 0.26 + 2.82}\right)^{1-1.5}$$

$$= 194.11$$



From Table A-28, the temperature corresponding to this K_p value is T = **2472 K**.

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16-30 Air is heated to a high temperature. The equilibrium composition at that temperature is to be determined.

Assumptions **1** The equilibrium composition consists of N_2, O_2, and NO. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

Stoichiometric: $\frac{1}{2}N_2 + \frac{1}{2}O_2 \Leftrightarrow NO$ (thus $\nu_{NO} = 1$, $\nu_{N_2} = \frac{1}{2}$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $3.76\,N_2 + O_2 \longrightarrow \underbrace{x\,NO}_{\text{prod.}} + \underbrace{y\,N_2 + z\,O_2}_{\text{reactants}}$

AIR
2000 K
2 atm

N balance: $7.52 = x + 2y$ or $y = 3.76 - 0.5x$

O balance: $2 = x + 2z$ or $z = 1 - 0.5x$

Total number of moles: $N_{\text{total}} = x + y + z = x + 4.76 - x = 4.76$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N_2}^{\nu_{N_2}} N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{NO} - \nu_{N_2} - \nu_{O_2})}$$

From Table A-28, ln K_p = -3.931 at 2000 K. Thus K_p = 0.01962. Substituting,

$$0.01962 = \frac{x}{(3.76 - 0.5x)^{0.5}(1 - 0.5x)^{0.5}} \left(\frac{2}{4.76}\right)^{1-1}$$

Solving for x,

$x = 0.0376$

Then,

$y = 3.76\text{-}0.5x = 3.7412$

$z = 1\text{-}0.5x = 0.9812$

Therefore, the equilibrium composition of the mixture at 2000 K and 2 atm is

$$0.0376NO + 3.7412N_2 + 0.9812O_2$$

The equilibrium constant for the reactions $O_2 \Leftrightarrow 2O$ (ln K_p = -14.622) and $N_2 \Leftrightarrow 2N$ (ln K_p = -41.645) are much smaller than that of the specified reaction (ln K_p = -3.931). Therefore, it is realistic to assume that no monatomic oxygen or nitrogen will be present in the equilibrium mixture. Also the equilibrium composition is in this case is independent of pressure since $\Delta\nu = 1 - 0.5 - 0.5 = 0$.

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16-31 Hydrogen is heated to a high temperature at a constant pressure. The percentage of H_2 that will dissociate into H is to be determined.

Assumptions **1** The equilibrium composition consists of H_2 and H. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions can be written as

Stoichiometric: $H_2 \Leftrightarrow 2H$ (thus $\nu_{H_2} = 1$ and $\nu_H = 2$)

Actual: $H_2 \longrightarrow \underbrace{x H_2}_{\text{react.}} + \underbrace{y H}_{\text{prod.}}$

H_2
4000 K
5 atm

H balance: $2 = 2x + y$ or $y = 2 - 2x$

Total number of moles: $N_{\text{total}} = x + y = x + 2 - 2x = 2 - x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_H^{\nu_H}}{N_{H_2}^{\nu_{H_2}}} \left(\frac{P}{N_{\text{total}}} \right)^{\nu_H - \nu_{H_2}}$$

From Table A-28, ln K_p = 0.934 at 4000 K. Thus K_p = 2.545. Substituting,

$$2.545 = \frac{(2 - 2x)^2}{x} \left(\frac{5}{2 - x} \right)^{2-1}$$

Solving for x,

$$x = 0.664$$

Thus the percentage of H_2 which dissociates to H at 3200 K and 8 atm is

$$1 - 0.664 = 0.336 \quad \text{or} \quad \mathbf{33.6\%}$$

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16-32E A mixture of CO, O_2, and N_2 is heated to a high temperature at a constant pressure. The equilibrium composition is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O_2, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

2 CO
2 O_2
6 N_2
4320 R
3 atm

Stoichiometric: $\mathrm{CO} + \frac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{CO_2}$ (thus $\nu_{\mathrm{CO_2}} = 1, \nu_{\mathrm{CO}} = 1,$ and $\nu_{\mathrm{O_2}} = \frac{1}{2}$)

Actual: $2\,\mathrm{CO} + 2\,\mathrm{O_2} + 6\,\mathrm{N_2} \longrightarrow \underbrace{x\,\mathrm{CO_2}}_{\text{products}} + \underbrace{y\,\mathrm{CO} + z\,\mathrm{O_2}}_{\text{reactants}} + \underbrace{6\,\mathrm{N_2}}_{\text{inert}}$

C balance: $2 = x + y \longrightarrow y = 2 - x$

O balance: $6 = 2x + y + 2z \longrightarrow z = 2 - 0.5x$

Total number of moles: $N_{\text{total}} = x + y + z + 6 = 10 - 0.5x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{\mathrm{CO_2}}^{\nu_{\mathrm{CO_2}}}}{N_{\mathrm{CO}}^{\nu_{\mathrm{CO}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{CO_2}} - \nu_{\mathrm{CO}} - \nu_{\mathrm{O_2}})}$$

From Table A-28, $\ln K_p = 3.860$ at $T = 4320$ $R = 2400\,\mathrm{K}$. Thus $K_p = 47.465$. Substituting,

$$47.465 = \frac{x}{(2-x)(2-0.5x)^{0.5}} \left(\frac{3}{10-0.5x}\right)^{1-1.5}$$

Solving for x,

$x = 1.930$

Then,

$y = 2 - x = 0.070$

$z = 2 - 0.5x = 1.035$

Therefore, the equilibrium composition of the mixture at 2400 K and 3 atm is

$$\mathbf{1.930CO_2 + 0.070CO + 1.035O_2 + 6N_2}$$

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16-33 A mixture of N_2, O_2, and Ar is heated to a high temperature at a constant pressure. The equilibrium composition is to be determined.

Assumptions **1** The equilibrium composition consists of N_2, O_2, Ar, and NO. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

3 N_2
1 O_2
0.1 Ar
2400 K
10 atm

Stoichiometric: $\frac{1}{2}\,N_2 + \frac{1}{2}\,O_2 \Leftrightarrow NO$ (thus $\nu_{NO} = 1$, $\nu_{N_2} = \frac{1}{2}$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $3\,N_2 + O_2 + 0.1\,Ar \longrightarrow \underbrace{x\,NO}_{\text{prod.}} + \underbrace{y\,N_2 + z\,O_2}_{\text{reactants}} + \underbrace{0.1\,Ar}_{\text{inert}}$

N balance: $6 = x + 2y \longrightarrow y = 3 - 0.5x$

O balance: $2 = x + 2z \longrightarrow z = 1 - 0.5x$

Total number of moles: $N_{\text{total}} = x + y + z + 0.1 = 4.1$

The equilibrium constant relation becomes,

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N_2}^{\nu_{N_2}} N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{NO}-\nu_{N_2}-\nu_{O_2})} = \frac{x}{y^{0.5} z^{0.5}} \left(\frac{P}{N_{\text{total}}}\right)^{1-0.5-0.5}$$

From Table A-28, $\ln K_p = -3.019$ at 2400 K. Thus $K_p = 0.04885$. Substituting,

$$0.04885 = \frac{x}{(3-0.5x)^{0.5}(1-0.5x)^{0.5}} \times 1$$

Solving for x,

$$x = 0.0823$$

Then,

$$y = 3 - 0.5x = 2.9589$$

$$z = 1 - 0.5x = 0.9589$$

Therefore, the equilibrium composition of the mixture at 2400 K and 10 atm is

$$\mathbf{0.0823NO + 2.9589N_2 + 0.9589O_2 + 0.1Ar}$$

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16-34 The mole fraction of sodium that ionizes according to the reaction $Na \Leftrightarrow Na^+ + e^-$ at 2000 K and 1.5 atm is to be determined.

Assumptions All components behave as ideal gases.

Analysis The stoichiometric and actual reactions can be written as

$Na \Leftrightarrow Na^+ + e^-$
2000 K
1.5 atm

Stoichiometric: $Na \Leftrightarrow Na^+ + e^-$ (thus $\nu_{Na} = 1,\ \nu_{Na^+} = 1$ and $\nu_{e^-} = 1$)

Actual: $Na \longrightarrow \underbrace{x\,Na}_{\text{react.}} + \underbrace{y\,Na^+ + y\,e^-}_{\text{products}}$

Na balance: $1 = x + y \quad or \quad y = 1 - x$

Total number of moles: $N_{total} = x + 2y = 2 - x$

The equilibrium constant relation becomes,

$$K_p = \frac{N_{Na}^{\nu_{Na}} N_{e^-}^{\nu_{e^-}}}{N_{Na}^{\nu_{Na}}} \left(\frac{P}{N_{total}}\right)^{(\nu_{Na^+} + \nu_{e^-} - \nu_{Na})} = \frac{y^2}{x}\left(\frac{P}{N_{total}}\right)^{1+1-1}$$

Substituting,

$$0.668 = \frac{(1-x)^2}{x}\left(\frac{1.5}{2-x}\right)$$

Solving for x,

$$x = 0.4449$$

Thus the fraction of Na which dissociates into Na^+ and e^- is

$$1 - 0.4449 = 0.555 \quad \text{or} \quad \mathbf{55.5\%}$$

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16-35 Oxygen is heated from a specified state to another state. The amount of heat required is to be determined without and with dissociation cases.

Assumptions **1** The equilibrium composition consists of O_2 and O. **2** The constituents of the mixture are ideal gases.

Analysis (*a*) Obtaining oxygen properties from table A-19, an energy balance gives

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$\begin{aligned} q_{\text{in}} &= \bar{u}_2 - \bar{u}_1 \\ &= 57{,}192 - 6203 \\ &= \mathbf{50{,}989\ kJ/kmol} \end{aligned}$$

O_2
2200 K
1 atm

(*b*) The stoichiometric and actual reactions in this case are

Stoichiometric: $O_2 \Leftrightarrow 2O$ (thus $\nu_{O2} = 1$ and $\nu_O = 2$)

Actual: $O_2 \longrightarrow \underbrace{xO_2}_{\text{react.}} + \underbrace{yO}_{\text{products}}$

O balance: $2 = 2x + y \longrightarrow y = 2 - 2x$

Total number of moles: $N_{\text{total}} = x + y = 2 - x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_O^{\nu_O}}{N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}} \right)^{\nu_O - \nu_{O2}}$$

From Table A-28, at 2200 K, $\ln K_p = -11.827$. Then,

$$K_p = \exp(-11.827) = 7.305 \times 10^{-6}$$

Substituting,

$$7.305 \times 10^{-6} = \frac{(2-2x)^2}{x} \left(\frac{1}{2-x} \right)^{2-1}$$

Solving for x,

$$x = 0.99865$$

Then,

$$y = 2 - 2x = 0.0027$$

Therefore, the equilibrium composition of the mixture at 2200 K and 1 atm is

$$0.99865\,O_2 + 0.0027\,O$$

Hence, the oxygen ions are negligible and the result is same as that in part (a),

$$q_{\text{in}} = \mathbf{50{,}989\ kJ/kmol}$$

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16-36 Air is heated from a specified state to another state. The amount of heat required is to be determined without and with dissociation cases.

Assumptions **1** The equilibrium composition consists of O_2 and O, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis (*a*) Obtaining air properties from table A-17, an energy balance gives

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$\begin{aligned} q_{\text{in}} &= u_2 - u_1 \\ &= 1872.4 - 212.64 \\ &= \mathbf{1660\ kJ/kg} \end{aligned}$$

O_2, $3.76N_2$
2200 K
1 atm

(*b*) The stoichiometric and actual reactions in this case are

Stoichiometric: $O_2 \Leftrightarrow 2O$ (thus $\nu_{O2} = 1$ and $\nu_O = 2$)

Actual: $$O_2 + 3.76N_2 \longrightarrow \underbrace{xO_2}_{\text{react.}} + \underbrace{yO}_{\text{products}} + \underbrace{3.76N_2}_{\text{inert}}$$

O balance: $2 = 2x + y \longrightarrow y = 2 - 2x$

Total number of moles: $N_{\text{total}} = x + y + 3.76 = 5.76 - x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_O^{\nu_O}}{N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}} \right)^{\nu_O - \nu_{O2}}$$

From Table A-28, at 2200 K, $\ln K_p = -11.827$. Then,

$$K_p = \exp(-11.827) = 7.305 \times 10^{-6}$$

Substituting,

$$7.305 \times 10^{-6} = \frac{(2-2x)^2}{x} \left(\frac{1}{5.76 - x} \right)^{2-1}$$

Solving for x,

$$x = 0.99706$$

Then,

$$y = 2 - 2x = 0.00588$$

Therefore, the equilibrium composition of the mixture at 2200 K and 1 atm is

$$0.99706\,O_2 + 0.00588\,O + 3.76\,N_2$$

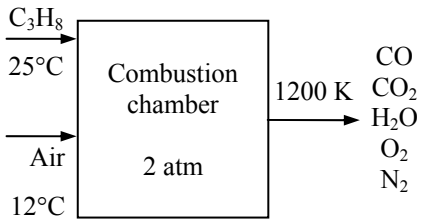
Hence, the atomic oxygen is negligible and the result is same as that in part (a),

$$q_{\text{in}} = \mathbf{1660\ kJ/kg}$$

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16-37 Liquid propane enters a combustion chamber. The equilibrium composition of product gases and the rate of heat transfer from the combustion chamber are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, H_2O, CO, N_2, and O_2. **2** The constituents of the mixture are ideal gases.



Analysis (*a*) Considering 1 kmol of C_3H_8, the stoichiometric combustion equation can be written as

$$C_3H_8(\ell) + a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 3.76a_{th}N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$2.5a_{th} = 3 + 2 + 1.5a_{th} \quad \longrightarrow \quad a_{th} = 5$$

Then the actual combustion equation with 150% excess air and some CO in the products can be written as

$$C_3H_8(\ell) + 12.5(O_2 + 3.76N_2) \quad \longrightarrow \quad xCO_2 + (3-x)CO + (9-0.5x)O_2 + 4H_2O + 47N_2$$

After combustion, there will be no C_3H_8 present in the combustion chamber, and H_2O will act like an inert gas. The equilibrium equation among CO_2, CO, and O_2 can be expressed as

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2}O_2 \quad (\text{thus } \nu_{CO_2} = 1,\ \nu_{CO} = 1,\ \text{and } \nu_{O_2} = \tfrac{1}{2})$$

and

$$K_p = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{CO}+\nu_{O_2}-\nu_{CO_2})}$$

where

$$N_{\text{total}} = x + (3-x) + (9-0.5x) + 4 + 47 = 63 - 0.5x$$

From Table A-28, $\ln K_p = -17.871$ at 1200 K. Thus $K_p = 1.73 \times 10^{-8}$. Substituting,

$$1.73 \times 10^{-8} = \frac{(3-x)(9-0.5x)^{0.5}}{x} \left(\frac{2}{63-0.5x}\right)^{1.5-1}$$

Solving for x,

$$x = 2.9999999 \cong 3.0$$

Therefore, the amount CO in the product gases is negligible, and it can be disregarded with no loss in accuracy. Then the combustion equation and the equilibrium composition can be expressed as

$$C_3H_8(\ell) + 12.5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 7.5O_2 + 4H_2O + 47N_2$$

and

$$\mathbf{3CO_2 + 7.5O_2 + 4H_2O + 47N_2}$$

(*b*) The heat transfer for this combustion process is determined from the steady-flow energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ on the combustion chamber with $W = 0$,

$$-Q_{\text{out}} = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables, (The $\overline{h}_f^\circ$ of liquid propane is obtained by adding the h_{fg} at 25°C to $\overline{h}_f^\circ$ of gaseous propane).

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Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{285\ K}$ kJ/kmol	$\bar{h}_{298\ K}$ kJ/kmol	$\bar{h}_{1200\ K}$ kJ/kmol
C_3H_8 (ℓ)	-118,910	---	---	---
O_2	0	8696.5	8682	38,447
N_2	0	8286.5	8669	36,777
H_2O (g)	-241,820	---	9904	44,380
CO_2	-393,520	---	9364	53,848

Substituting,

$$\begin{aligned}-Q_{\text{out}} &= 3(-393{,}520 + 53{,}848 - 9364) + 4(-241{,}820 + 44{,}380 - 9904) \\ &\quad + 7.5(0 + 38{,}447 - 8682) + 47(0 + 36{,}777 - 8669) \\ &\quad - 1(-118{,}910 + h_{298} - h_{298}) - 12.5(0 + 8296.5 - 8682) \\ &\quad - 47(0 + 8186.5 - 8669) \\ &= -185{,}764 \text{ kJ / kmol of } C_3H_8\end{aligned}$$

or

$$Q_{\text{out}} = 185{,}764 \text{ kJ / kmol of } C_3H_8$$

The mass flow rate of C_3H_8 can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{1.2 \text{ kg / min}}{44 \text{ kg / kmol}} = 0.02727 \text{ kmol / min}$$

Thus the rate of heat transfer is

$$\dot{Q}_{\text{out}} = \dot{N} \times Q_{\text{out}} = (0.02727 \text{ kmol/min})(185{,}746 \text{ kJ/kmol}) = \mathbf{5066\ kJ/min}$$

The equilibrium constant for the reaction $\frac{1}{2}N_2 + \frac{1}{2}O_2 \Leftrightarrow NO$ is $\ln K_p = -7.569$, which is very small. This indicates that the amount of NO formed during this process will be very small, and can be disregarded.

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16-38 Problem 16-37 is reconsidered. It is to be investigated if it is realistic to disregard the presence of NO in the product gases.

Analysis The problem is solved using EES, and the solution is given below.

```
"To solve this problem,  the Gibbs function of the product gases is minimized. Click on the Min/Max icon."

For this problem at 1200 K the moles of  CO are 0.000 and moles of NO are 0.000, thus we
can disregard both the CO and NO.  However, try some product temperatures above 1286 K
and observe the sign change on the Q_out and the amout of CO and NO present as the
product temperature increases."

"The reaction of C3H8(liq) with excess air can be written:

    C3H8(l) + (1+Ex)A_th (O2+3.76N2) = a C02 + b CO + c H2O + d N2 + e O2 + f NO

The coefficients A_th and EX are the theoretical oxygen and the percent excess air on a decimal
basis.   Coefficients a, b, c, d, e, and f are found by minimiming the Gibbs Free Energy at a total
pressure of the product gases P_Prod and the product  temperature T_Prod.

The equilibrium solution can be found by applying the Law of Mass Action or by minimizing the Gibbs function.
In this problem, the Gibbs function is directly minimized using the optimization capabilities built into EES.

To run this program, click on the Min/Max icon.  There are six compounds present  in the products subject to four
specie balances, so there are two degrees of freedom.   Minimize the Gibbs function of the product gases with
respect to two molar quantities such as coefficients b and f.  The equilibrium mole numbers a, b, c, d, e, and f
will be determined and displayed in the Solution window."

PercentEx = 150 [%]
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =2*P_atm
T_Prod=1200 [K]
m_dot_fuel = 0.5 [kg/s]
Fuel$='C3H8'
T_air = 12+273 "[K]"
T_fuel = 25+273 "[K]"
P_atm = 101.325 [kPa]
R_u=8.314 [kJ/kmol-K]

"Theoretical combustion of C3H8 with oxygen:
C3H8 + A_th O2 = 3 C02 + 4 H2O "
2*A_th = 3*2 + 4*1

"Balance the reaction for 1 kmol of C3H8"
"C3H8(l) + (1+Ex)A_th (O2+3.76N2) = a C02 + b CO + c H2O + d N2 + e O2 + f NO"
b_max = 3
f_max = (1+Ex)*A_th*3.76*2
e_guess=Ex*A_th
1*3 = a*1+b*1 "Carbon balance"
1*8=c*2
"Hydrogen balance"
(1+Ex)*A_th*2=a*2+b*1+c*1+e*2+f*1 "Oxygen balance"
(1+Ex)*A_th*3.76*2=d*2+f*1 "Nitrogen balance"
```

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```
"Total moles and mole fractions"
N_Total=a+b+c+d+e+f
y_CO2=a/N_Total;  y_CO=b/N_Total; y_H2O=c/N_Total; y_N2=d/N_Total; y_O2=e/N_Total; y_NO=f/N_Total

"The following equations provide the specific Gibbs function for each component as a function of its molar amount"
g_CO2=Enthalpy(CO2,T=T_Prod)-T_Prod*Entropy(CO2,T=T_Prod,P=P_Prod*y_CO2)
g_CO=Enthalpy(CO,T=T_Prod)-T_Prod*Entropy(CO,T=T_Prod,P=P_Prod*y_CO)
g_H2O=Enthalpy(H2O,T=T_Prod)-T_Prod*Entropy(H2O,T=T_Prod,P=P_Prod*y_H2O)
g_N2=Enthalpy(N2,T=T_Prod)-T_Prod*Entropy(N2,T=T_Prod,P=P_Prod*y_N2)
g_O2=Enthalpy(O2,T=T_Prod)-T_Prod*Entropy(O2,T=T_Prod,P=P_Prod*y_O2)
g_NO=Enthalpy(NO,T=T_Prod)-T_Prod*Entropy(NO,T=T_Prod,P=P_Prod*y_NO)

"The extensive Gibbs function is the sum of the products of the specific Gibbs function and the molar amount of each substance"
Gibbs=a*g_CO2+b*g_CO+c*g_H2O+d*g_N2+e*g_O2+f*g_NO

"For the energy balance, we adjust the value of the enthalpy of gaseous propane given by EES:"
h_fg_fuel = 15060"[kJ/kmol]" "Table A.27"
h_fuel = enthalpy(Fuel$,T=T_fuel)-h_fg_fuel
"Energy balance for the combustion process:"
"C3H8(l) + (1+Ex)A_th (O2+3.76N2) = a C02 + b CO + c H2O + d N2 + e O2 + f NO"
HR =Q_out+HP
HR=h_fuel+ (1+Ex)*A_th*(enthalpy(O2,T=T_air)+3.76*enthalpy(N2,T=T_air))
HP=a*enthalpy(CO2,T=T_prod)+b*enthalpy(CO,T=T_prod)+c*enthalpy(H2O,T=T_prod)+d*enthalpy(N2,T=T_prod)+e*enthalpy(O2,T=T_prod)+f*enthalpy(NO,T=T_prod)
"The heat transfer rate is:"
Q_dot_out=Q_out/molarmass(Fuel$)*m_dot_fuel "[kW]"
```

SOLUTION

```
a=3.000 [kmol]
A_th=5
b=0.000 [kmol]
b_max=3
c=4.000 [kmol]
d=47.000 [kmol]
e=7.500 [kmol]
Ex=1.5
e_guess=7.5
f=0.000 [kmol]
Fuel$='C3H8'
f_max=94
Gibbs=-17994897 [kJ]
g_CO=-703496 [kJ/kmol]
g_CO2=-707231 [kJ/kmol]
g_H2O=-515974 [kJ/kmol]
g_N2=-248486 [kJ/kmol]
g_NO=-342270 [kJ/kmol]
g_O2=-284065 [kJ/kmol]
HP=-330516.747 [kJ/kmol]
HR=-141784.529 [kJ/kmol]
h_fg_fuel=15060 [kJ/kmol]
h_fuel=-118918 [kJ/kmol]
m_dot_fuel=0.5 [kg/s]
N_Total=61.5 [kmol/kmol_fuel]
PercentEx=150 [%]
P_atm=101.3 [kPa]
P_prod=202.7 [kPa]
Q_dot_out=2140 [kW]
Q_out=188732 [kJ/kmol_fuel]
R_u=8.314 [kJ/kmol-K]
T_air=285 [K]
T_fuel=298 [K]
T_Prod=1200.00 [K]
y_CO=1.626E-15
y_CO2=0.04878
y_H2O=0.06504
y_N2=0.7642
y_NO=7.857E-08
y_O2=0.122
```

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16-39 Oxygen is heated during a steady-flow process. The rate of heat supply needed during this process is to be determined for two cases.

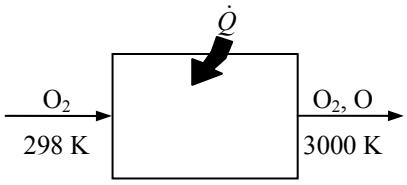
Assumptions **1** The equilibrium composition consists of O_2 and O. **2** All components behave as ideal gases.

Analysis (*a*) Assuming some O_2 dissociates into O, the dissociation equation can be written as

$$O_2 \longrightarrow x\,O_2 + 2(1-x)O$$

The equilibrium equation among O_2 and O can be expressed as

$$O_2 \Leftrightarrow 2O \quad (\text{thus } \nu_{O_2} = 1 \text{ and } \nu_O = 2)$$



Assuming ideal gas behavior for all components, the equilibrium constant relation can be expressed as

$$K_p = \frac{N_O^{\nu_O}}{N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{\text{total}}}\right)^{\nu_O - \nu_{O_2}}$$

where $N_{\text{total}} = x + 2(1-x) = 2 - x$

From Table A-28, $\ln K_p = -4.357$ at 3000 K. Thus $K_p = 0.01282$. Substituting,

$$0.01282 = \frac{(2-2x)^2}{x}\left(\frac{1}{2-x}\right)^{2-1}$$

Solving for x gives

$$x = 0.943$$

Then the dissociation equation becomes

$$O_2 \longrightarrow 0.943\,O_2 + 0.114\,O$$

The heat transfer for this combustion process is determined from the steady-flow energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ on the combustion chamber with $W = 0$,

$$Q_{\text{in}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the O_2 and O to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\text{ K}}$ kJ/kmol	$\bar{h}_{3000\text{ K}}$ kJ/kmol
O	249,190	6852	63,425
O_2	0	8682	106,780

Substituting,

$$Q_{\text{in}} = 0.943(0 + 106{,}780 - 8682) + 0.114(249{,}190 + 63{,}425 - 6852) - 0 = 127{,}363 \text{ kJ / kmol } O_2$$

The mass flow rate of O_2 can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{0.5 \text{ kg/min}}{32 \text{ kg/kmol}} = 0.01563 \text{ kmol/min}$$

Thus the rate of heat transfer is

$$\dot{Q}_{\text{in}} = \dot{N} \times Q_{\text{in}} = (0.01563 \text{ kmol/min})(127{,}363 \text{ kJ/kmol}) = \mathbf{1990\ kJ/min}$$

(*b*) If no O_2 dissociates into O, then the process involves no chemical reactions and the heat transfer can be determined from the steady-flow energy balance for nonreacting systems to be

$$\dot{Q}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{N}(\bar{h}_2 - \bar{h}_1) = (0.01563 \text{ kmol/min})(106{,}780 - 8682) \text{ kJ/kmol} = \mathbf{1533\ kJ/min}$$

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16-40 The equilibrium constant, K_p is to be estimated at 3000 K for the reaction CO + H2O = CO2 + H2.

Analysis (*a*) The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{CO2}\bar{g}^*_{CO2}(T) + \nu_{H2}\bar{g}^*_{H2}(T) - \nu_{CO}\bar{g}^*_{CO}(T) - \nu_{H2O}\bar{g}^*_{H2O}(T)$$

At 3000 K,

$$\begin{aligned}\Delta G^*(T) &= \nu_{CO2}\bar{g}^*_{CO2}(T) + \nu_{H2}\bar{g}^*_{H2}(T) - \nu_{CO}\bar{g}^*_{CO}(T) - \nu_{H2O}\bar{g}^*_{H2O}(T)\\ &= \nu_{CO2}(\bar{h} - T\bar{s})_{CO2} + \nu_{H2}(\bar{h} - T\bar{s})_{H2} - \nu_{CO}(\bar{h} - T\bar{s})_{CO} - \nu_{H2O}(\bar{h} - T\bar{s})_{H2O}\\ &= 1[(-393{,}520 + 162{,}226 - 9364) - (3000)(334.084)]\\ &\quad + 1[(0 + 97{,}211 - 8468) - (3000)(202.778)]\\ &\quad - 1[(-110{,}530 + 102{,}210 - 8669) - (3000)(273.508]\\ &\quad - 1[(-241{,}820 + 136{,}264 - 9904) - (3000)(286.273]\\ &= 49{,}291\ \text{kJ/kmol}\end{aligned}$$

Substituting,

$$\ln K_p = -\frac{49{,}291\ \text{kJ/kmol}}{(8.314\ \text{kJ/kmol}\cdot\text{K})(3000\ \text{K})} = -1.9762 \longrightarrow K_p = \mathbf{0.1386}$$

The equilibrium constant may be estimated using the integrated van't Hoff equation:

$$\ln\left(\frac{K_{p,\text{est}}}{K_{p1}}\right) = \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_R} - \frac{1}{T}\right)$$

$$\ln\left(\frac{K_{p,\text{est}}}{0.2209}\right) = \frac{-26{,}176\ \text{kJ/kmol}}{8.314\ \text{kJ/kmol.K}}\left(\frac{1}{2000\ \text{K}} - \frac{1}{3000\ \text{K}}\right) \longrightarrow K_{p,\text{est}} = \mathbf{0.1307}$$

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16-41 A constant volume tank contains a mixture of H_2 and O_2. The contents are ignited. The final temperature and pressure in the tank are to be determined.

Analysis The reaction equation with products in equilibrium is

$$H_2 + O_2 \longrightarrow a\,H_2 + b\,H_2O + c\,O_2$$

The coefficients are determined from the mass balances

Hydrogen balance: $2 = 2a + 2b$

Oxygen balance: $2 = b + 2c$

The assumed equilibrium reaction is

$$H_2O \longleftrightarrow H_2 + 0.5O_2$$

The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{H2}\,\overline{g}^*_{H2}(T_{prod}) + \nu_{O2}\,\overline{g}^*_{O2}(T_{prod}) - \nu_{H2O}\,\overline{g}^*_{H2O}(T_{prod})$$

and the Gibbs functions are given by

$$\overline{g}^*_{H2}(T_{prod}) = (\overline{h} - T_{prod}\overline{s})_{H2}$$
$$\overline{g}^*_{O2}(T_{prod}) = (\overline{h} - T_{prod}\overline{s})_{O2}$$
$$\overline{g}^*_{H2O}(T_{prod}) = (\overline{h} - T_{prod}\overline{s})_{H2O}$$

The equilibrium constant is also given by

$$K_p = \frac{a^1 c^{0.5}}{b^1}\left(\frac{P}{N_{tot}}\right)^{1+0.5-1} = \frac{ac^{0.5}}{b}\left(\frac{P_2/101.3}{a+b+c}\right)^{0.5}$$

An energy balance on the tank under adiabatic conditions gives

$$U_R = U_P$$

where

$$U_R = 1(\overline{h}_{H2@25°C} - R_u T_{reac}) + 1(\overline{h}_{O2@25°C} - R_u T_{reac})$$
$$= 0 - (8.314\ \text{kJ/kmol.K})(298.15\ \text{K}) + 0 - (8.314\ \text{kJ/kmol.K})(298.15\ \text{K}) = -4958\ \text{kJ/kmol}$$
$$U_P = a(\overline{h}_{H2@T_{prod}} - R_u T_{prod}) + b(\overline{h}_{H2O@T_{prod}} - R_u T_{prod}) + c(\overline{h}_{O2@T_{prod}} - R_u T_{prod})$$

The relation for the final pressure is

$$P_2 = \frac{N_{tot}}{N_1}\frac{T_{prod}}{T_{reac}}P_1 = \left(\frac{a+b+c}{2}\right)\left(\frac{T_{prod}}{298.15\,\text{K}}\right)(101.3\,\text{kPa})$$

Solving all the equations simultaneously using EES, we obtain the final temperature and pressure in the tank to be

$$T_{prod} = \mathbf{3857\ K}$$
$$P_2 = \mathbf{1043\ kPa}$$

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16-42 It is to be shown that as long as the extent of the reaction, α, for the disassociation reaction $X_2 \Leftrightarrow 2X$ is smaller than one, α is given by $\alpha = \sqrt{\dfrac{K_P}{4+K_P}}$

Assumptions The reaction occurs at the reference temperature.

Analysis The stoichiometric and actual reactions can be written as

Stoichiometric: $X_2 \Leftrightarrow 2X$ (thus $\nu_{X2} = 1$ and $\nu_X = 2$)

Actual: $X_2 \Leftrightarrow \underbrace{(1-\alpha)X_2}_{\text{react.}} + \underbrace{2\alpha X}_{\text{prod.}}$

The equilibrium constant K_p is given by

$$K_p = \frac{N_X^{\nu_X}}{N_{X2}^{\nu_{X2}}}\left(\frac{P}{N_{\text{total}}}\right)^{\nu_X-\nu_{X2}} = \frac{(2\alpha)^2}{(1-\alpha)}\left(\frac{1}{\alpha+1}\right)^{2-1} = \frac{4\alpha^2}{(1-\alpha)(1+\alpha)}$$

Solving this expression for α gives

$$\alpha = \sqrt{\frac{K_P}{4+K_P}}$$

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Simultaneous Reactions

16-43C It can be expresses as "$(dG)_{T,P} = 0$ for each reaction." Or as "the K_p relation for each reaction must be satisfied."

16-44C The number of K_p relations needed to determine the equilibrium composition of a reacting mixture is equal to the difference between the number of species present in the equilibrium mixture and the number of elements.

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16-45 Two chemical reactions are occurring in a mixture. The equilibrium composition at a specified temperature is to be determined.

Assumptions **1** The equilibrium composition consists of H_2O, OH, O_2, and H_2.
2 The constituents of the mixture are ideal gases.

$$H_2O \Rightarrow \begin{matrix} H_2O,OH \\ O_2,H_2 \end{matrix}$$
3400 K
1 atm

Analysis The reaction equation during this process can be expressed as

$$H_2O \longrightarrow x\,H_2O + y\,H_2 + z\,O_2 + w\,OH$$

Mass balances for hydrogen and oxygen yield

H balance: $2 = 2x + 2y + w$ (1)

O balance: $1 = x + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the K_p relations) to determine the equilibrium composition of the mixture. They are

$$H_2O \Leftrightarrow H_2 + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$H_2O \Leftrightarrow \tfrac{1}{2}H_2 + OH \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3400 K are determined from Table A-28 to be

$$\ln K_{P1} = -1.891 \longrightarrow K_{P1} = 0.15092$$
$$\ln K_{P2} = -1.576 \longrightarrow K_{P2} = 0.20680$$

The K_p relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{O_2}-\nu_{H_2O})} \quad \text{and} \quad K_{P2} = \frac{N_{H_2}^{\nu_{H_2}} N_{OH}^{\nu_{OH}}}{N_{H_2O}^{\nu_{H_2O}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{OH}-\nu_{H_2O})}$$

where $N_{\text{total}} = N_{H_2O} + N_{H_2} + N_{O_2} + N_{OH} = x + y + z + w$

Substituting,

$$0.15092 = \frac{(y)(z)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$0.20680 = \frac{(w)(y)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns x, y, z, and w yields

$x = 0.574$ $y = 0.308$ $z = 0.095$ $w = 0.236$

Therefore, the equilibrium composition becomes

$$\mathbf{0.574H_2O + 0.308H_2 + 0.095O_2 + 0.236OH}$$

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16-46 Two chemical reactions are occurring in a mixture. The equilibrium composition at a specified temperature is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O_2, and O. **2** The constituents of the mixture are ideal gases.

Analysis The reaction equation during this process can be expressed as

$$2\,CO_2 + O_2 \longrightarrow x\,CO_2 + y\,CO + z\,O_2 + w\,O$$

CO_2, CO, O_2, O
2000 K
4 atm

Mass balances for carbon and oxygen yield

C balance: $2 = x + y$ (1)

O balance: $6 = 2x + y + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the K_P relations) to determine the equilibrium composition of the mixture. They are

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$O_2 \Leftrightarrow 2O \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 2000 K are determined from Table A-28 to be

$$\ln K_{P1} = -6.635 \longrightarrow K_{P1} = 0.001314$$
$$\ln K_{P2} = -14.622 \longrightarrow K_{P2} = 4.464 \times 10^{-7}$$

The K_P relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{CO}+\nu_{O_2}-\nu_{CO_2})}$$

$$K_{P2} = \frac{N_{O}^{\nu_{O}}}{N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{\nu_{O}-\nu_{O_2}}$$

where

$$N_{\text{total}} = N_{CO_2} + N_{O_2} + N_{CO} + N_{O} = x + y + z + w$$

Substituting,

$$0.001314 = \frac{(y)(z)^{1/2}}{x}\left(\frac{4}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$4.464 \times 10^{-7} = \frac{w^2}{z}\left(\frac{4}{x+y+z+w}\right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously using an equation solver such as EES for the four unknowns x, y, z, and w yields

$x = 1.998$ $y = 0.002272$ $z = 1.001$ $w = 0.000579$

Thus the equilibrium composition is

$$\mathbf{1.998CO_2 + 0.002272CO + 1.001O_2 + 0.000579O}$$

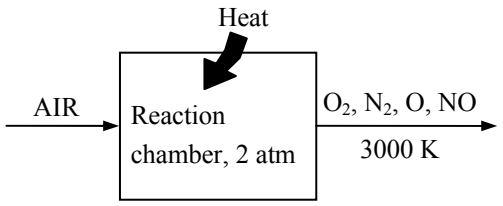
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16-47 Two chemical reactions are occurring at high-temperature air. The equilibrium composition at a specified temperature is to be determined.

Assumptions **1** The equilibrium composition consists of O_2, N_2, O, and NO. **2** The constituents of the mixture are ideal gases.

Analysis The reaction equation during this process can be expressed as

$$O_2 + 3.76\,N_2 \longrightarrow x\,N_2 + y\,NO + z\,O_2 + w\,O$$



Mass balances for nitrogen and oxygen yield

N balance: $7.52 = 2x + y$ (1)

O balance: $2 = y + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the K_p relations) to determine the equilibrium composition of the mixture. They are

$$\tfrac{1}{2}N_2 + \tfrac{1}{2}O_2 \Leftrightarrow NO \qquad \text{(reaction 1)}$$

$$O_2 \Leftrightarrow 2O \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3000 K are determined from Table A-28 to be

$$\ln K_{P1} = -2.114 \longrightarrow K_{P1} = 0.12075$$

$$\ln K_{P2} = -4.357 \longrightarrow K_{P2} = 0.01282$$

The K_P relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{NO}^{\nu_{NO}}}{N_{N_2}^{\nu_{N_2}} N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{total}}\right)^{(\nu_{NO}-\nu_{N_2}-\nu_{O_2})}$$

$$K_{P2} = \frac{N_{O}^{\nu_{O}}}{N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{total}}\right)^{\nu_{O}-\nu_{O_2}}$$

where $N_{total} = N_{N_2} + N_{NO} + N_{O_2} + N_O = x + y + z + w$

Substituting,

$$0.12075 = \frac{y}{x^{0.5} z^{0.5}} \left(\frac{2}{x+y+z+w}\right)^{1-0.5-0.5} \qquad (3)$$

$$0.01282 = \frac{w^2}{z} \left(\frac{2}{x+y+z+w}\right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously using EES for the four unknowns x, y, z, and w yields

$x = 3.656$ $y = 0.2086$ $z = 0.8162$ $w = 0.1591$

Thus the equilibrium composition is

$$\mathbf{3.656N_2 + 0.2086NO + 0.8162O_2 + 0.1591O}$$

The equilibrium constant of the reaction $N_2 \Leftrightarrow 2N$ at 3000 K is $\ln K_P$ = -22.359, which is much smaller than the K_P values of the reactions considered. Therefore, it is reasonable to assume that no N will be present in the equilibrium mixture.

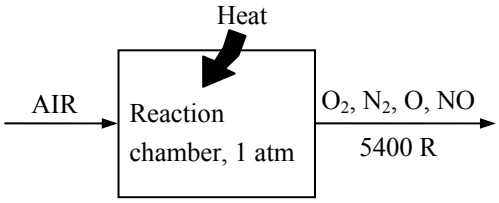
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16-48E Two chemical reactions are occurring in air. The equilibrium composition at a specified temperature is to be determined.

Assumptions **1** The equilibrium composition consists of O_2, N_2, O, and NO. **2** The constituents of the mixture are ideal gases.

Analysis The reaction equation during this process can be expressed as

$$O_2 + 3.76\,N_2 \longrightarrow x\,N_2 + y\,NO + z\,O_2 + w\,O$$



Mass balances for nitrogen and oxygen yield

N balance: $7.52 = 2x + y$ (1)

O balance: $2 = y + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the K_p relations) to determine the equilibrium composition of the mixture. They are

$$\tfrac{1}{2}N_2 + \tfrac{1}{2}O_2 \Leftrightarrow NO \qquad \text{(reaction 1)}$$

$$O_2 \Leftrightarrow 2O \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at T = 5400 R = 3000 K are determined from Table A-28 to be

$$\ln K_{P1} = -2.114 \longrightarrow K_{P1} = 0.12075$$

$$\ln K_{P2} = -4.357 \longrightarrow K_{P2} = 0.01282$$

The K_P relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{NO}^{\nu_{NO}}}{N_{N_2}^{\nu_{N_2}} N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{total}}\right)^{(\nu_{NO}-\nu_{N_2}-\nu_{O_2})}$$

$$K_{P2} = \frac{N_{O}^{\nu_{O}}}{N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{total}}\right)^{\nu_{O}-\nu_{O_2}}$$

where $N_{total} = N_{N_2} + N_{NO} + N_{O_2} + N_O = x + y + z + w$

Substituting,

$$0.12075 = \frac{y}{x^{0.5} z^{0.5}} \left(\frac{1}{x+y+z+w}\right)^{1-0.5-0.5} \qquad (3)$$

$$0.01282 = \frac{w^2}{z} \left(\frac{1}{x+y+z+w}\right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns x, y, z, and w yields

$x = 3.658$ $y = 0.2048$ $z = 0.7868$ $w = 0.2216$

Thus the equilibrium composition is

$$\mathbf{3.658N_2 + 0.2048NO + 0.7868O_2 + 0.2216O}$$

The equilibrium constant of the reaction $N_2 \Leftrightarrow 2N$ at 5400 R is $\ln K_P$ = -22.359, which is much smaller than the K_P values of the reactions considered. Therefore, it is reasonable to assume that no N will be present in the equilibrium mixture.

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14-49E Problem 16-48E is reconsidered. Using EES (or other) software, the equilibrium solution is to be obtained by minimizing the Gibbs function by using the optimization capabilities built into EES. This solution technique is to be compared with that used in the previous problem.

Analysis The problem is solved using EES, and the solution is given below.

```
"This example illustrates how EES can be used to solve multi-reaction chemical equilibria
problems by directly minimizing the Gibbs function.

      0.21 O2+0.79 N2 = a O2+b O + c N2 + d NO

Two of the four coefficients, a, b, c, and d, are found by minimiming the Gibbs function at a total
pressure of 1 atm and a temperature of 5400 R.  The other two are found from mass balances.

The equilibrium solution can be found by applying the Law of Mass Action to two
simultaneous equilibrium reactions or by minimizing the Gibbs function.  In this problem, the Gibbs function is
directly minimized using the optimization capabilities built into EES.

To run this program, select MinMax from the Calculate menu.  There are four compounds present
in the products subject to two elemental balances, so there are two degrees of freedom.  Minimize
Gibbs with respect to two molar quantities such as coefficients b and d.  The equilibrium mole
numbers of each specie will be determined and displayed in the Solution window.

Minimizing the Gibbs function to find the equilibrium composition requires good initial guesses."

"Data from Data Input Window"
{T=5400 "R"
P=1 "atm" }

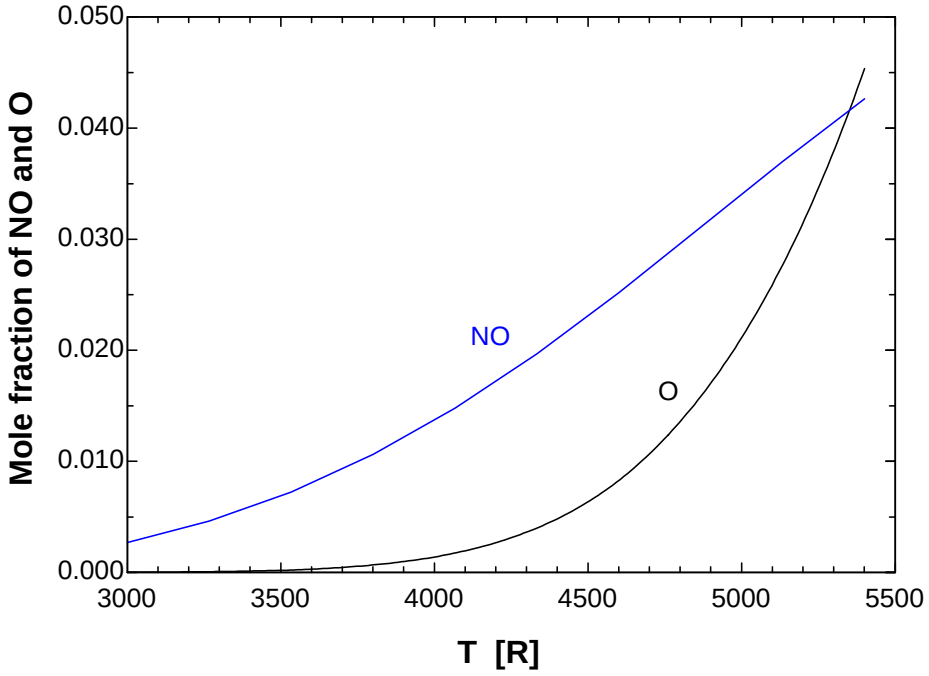
AO2=0.21; BN2=0.79 "Composition  of air"
AO2*2=a*2+b+d "Oxygen balance"
BN2*2=c*2+d "Nitrogen balance"

"The total moles at equilibrium are"
N_tot=a+b+c+d
y_O2=a/N_tot;  y_O=b/N_tot; y_N2=c/N_tot; y_NO=d/N_tot

"The following equations provide the specific Gibbs function for three of the components."
g_O2=Enthalpy(O2,T=T)-T*Entropy(O2,T=T,P=P*y_O2)
g_N2=Enthalpy(N2,T=T)-T*Entropy(N2,T=T,P=P*y_N2)
g_NO=Enthalpy(NO,T=T)-T*Entropy(NO,T=T,P=P*y_NO)
"EES does not have a built-in property function for monatomic oxygen so we will use the JANAF
procedure,  found under Options/Function Info/External Procedures.   The units for the JANAF
procedure are kgmole, K, and kJ so we must convert h and s to English units."
T_K=T*Convert(R,K) "Convert R to K"
Call JANAF('O',T_K:Cp`,h`,S`) "Units from JANAF are SI"
S_O=S`*Convert(kJ/kgmole-K, Btu/lbmole-R)
h_O=h`*Convert(kJ/kgmole, Btu/lbmole)
 "The entropy from JANAF is for one atmosphere so it must be corrected for partial pressure."
g_O=h_O-T*(S_O-R_u*ln(Y_O))
R_u=1.9858 "The universal gas constant in Btu/mole-R "
"The extensive Gibbs function is the sum of the products of the specific Gibbs function and the
molar amount of each substance."
Gibbs=a*g_O2+b*g_O+c*g_N2+d*g_NO
```

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d [lbmol]	b [lbmol]	Gibbs [Btu/lbmol]	y_{O2}	y_O	y_{NO}	y_{N2}	T [R]
0.002698	0.00001424	-162121	0.2086	0.0000	0.0027	0.7886	3000
0.004616	0.00006354	-178354	0.2077	0.0001	0.0046	0.7877	3267
0.007239	0.0002268	-194782	0.2062	0.0002	0.0072	0.7863	3533
0.01063	0.000677	-211395	0.2043	0.0007	0.0106	0.7844	3800
0.01481	0.001748	-228188	0.2015	0.0017	0.0148	0.7819	4067
0.01972	0.004009	-245157	0.1977	0.0040	0.0197	0.7786	4333
0.02527	0.008321	-262306	0.1924	0.0083	0.0252	0.7741	4600
0.03132	0.01596	-279641	0.1849	0.0158	0.0311	0.7682	4867
0.03751	0.02807	-297179	0.1748	0.0277	0.0370	0.7606	5133
0.04361	0.04641	-314941	0.1613	0.0454	0.0426	0.7508	5400



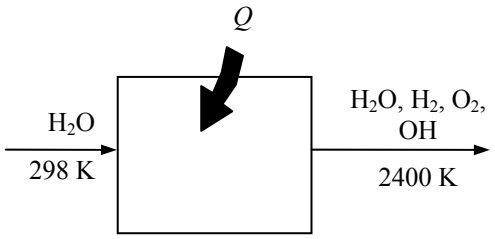
Discussion The equilibrium composition in the above table are based on the reaction in which the reactants are 0.21 kmol O_2 and 0.79 kmol N_2. If you multiply the equilibrium composition mole numbers above with 4.76, you will obtain equilibrium composition for the reaction in which the reactants are 1 kmol O_2 and 3.76 kmol N_2.This is the case in problem 16-43E.

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16-50 Water vapor is heated during a steady-flow process. The rate of heat supply for a specified exit temperature is to be determined for two cases.

Assumptions **1** The equilibrium composition consists of H_2O, OH, O_2, and H_2. **2** The constituents of the mixture are ideal gases.

Analysis (*a*) Assuming some H_2O dissociates into H_2, O_2, and O, the dissociation equation can be written as



$$H_2O \longrightarrow x\,H_2O + y\,H_2 + z\,O_2 + w\,OH$$

Mass balances for hydrogen and oxygen yield

H balance: $2 = 2x + 2y + w$ (1)

O balance: $1 = x + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the K_P relations) to determine the equilibrium composition of the mixture. They are

$$H_2O \Leftrightarrow H_2 + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$H_2O \Leftrightarrow \tfrac{1}{2}H_2 + OH \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 2400 K are determined from Table A-28 to be

$$\ln K_{P1} = -5.619 \longrightarrow K_{P1} = 0.003628$$
$$\ln K_{P2} = -5.832 \longrightarrow K_{P2} = 0.002932$$

The K_P relations for these three simultaneous reactions are

$$K_{P1} = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{O_2}-\nu_{H_2O})}$$

$$K_{P2} = \frac{N_{H_2}^{\nu_{H_2}} N_{OH}^{\nu_{OH}}}{N_{H_2O}^{\nu_{H_2O}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{OH}-\nu_{H_2O})}$$

where

$$N_{\text{total}} = N_{H_2O} + N_{H_2} + N_{O_2} + N_{OH} = x + y + z + w$$

Substituting,

$$0.003628 = \frac{(y)(z)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$0.002932 = \frac{(w)(y)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns x, y, z, and w yields

$x = 0.960$ $\quad y = 0.03204$ $\quad z = 0.01205$ $\quad w = 0.01588$

Thus the balanced equation for the dissociation reaction is

$$H_2O \longrightarrow 0.960H_2O + 0.03204H_2 + 0.01205O_2 + 0.01588OH$$

The heat transfer for this dissociation process is determined from the steady-flow energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ with $W = 0$,

$$Q_{\text{in}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

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Assuming the O_2 and O to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\,K}$ kJ/kmol	$\bar{h}_{3000\,K}$ kJ/kmol
H_2O	-241,820	9904	103,508
H_2	0	8468	75,383
O_2	0	8682	83,174
OH	39,460	9188	77,015

Substituting,

$$\begin{aligned} Q_{\text{in}} &= 0.960(-241{,}820 + 103{,}508 - 9904) \\ &\quad + 0.03204(0 + 75{,}383 - 8468) \\ &\quad + 0.01205(0 + 83{,}174 - 8682) \\ &\quad + 0.01588(39{,}460 + 77{,}015 - 9188) - (-241{,}820) \\ &= 103{,}380\ \text{kJ/kmol}\ H_2O \end{aligned}$$

The mass flow rate of H_2O can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{0.6\ \text{kg/min}}{18\ \text{kg/kmol}} = 0.03333\ \text{kmol/min}$$

Thus,

$$\dot{Q}_{\text{in}} = \dot{N} \times Q_{\text{in}} = (0.03333\ \text{kmol/min})(103{,}380\ \text{kJ/kmol}) = \mathbf{3446\ kJ/min}$$

(*b*) If no dissociates takes place, then the process involves no chemical reactions and the heat transfer can be determined from the steady-flow energy balance for nonreacting systems to be

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}(h_2 - h_1) = \dot{N}(\bar{h}_2 - \bar{h}_1) \\ &= (0.03333\ \text{kmol/min})(103{,}508 - 9904)\ \text{kJ/kmol} \\ &= \mathbf{3120\ kJ/min} \end{aligned}$$

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16-51 Problem 16-50 is reconsidered. The effect of the final temperature on the rate of heat supplied for the two cases is to be studied.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
T1=298 [K]
"T2=2400 [K]"
P=1 [atm]
m_dot=0.6 [kg/min]
T0=298 [K]

"The equilibrium constant for these two reactions at 2400 K are determined from Table A-28"
K_p1=exp(-5.619)
K_p2=exp(-5.832)

"Properties"
MM_H2O=molarmass(H2O)

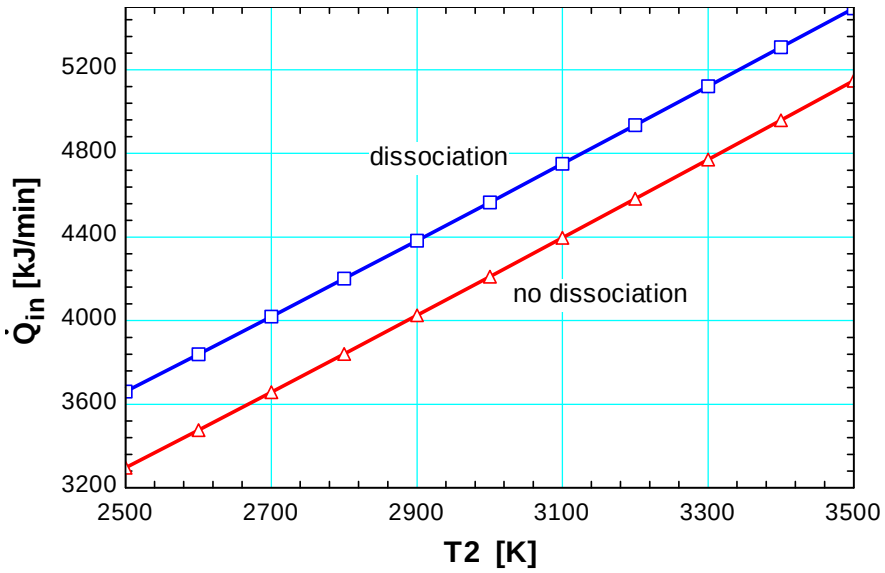
"Analysis"
"(a)"
"Actual reaction:  H2O = N_H2O H2O + N_H2 H2 + N_O2 O2 + N_OH OH"
2=2*N_H2O+2*N_H2+N_OH "H balance"
1=N_H2O+2*N_O2+N_OH "O balance"
N_total=N_H2O+N_H2+N_O2+N_OH
"Stoichiometric reaction 1:  H2O = H2 + 1/2 O2"
"Stoichiometric coefficients for reaction 1"
nu_H2O_1=1
nu_H2_1=1
nu_O2_1=1/2
"Stoichiometric reaction 2:  H2O = 1/2 H2 + OH"
"Stoichiometric coefficients for reaction 2"
nu_H2O_2=1
nu_H2_2=1/2
nu_OH_2=1
"K_p relations are"
K_p1=(N_H2^nu_H2_1*N_O2^nu_O2_1)/N_H2O^nu_H2O_1*(P/N_total)^(nu_H2_1+nu_O2_1-nu_H2O_1)
K_p2=(N_H2^nu_H2_2*N_OH^nu_OH_2)/N_H2O^nu_H2O_2*(P/N_total)^(nu_H2_2+nu_OH_2-nu_H2O_2)

"Enthalpy of formation data from Table A-26"
h_f_OH=39460
"Enthalpies of products"
h_H2O_R=enthalpy(H2O, T=T1)
h_H2O_P=enthalpy(H2O, T=T2)
h_H2=enthalpy(H2, T=T2)
h_O2=enthalpy(O2, T=T2)
h_OH=98763 "at T2 from the ideal gas tables in the text"
"Standard state enthalpies"
h_o_OH=9188 "at T0 from the ideal gas tables in the text"

"Heat transfer"
H_P=N_H2O*h_H2O_P+N_H2*h_H2+N_O2*h_O2+N_OH*(h_f_OH+h_OH-h_o_OH)
H_R=N_H2O_R*h_H2O_R
N_H2O_R=1
Q_in_a=H_P-H_R
Q_dot_in_a=(m_dot/MM_H2O)*Q_in_a
"(b)"
Q_in_b=N_H2O_R*(h_H2O_P-h_H2O_R)
Q_dot_in_b=(m_dot/MM_H2O)*Q_in_b
```

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T_{prod} [K]	$Q_{in,Dissoc}$ [kJ/min]	$Q_{in,NoDissoc}$ [kJ/min]
2500	3660	3295
2600	3839	3475
2700	4019	3657
2800	4200	3840
2900	4382	4024
3000	4566	4210
3100	4750	4396
3200	4935	4583
3300	5121	4771
3400	5307	4959
3500	5494	5148



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16-52 Ethyl alcohol C_2H_5OH (gas) is burned in a steady-flow adiabatic combustion chamber with 40 percent excess air. The adiabatic flame temperature of the products is to be determined and the adiabatic flame temperature as a function of the percent excess air is to be plotted.

Analysis The complete combustion reaction in this case can be written as

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{\text{th}}\left[O_2 + 3.76N_2\right] \longrightarrow 2\,CO_2 + 3\,H_2O + (Ex)(a_{\text{th}})\,O_2 + f\,N_2$$

where a_{th} is the stoichiometric coefficient for air. The oxygen balance gives

$$1 + (1+Ex)a_{\text{th}} \times 2 = 2\times 2 + 3\times 1 + (Ex)(a_{\text{th}})\times 2$$

The reaction equation with products in equilibrium is

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{\text{th}}\left[O_2 + 3.76N_2\right] \longrightarrow a\,CO_2 + b\,CO + d\,H_2O + e\,O_2 + f\,N_2 + g\,NO$$

The coefficients are determined from the mass balances

Carbon balance: $2 = a + b$

Hydrogen balance: $6 = 2d \longrightarrow d = 3$

Oxygen balance: $1 + (1+Ex)a_{\text{th}} \times 2 = a\times 2 + b + d + e\times 2 + g$

Nitrogen balance: $(1+Ex)a_{\text{th}} \times 3.76 \times 2 = f\times 2 + g$

Solving the above equations, we find the coefficients to be

$$Ex = 0.4,\ a_{\text{th}} = 3,\ a = 1.995,\ b = 0.004712,\ d = 3,\ e = 1.17,\ f = 15.76,\ g = 0.06428$$

Then, we write the balanced reaction equation as

$$C_2H_5OH\,(\text{gas}) + 4.2\left[O_2 + 3.76N_2\right]$$
$$\longrightarrow 1.995\,CO_2 + 0.004712\,CO + 3\,H_2O + 1.17\,O_2 + 15.76\,N_2 + 0.06428\,NO$$

Total moles of products at equilibrium are

$$N_{\text{tot}} = 1.995 + 0.004712 + 3 + 1.17 + 15.76 = 21.99$$

The first assumed equilibrium reaction is

$$CO_2 \longleftrightarrow CO + 0.5O_2$$

The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_{p1} = \exp\left(\frac{-\Delta G_1^*(T_{\text{prod}})}{R_u T_{\text{prod}}}\right)$$

Where $\Delta G_1^*(T_{\text{prod}}) = \nu_{CO}\,\bar{g}^*_{CO}(T_{\text{prod}}) + \nu_{O2}\,\bar{g}^*_{O2}(T_{\text{prod}}) - \nu_{CO2}\,\bar{g}^*_{CO2}(T_{\text{prod}})$

and the Gibbs functions are defined as

$$\bar{g}^*_{CO}(T_{\text{prod}}) = (\bar{h} - T_{\text{prod}}\bar{s})_{CO}$$
$$\bar{g}^*_{O2}(T_{\text{prod}}) = (\bar{h} - T_{\text{prod}}\bar{s})_{O2}$$
$$\bar{g}^*_{CO2}(T_{\text{prod}}) = (\bar{h} - T_{\text{prod}}\bar{s})_{CO2}$$

The equilibrium constant is also given by

$$K_{p1} = \frac{be^{0.5}}{a}\left(\frac{P}{N_{\text{tot}}}\right)^{1+0.5-1} = \frac{(0.004712)(1.17)^{0.5}}{1.995}\left(\frac{1}{21.99}\right)^{0.5} = 0.0005447$$

The second assumed equilibrium reaction is

$$0.5N_2 + 0.5O_2 \longleftrightarrow NO$$

Also, for this reaction, we have

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$$\overline{g}^*_{NO}(T_{prod}) = (\overline{h} - T_{prod}\overline{s})_{NO}$$

$$\overline{g}^*_{N2}(T_{prod}) = (\overline{h} - T_{prod}\overline{s})_{N2}$$

$$\overline{g}^*_{O2}(T_{prod}) = (\overline{h} - T_{prod}\overline{s})_{O2}$$

$$\Delta G_2{}^*(T_{prod}) = \nu_{NO}\overline{g}^*_{NO}(T_{prod}) - \nu_{N2}\overline{g}^*_{N2}(T_{prod}) - \nu_{O2}\overline{g}^*_{O2}(T_{prod})$$

$$K_{p2} = \exp\left(\frac{-\Delta G_2{}^*(T_{prod})}{R_u T_{prod}}\right)$$

$$K_{p2} = \left(\frac{P}{N_{tot}}\right)^{1-0.5-0.5} \frac{g}{e^{o.5} f^{0.5}} = \left(\frac{1}{21.99}\right)^0 \frac{0.06428}{(1.17)^{0.5}(15.76)^{0.5}} = 0.01497$$

A steady flow energy balance gives

$$H_R = H_P$$

where

$$H_R = \overline{h}^o_{f\ fuel@25°C} + 4.2\overline{h}_{O2@25°C} + 15.79\overline{h}_{N2@25°C}$$
$$= (-235{,}310\ \text{kJ/kmol}) + 4.2(0) + 15.79(0) = -235{,}310\ \text{kJ/kmol}$$

$$H_P = 1.995\overline{h}_{CO2@T_{prod}} + 0.004712\overline{h}_{CO@T_{prod}} + 3h_{H2O@T_{prod}} + 1.17\overline{h}_{O2@T_{prod}} + 15.76\overline{h}_{N2@T_{prod}} + 0.06428\overline{h}_{NO@T_{prod}}$$

Solving the energy balance equation using EES, we obtain the adiabatic flame temperature

$$T_{prod} = \mathbf{1901\ K}$$

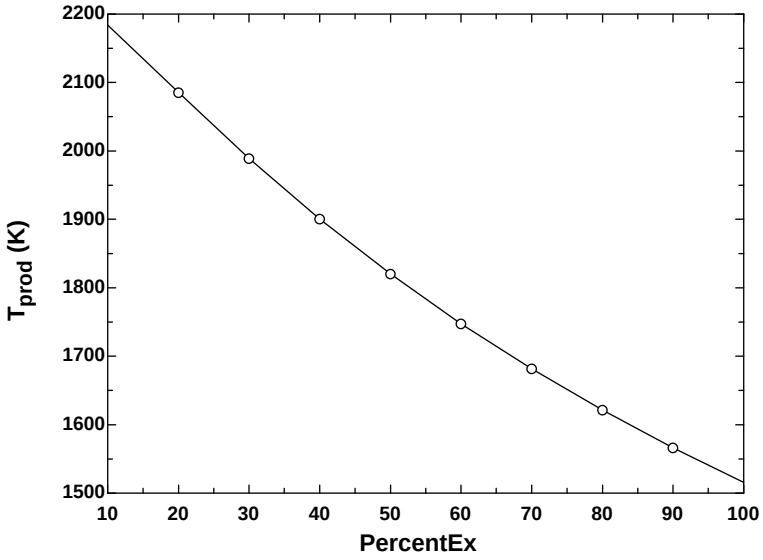
The copy of entire EES solution including parametric studies is given next:

```
"The reactant temperature is:"
T_reac= 25+273 "[K]"
"For adiabatic combustion of 1 kmol of fuel: "
Q_out = 0 "[kJ]"
PercentEx = 40  "Percent excess air"
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3"[kPa]"
R_u=8.314 "[kJ/kmol-K]"
"The complete combustion reaction equation for excess air  is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=2 CO2 +  3 H2O + Ex*A_th O2 + f N2 "
"Oxygen Balance for complete combustion:"
1 + (1+Ex)*A_th*2=2*2+3*1 + Ex*A_th*2
"The reaction equation for excess air and products in equilibrium is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=a CO2 + b CO+ d H2O + e O2 + f N2 + g NO"
"Carbon Balance:"
2=a + b
"Hydrogen Balance:"
6=2*d
"Oxygen Balance:"
1 + (1+Ex)*A_th*2=a*2+b + d + e*2 +g
"Nitrogen Balance:"
(1+Ex)*A_th*3.76 *2= f*2 + g
N_tot =a +b + d + e + f +g "Total kilomoles of products at equilibrium"
"The first assumed equilibrium reaction is CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
```

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"The standard-state Gibbs function is"
DELTAG_1 =1*g_CO+0.5*g_O2-1*g_CO2
"The equilibrium constant is given by Eq. 15-14."
K_P_1 = exp(-DELTAG_1 /(R_u*T_prod))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P_1 = (P/N_tot)^(1+0.5-1)*(b^1*e^0.5)/(a^1)"
sqrt(P/N_tot) *b *sqrt(e) =K_P_1*a
"The econd assumed equilibrium reaction is 0.5N2+0.5O2=NO"
g_NO=Enthalpy(NO,T=T_prod)-T_prod *Entropy(NO,T=T_prod ,P=101.3)
g_N2=Enthalpy(N2,T=T_prod)-T_prod *Entropy(N2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG_2 =1*g_NO-0.5*g_O2-0.5*g_N2
"The equilibrium constant is given by Eq. 15-14."
K_P_2 = exp(-DELTAG_2 /(R_u*T_prod))
"The equilibrium constant is also given by Eq. 15-15."
"K_ P_2 = (P/N_tot)^(1-0.5-0.5)*(g^1)/(e^0.5*f^0.5)"
g=K_P_2 *sqrt(e*f)
"The steady-flow energy balance is:"
H_R = Q_out+H_P
h_bar_f_C2H5OHgas=-235310 "[kJ/kmol]"
H_R=1*(h_bar_f_C2H5OHgas)
+(1+Ex)*A_th*ENTHALPY(O2,T=T_reac)+(1+Ex)*A_th*3.76*ENTHALPY(N2,T=T_reac) "[kJ/kmol]"
H_P=a*ENTHALPY(CO2,T=T_prod)+b*ENTHALPY(CO,T=T_prod)+d*ENTHALPY(H2O,T=T_prod)+e*ENTHAL
PY(O2,T=T_prod)+f*ENTHALPY(N2,T=T_prod)+g*ENTHALPY(NO,T=T_prod) "[kJ/kmol]"

a_{th}	a	b	d	e	f	g	PercentEx [%]	T_{prod} [K]
3	1.922	0.07779	3	0.3081	12.38	0.0616	10	2184
3	1.971	0.0293	3	0.5798	13.5	0.06965	20	2085
3	1.988	0.01151	3	0.8713	14.63	0.06899	30	1989
3	1.995	0.004708	3	1.17	15.76	0.06426	40	1901
3	1.998	0.001993	3	1.472	16.89	0.05791	50	1820
3	1.999	0.0008688	3	1.775	18.02	0.05118	60	1747
3	2	0.0003884	3	2.078	19.15	0.04467	70	1682
3	2	0.0001774	3	2.381	20.28	0.03867	80	1621
3	2	0.00008262	3	2.683	21.42	0.0333	90	1566
3	2	0.00003914	3	2.986	22.55	0.02856	100	1516



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Variations of K_p with Temperature

16-53C It enables us to determine the enthalpy of reaction $\overline{h}_R$ from a knowledge of equilibrium constant K_P.

16-54C At 2000 K since combustion processes are exothermic, and exothermic reactions are more complete at lower temperatures.

16-55 The $\overline{h}_R$ value for the dissociation process $O_2 \Leftrightarrow 2O$ at a specified temperature is to be determined using enthalpy and K_P data.

Assumptions Both the reactants and products are ideal gases.

Analysis (*a*) The dissociation equation of O_2 can be expressed as

$$O_2 \Leftrightarrow 2O$$

The $\overline{h}_R$ of the dissociation process of O_2 at 3100 K is the amount of energy absorbed or released as one kmol of O_2 dissociates in a steady-flow combustion chamber at a temperature of 3100 K, and can be determined from

$$\overline{h}_R = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the O_2 and O to be ideal gases, we have $h = h\,(T)$. From the tables,

Substance	$\overline{h}_f^\circ$ kJ/kmol	$\overline{h}_{298\,K}$ kJ/kmol	$\overline{h}_{2900\,K}$ kJ/kmol
O	249,190	6852	65,520
O_2	0	8682	110,784

Substituting,

$$\overline{h}_R = 2(249{,}190 + 65{,}520 - 6852) - 1(0 + 110{,}784 - 8682)$$
$$= \mathbf{513{,}614\ kJ/kmol}$$

(*b*) The $\overline{h}_R$ value at 3100 K can be estimated by using K_P values at 3000 K and 3200 K (the closest two temperatures to 3100 K for which K_P data are available) from Table A-28,

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$-3.072 - (-4.357) \cong \frac{\overline{h}_R}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{3000\ \text{K}} - \frac{1}{3200\ \text{K}}\right)$$

$$\overline{h}_R \cong \mathbf{512{,}808\ kJ/kmol}$$

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16-56 The $\bar{h}_R$ at a specified temperature is to be determined using the enthalpy and K_P data.

Assumptions Both the reactants and products are ideal gases.

Analysis (*a*) The complete combustion equation of CO can be expressed as

$$CO + \tfrac{1}{2}O_2 \Leftrightarrow CO_2$$

The $\bar{h}_R$ of the combustion process of CO at 2200 K is the amount of energy released as one kmol of CO is burned in a steady-flow combustion chamber at a temperature of 2200 K, and can be determined from

$$\bar{h}_R = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the CO, O_2 and CO_2 to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\,K}$ kJ/kmol	$\bar{h}_{2200\,K}$ kJ/kmol
CO_2	-393,520	9364	112,939
CO	-110,530	8669	72,688
O_2	0	8682	75,484

Substituting,

$$\begin{aligned}\bar{h}_R &= 1(-393{,}520 + 112{,}939 - 9364) \\ &\quad - 1(-110{,}530 + 72{,}688 - 8669) \\ &\quad - 0.5(0 + 75{,}484 - 8682) \\ &= \mathbf{-276{,}835\ kJ/kmol}\end{aligned}$$

(*b*) The $\bar{h}_R$ value at 2200 K can be estimated by using K_P values at 2000 K and 2400 K (the closest two temperatures to 2200 K for which K_P data are available) from Table A-28,

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$3.860 - 6.635 \cong \frac{\bar{h}_R}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{2000\ \text{K}} - \frac{1}{2400\ \text{K}}\right)$$

$$\bar{h}_R \cong \mathbf{-276{,}856\ kJ/kmol}$$

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16-57E The $\bar{h}_R$ at a specified temperature is to be determined using the enthalpy and K_P data.

Assumptions Both the reactants and products are ideal gases.

Analysis (*a*) The complete combustion equation of CO can be expressed as

$$CO + \tfrac{1}{2} O_2 \Leftrightarrow CO_2$$

The $\bar{h}_R$ of the combustion process of CO at 3960 R is the amount of energy released as one kmol of H_2 is burned in a steady-flow combustion chamber at a temperature of 3960 R, and can be determined from

$$\bar{h}_R = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the CO, O_2 and CO_2 to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{\mathbf{h}}_\mathbf{f}^\circ$ **Btu/lbmol**	$\bar{\mathbf{h}}_{\mathbf{537\ R}}$ **Btu/lbmol**	$\bar{\mathbf{h}}_{\mathbf{3960\ R}}$ **Btu/lbmol**
CO_2	-169,300	4027.5	48,647
CO	-47,540	3725.1	31,256.5
O_2	0	3725.1	32,440.5

Substituting,

$$\begin{aligned}\bar{h}_R &= 1(-169{,}300 + 48{,}647 - 4027.5) \\ &\quad - 1(-47{,}540 + 31{,}256.5 - 3725.1) \\ &\quad - 0.5(0 + 32{,}440.5 - 3725.1) \\ &= \mathbf{-119{,}030\ Btu / lbmol}\end{aligned}$$

(*b*) The $\bar{h}_R$ value at 3960 R can be estimated by using K_P values at 3600 R and 4320 R (the closest two temperatures to 3960 R for which K_P data are available) from Table A-28,

$$\ln \frac{K_{P2}}{K_{P1}} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$3.860 - 6.635 \cong \frac{\bar{h}_R}{1.986\ \text{Btu/lbmol}\cdot\text{R}}\left(\frac{1}{3600\ \text{R}} - \frac{1}{4320\ \text{R}}\right)$$

$$\bar{h}_R \cong \mathbf{-119{,}041\ Btu/lbmol}$$

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16-58 The K_P value of the combustion process $H_2 + 1/2O_2 \Leftrightarrow H_2O$ is to be determined at a specified temperature using $\bar{h}_R$ data and K_P value .

Assumptions Both the reactants and products are ideal gases.

Analysis The $\bar{h}_R$ and K_P data are related to each other by

$$\ln \frac{K_{P2}}{K_{P1}} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1}-\frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1}-\frac{1}{T_2}\right)$$

The $\bar{h}_R$ of the specified reaction at 3000 K is the amount of energy released as one kmol of H_2 is burned in a steady-flow combustion chamber at a temperature of 3000 K, and can be determined from

$$\bar{h}_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the H_2O, H_2 and O_2 to be ideal gases, we have $h = h\,(T)$. From the tables,

Substance	$\bar{\mathbf{h}}_\mathbf{f}^\circ$ kJ/kmol	$\bar{\mathbf{h}}_{\mathbf{298\,K}}$ kJ/kmol	$\bar{\mathbf{h}}_{\mathbf{3000K}}$ kJ/kmol
H_2O	-241,820	9904	136,264
H_2	0	8468	97,211
O_2	0	8682	106,780

Substituting,

$$\begin{aligned}\bar{h}_R &= 1(-241{,}820 + 136{,}264 - 9904)\\ &\quad - 1(0 + 97{,}211 - 8468)\\ &\quad - 0.5(0 + 106{,}780 - 8682)\\ &= -253{,}250\ \text{kJ/kmol}\end{aligned}$$

The K_P value at 3200 K can be estimated from the equation above by using this $\bar{h}_R$ value and the K_P value at 2800 K which is $\ln K_{P1} = 3.812$ or $K_{P1} = 45.24$,

$$\ln(K_{P2}/45.24) \cong \frac{-253{,}250\ \text{kJ/kmol}}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{2800\ \text{K}}-\frac{1}{3200\ \text{K}}\right)$$

$$\ln K_{P2} = 2.452 \quad (\text{Table A - 28}: \ln K_{P2} = 2.451)$$

or

$$K_{P2} = \mathbf{11.6}$$

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16-59 The $\overline{h}_R$ value for the dissociation process $CO_2 \Leftrightarrow CO + 1/2O_2$ at a specified temperature is to be determined using enthalpy and K_p data.

Assumptions Both the reactants and products are ideal gases.

Analysis (*a*) The dissociation equation of CO_2 can be expressed as

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2} O_2$$

The $\overline{h}_R$ of the dissociation process of CO_2 at 2200 K is the amount of energy absorbed or released as one kmol of CO_2 dissociates in a steady-flow combustion chamber at a temperature of 2200 K, and can be determined from

$$\overline{h}_R = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the CO, O_2 and CO_2 to be ideal gases, we have $h = h\,(T)$. From the tables,

Substance	$\overline{h}_f^\circ$ kJ/kmol	$\overline{h}_{298\,K}$ kJ/kmol	$\overline{h}_{2200\,K}$ kJ/kmol
CO_2	-393,520	9364	112,939
CO	-110,530	8669	72,688
O_2	0	8682	75,484

Substituting,

$$\begin{aligned}\overline{h}_R &= 1(-110{,}530 + 72{,}688 - 8669) \\ &\quad + 0.5(0 + 75{,}484 - 8682) \\ &\quad - 1(-393{,}520 + 112{,}939 - 9364) \\ &= \mathbf{276{,}835\ kJ\ /\ kmol}\end{aligned}$$

(*b*) The $\overline{h}_R$ value at 2200 K can be estimated by using K_P values at 2000 K and 2400 K (the closest two temperatures to 2200 K for which K_P data are available) from Table A-28,

$$\ln \frac{K_{P2}}{K_{P1}} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$-3.860 - (-6.635) \cong \frac{\overline{h}_R}{8.314\,\text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{2000\,\text{K}} - \frac{1}{2400\,\text{K}}\right)$$

$$\overline{h}_R \cong \mathbf{276{,}856\ kJ/kmol}$$

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16-60 The enthalpy of reaction for the equilibrium reaction CH4 + 2O2 = CO2 + 2H2O at 2000 K is to be estimated using enthalpy data and equilibrium constant, K_p data.

Analysis The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{CO2}\overline{g}^*_{CO2}(T) + \nu_{H2O}\overline{g}^*_{H2O}(T) - \nu_{CH4}\overline{g}^*_{CH4}(T) - \nu_{O2}\overline{g}^*_{O2}(T)$$

At T_1 = 2000 - 10 = 1990 K:

$$\begin{aligned}\Delta G_1{}^*(T) &= \nu_{CO2}\overline{g}^*_{CO2}(T_1) + \nu_{H2O}\overline{g}^*_{H2O}(T_1) - \nu_{CH4}\overline{g}^*_{CH4}(T_1) - \nu_{O2}\overline{g}^*_{O2}(T_1)\\ &= 1(-917{,}176) + 2(-695{,}638) - 1(-559{,}718) - 2(-475{,}399)\\ &= -797{,}938\ \text{kJ/kmol}\end{aligned}$$

At T_2 = 2000 + 10 = 2010 K:

$$\begin{aligned}\Delta G_2{}^*(T) &= \nu_{CO2}\overline{g}^*_{CO2}(T_2) + \nu_{H2O}\overline{g}^*_{H2O}(T_2) - \nu_{CH4}\overline{g}^*_{CH4}(T_2) - \nu_{O2}\overline{g}^*_{O2}(T_2)\\ &= 1(-923{,}358) + 2(-700{,}929) - 1(-565{,}835) - 2(-480{,}771)\\ &= -797{,}839\ \text{kJ/kmol}\end{aligned}$$

The Gibbs functions are obtained from enthalpy and entropy properties using EES. Substituting,

$$K_{p1} = \exp\left(-\frac{-797{,}938\ \text{kJ/kmol}}{(8.314\ \text{kJ/kmol}\cdot\text{K})(1990\ \text{K})}\right) = 8.820\times10^{20}$$

$$K_{p2} = \exp\left(-\frac{-797{,}839\ \text{kJ/kmol}}{(8.314\ \text{kJ/kmol}\cdot\text{K})(2010\ \text{K})}\right) = 5.426\times10^{20}$$

The enthalpy of reaction is determined by using the integrated van't Hoff equation:

$$\ln\left(\frac{K_{p2}}{K_{p1}}\right) = \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$\ln\left(\frac{5.426\times10^{20}}{8.820\times10^{20}}\right) = \frac{\overline{h}_R}{8.314\ \text{kJ/kmol.K}}\left(\frac{1}{1990\ \text{K}} - \frac{1}{2010\ \text{K}}\right) \longrightarrow \overline{h}_R = \mathbf{-807{,}752\ kJ/kmol}$$

The enthalpy of reaction can also be determined from an energy balance to be

$$\overline{h}_R = H_P - H_R$$

where

$$H_R = 1\overline{h}_{CH4\,@\,2000\,K} + 2\overline{h}_{O2\,@\,2000\,K} = 48{,}947 + 2(59{,}193) = 167{,}333\ \text{kJ/kmol}$$

$$H_P = 1\overline{h}_{CO2\,@\,2000\,K} + 2\overline{h}_{H2O\,@\,2000\,K} = (-302{,}094) + 2(-169{,}162) = -640{,}419\ \text{kJ/kmol}$$

The enthalpies are obtained from EES. Substituting,

$$\overline{h}_R = H_P - H_R = (-640{,}419) - (167{,}333) = \mathbf{-807{,}752\ kJ/kmol}$$

which is identical to the value obtained using K_p data.

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Phase Equilibrium

16-61C No. Because the specific gibbs function of each phase will not be affected by this process; i.e., we will still have $g_f = g_g$.

16-62C Yes. Because the number of independent variables for a two-phase (PH=2), two-component (C=2) mixture is, from the phase rule,

$$\text{IV} = C - \text{PH} + 2 = 2 - 2 + 2 = 2$$

Therefore, two properties can be changed independently for this mixture. In other words, we can hold the temperature constant and vary the pressure and still be in the two-phase region. Notice that if we had a single component (C=1) two phase system, we would have IV=1, which means that fixing one independent property automatically fixes all the other properties.

11-63C Using solubility data of a solid in a specified liquid, the mass fraction w of the solid A in the liquid at the interface at a specified temperature can be determined from

$$\text{mf}_A = \frac{m_{\text{solid}}}{m_{\text{solid}} + m_{\text{liquid}}}$$

where m_{solid} is the maximum amount of solid dissolved in the liquid of mass m_{liquid} at the specified temperature.

11-64C The molar concentration C_{i} of the gas species i in the solid at the interface $C_{\text{i, solid side}}(0)$ is proportional to the *partial pressure* of the species i in the gas $P_{\text{i, gas side}}(0)$ on the gas side of the interface, and is determined from

$$C_{\text{i, solid side}}(0) = \text{S} \times P_{\text{i, gas side}}(0) \qquad (\text{kmol/m}^3)$$

where S is the *solubility* of the gas in that solid at the specified temperature.

11-65C Using Henry's constant data for a gas dissolved in a liquid, the mole fraction of the gas dissolved in the liquid at the interface at a specified temperature can be determined from Henry's law expressed as

$$y_{\text{i, liquid side}}(0) = \frac{P_{\text{i, gas side}}(0)}{H}$$

where H is *Henry's constant* and $P_{\text{i,gas side}}(0)$ is the partial pressure of the gas i at the gas side of the interface. This relation is applicable for dilute solutions (gases that are weakly soluble in liquids).

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16-66E The maximum partial pressure of the water evaporated into the air as it emerges from a porous media is to be determined.

Assumptions The air and water-air solution behave as ideal solutions so that Raoult's law may be used.

Analysis The saturation temperature of water at 70°F is

$$P_{\text{sat@70°F}} = 0.36334\,\text{psia}$$

Since the mole fraction of the air in the liquid water is essentially zero,

$$P_{v,\max} = 1 \times P_{\text{sat@70°F}} = \mathbf{0.36334\ psia}$$

16-67 The number of independent properties needed to fix the state of a mixture of oxygen and nitrogen in the gas phase is to be determined.

Analysis In this case the number of components is $C = 2$ and the number of phases is PH = 1. Then the number of independent variables is determined from the phase rule to be

$$\text{IV} = C - \text{PH} + 2 = 2 - 1 + 2 = 3$$

Therefore, three independent properties need to be specified to fix the state. They can be temperature, the pressure, and the mole fraction of one of the gases.

16-68 It is to be shown that a saturated liquid-vapor mixture of refrigerant-134a at -20°C satisfies the criterion for phase equilibrium.

Analysis Using the definition of Gibbs function and enthalpy and entropy data from Table A-11,

$$g_f = h_f - Ts_f = (25.49\,\text{kJ/kg}) - (253.15\,\text{K})(0.10463\ \text{kJ/kg}\cdot\text{K}) = -0.9967\,\text{kJ/kg}$$
$$g_g = h_g - Ts_g = (238.41\,\text{kJ/kg}) - (253.15\,\text{K})(0.94564\ \text{kJ/kg}\cdot\text{K}) = -0.9842\,\text{kJ/kg}$$

which are sufficiently close. Therefore, the criterion for phase equilibrium is satisfied.

16-69 It is to be shown that a mixture of saturated liquid water and saturated water vapor at 300 kPa satisfies the criterion for phase equilibrium.

Analysis The saturation temperature at 300 kPa is 406.7 K. Using the definition of Gibbs function and enthalpy and entropy data from Table A-5,

$$g_f = h_f - Ts_f = (561.43\,\text{kJ/kg}) - (406.7\,\text{K})(1.6717\,\text{kJ/kg}\cdot\text{K}) = -118.5\,\text{kJ/kg}$$
$$g_g = h_g - Ts_g = (2724.9\,\text{kJ/kg}) - (406.7\,\text{K})(6.9917\,\text{kJ/kg}\cdot\text{K}) = -118.6\,\text{kJ/kg}$$

which are practically same. Therefore, the criterion for phase equilibrium is satisfied.

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16-70 The values of the Gibbs function for saturated refrigerant-134a at 280 kPa are to be calculated.

Analysis The saturation temperature of R-134a at 280 kPa is −1.25°C (Table A-12). Obtaining other properties from Table A-12, the Gibbs function for the liquid phase is,

$$g_f = h_f - Ts_f = 50.18\,\text{kJ/kg} - (-1.25 + 273.15\,\text{K})(0.19829\,\text{kJ/kg}\cdot\text{K}) = \mathbf{-3.74\,kJ/kg}$$

R-134a
280 kPa
$x = 0.7$

For the vapor phase,

$$g_g = h_g - Ts_g = 249.72\,\text{kJ/kg} - (-1.25 + 273.15\,\text{K})(0.93210\,\text{kJ/kg}\cdot\text{K}) = \mathbf{-3.72\,kJ/kg}$$

The results agree and demonstrate that phase equilibrium exists.

16-71E The values of the Gibbs function for saturated steam at 300°F as a saturated liquid, saturated vapor, and a mixture of liquid and vapor are to be calculated.

Analysis Obtaining properties from Table A-4E, the Gibbs function for the liquid phase is,

$$g_f = h_f - Ts_f = 269.73\,\text{Btu/lbm} - (759.67\,\text{R})(0.43720\,\text{Btu/lbm}\cdot\text{R}) = \mathbf{-62.40\,Btu/lbm}$$

Steam
300°F

For the vapor phase,

$$g_g = h_g - Ts_g = 1180.0\,\text{Btu/lbm} - (759.67\,\text{R})(1.6354\,\text{Btu/lbm}\cdot\text{R}) = \mathbf{-62.36\,Btu/lbm}$$

For the saturated mixture with a quality of 60%,

$$h = h_f + xh_{fg} = 269.73\,\text{Btu/lbm} + (0.60)(910.24\,\text{Btu/lbm}) = 815.87\,\text{Btu/lbm}$$
$$s = s_f + xs_{fg} = 0.43720\,\text{Btu/lbm}\cdot\text{R} + (0.60)(1.19818\,\text{Btu/lbm}\cdot\text{R}) = 1.1561\,\text{Btu/lbm}\cdot\text{R}$$
$$g = h - Ts = 815.87\,\text{Btu/lbm} - (759.67\,\text{R})(1.1561\,\text{Btu/lbm}\cdot\text{R}) = \mathbf{-62.38\,Btu/lbm}$$

The results agree and demonstrate that phase equilibrium exists.

16-72 A liquid-vapor mixture of ammonia and water in equilibrium at a specified temperature is considered. The pressure of ammonia is to be determined for two compositions of the liquid phase.

Assumptions The mixture is ideal and thus Raoult's law is applicable.

Analysis According to Raoults's law, when the mole fraction of the ammonia liquid is 20%,

$$P_{\text{NH3}} = y_{f,\text{NH3}} P_{\text{sat,NH3}}(T) = 0.20(615.3\,\text{kPa}) = \mathbf{123.1\,kPa}$$

$H_2O + NH_3$
10°C

When the mole fraction of the ammonia liquid is 80%,

$$P_{\text{NH3}} = y_{f,\text{NH3}} P_{\text{sat,NH3}}(T) = 0.80(615.3\,\text{kPa}) = \mathbf{492.2\,kPa}$$

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16-73 Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture, the composition of each phase at a specified temperature and pressure is to be determined.

Analysis From the equilibrium diagram (Fig. 16-21) we read

Liquid: $65\%\ N_2$ and $35\%\ O_2$

Vapor: $90\%\ N_2$ and $10\%\ O_2$

16-74 Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture at a specified pressure, the temperature is to be determined for a specified composition of the vapor phase.

Analysis From the equilibrium diagram (Fig. 16-21) we read $T = \mathbf{82\ K}$.

16-75 Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture at a specified pressure, the temperature is to be determined for a specified composition of the nitrogen.

Properties The molar masses of O_2 is 32 kg/kmol and that of N_2 is 28 kg/kmol (Table A-1).

Analysis For 100 kg of liquid phase, the mole numbers are

$$N_{f,\mathrm{O2}} = \frac{m_{f,\mathrm{O2}}}{M_{\mathrm{O2}}} = \frac{40\ \mathrm{kg}}{32\ \mathrm{kg/kmol}} = 1.25\ \mathrm{kmol}$$

$$N_{f,\mathrm{N2}} = \frac{m_{f,\mathrm{N2}}}{M_{\mathrm{N2}}} = \frac{60\ \mathrm{kg}}{28\ \mathrm{kg/kmol}} = 2.143\ \mathrm{kmol}$$

$$N_{f,\mathrm{total}} = 1.25 + 2.143 = 3.393\ \mathrm{kmol}$$

The mole fractions in the liquid phase are

$$y_{f,\mathrm{O2}} = \frac{N_{f,\mathrm{O2}}}{N_{f,\mathrm{total}}} = \frac{1.25\ \mathrm{kmol}}{3.393\ \mathrm{kmol}} = 0.3684$$

$$y_{f,\mathrm{N2}} = \frac{N_{f,\mathrm{N2}}}{N_{f,\mathrm{total}}} = \frac{2.143\ \mathrm{kmol}}{3.393\ \mathrm{kmol}} = 0.6316$$

From the equilibrium diagram (Fig. 16-21) we read $T = \mathbf{80.5\ K}$.

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16-76 Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture at a specified pressure, the mass of the oxygen in the liquid and gaseous phases is to be determined for a specified composition of the mixture.

Properties The molar masses of O_2 is 32 kg/kmol and that of N_2 is 28 kg/kmol (Table A-1).

Analysis From the equilibrium diagram (Fig. 16-21) at $T = 84$ K, the oxygen mole fraction in the vapor phase is 34% and that in the liquid phase is 70%. That is,

$$y_{f,\mathrm{O2}} = 0.70 \quad \text{and} \quad y_{g,\mathrm{O2}} = 0.34$$

The mole numbers are

$$N_{\mathrm{O2}} = \frac{m_{\mathrm{O2}}}{M_{\mathrm{O2}}} = \frac{30\ \mathrm{kg}}{32\ \mathrm{kg/kmol}} = 0.9375\ \mathrm{kmol}$$

$$N_{\mathrm{N2}} = \frac{m_{\mathrm{N2}}}{M_{\mathrm{N2}}} = \frac{40\ \mathrm{kg}}{28\ \mathrm{kg/kmol}} = 1.429\ \mathrm{kmol}$$

$$N_{\mathrm{total}} = 0.9375 + 1.429 = 2.366\ \mathrm{kmol}$$

The total number of moles in this system is

$$N_f + N_g = 2.366 \qquad (1)$$

The total number of moles of oxygen in this system is

$$0.7N_f + 0.34N_g = 0.9375 \qquad (2)$$

Solving equations (1) and (2) simultneously, we obtain

$$N_f = 0.3696$$

$$N_g = 1.996$$

Then, the mass of oxygen in the liquid and vapor phases is

$$m_{f,\mathrm{O2}} = y_{f,\mathrm{O2}} N_f M_{\mathrm{O2}} = (0.7)(0.3696\ \mathrm{kmol})(32\ \mathrm{kg/kmol}) = \mathbf{8.28\ kg}$$

$$m_{g,\mathrm{O2}} = y_{g,\mathrm{O2}} N_g M_{\mathrm{O2}} = (0.34)(1.996\ \mathrm{kmol})(32\ \mathrm{kg/kmol}) = \mathbf{21.72\ kg}$$

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16-77 Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture at a specified pressure, the total mass of the liquid phase is to be determined.

Properties The molar masses of O_2 is 32 kg/kmol and that of N_2 is 28 kg/kmol (Table A-1).

Analysis From the equilibrium diagram (Fig. 16-21) at $T = 84$ K, the oxygen mole fraction in the vapor phase is 34% and that in the liquid phase is 70%. That is,

$$y_{f,\mathrm{O2}} = 0.70 \quad \text{and} \quad y_{g,\mathrm{O2}} = 0.34$$

Also,

$$y_{f,\mathrm{N2}} = 0.30 \quad \text{and} \quad y_{g,\mathrm{N2}} = 0.66$$

The mole numbers are

$$N_{\mathrm{O2}} = \frac{m_{\mathrm{O2}}}{M_{\mathrm{O2}}} = \frac{30\ \mathrm{kg}}{32\ \mathrm{kg/kmol}} = 0.9375\ \mathrm{kmol}$$

$$N_{\mathrm{N2}} = \frac{m_{\mathrm{N2}}}{M_{\mathrm{N2}}} = \frac{40\ \mathrm{kg}}{28\ \mathrm{kg/kmol}} = 1.429\ \mathrm{kmol}$$

$$N_{\mathrm{total}} = 0.9375 + 1.429 = 2.366\ \mathrm{kmol}$$

The total number of moles in this system is

$$N_f + N_g = 2.366 \qquad (1)$$

The total number of moles of oxygen in this system is

$$0.7N_f + 0.34N_g = 0.9375 \qquad (2)$$

Solving equations (1) and (2) simultneously, we obtain

$$N_f = 0.3696$$

$$N_g = 1.996$$

The total mass of liquid in the mixture is then

$$\begin{aligned} m_{f,\mathrm{total}} &= m_{f,\mathrm{O2}} + m_{f,\mathrm{O2}} \\ &= y_{f,\mathrm{O2}} N_f M_{\mathrm{O2}} + y_{f,\mathrm{N2}} N_f M_{\mathrm{N2}} \\ &= (0.7)(0.3696\ \mathrm{kmol})(32\ \mathrm{kg/kmol}) + (0.3)(0.3696\ \mathrm{kmol})(28\ \mathrm{kg/kmol}) \\ &= \mathbf{11.38\ kg} \end{aligned}$$

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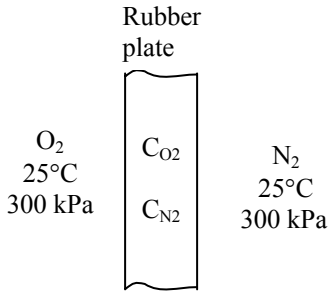
16-78 A rubber wall separates O_2 and N_2 gases. The molar concentrations of O_2 and N_2 in the wall are to be determined.

Assumptions The O_2 and N_2 gases are in phase equilibrium with the rubber wall.

Properties The molar mass of oxygen and nitrogen are 32.0 and 28.0 kg/kmol, respectively (Table A-1). The solubility of oxygen and nitrogen in rubber at 298 K are 0.00312 and 0.00156 kmol/m^3·bar, respectively (Table 16-3).

Analysis Noting that 300 kPa = 3 bar, the molar densities of oxygen and nitrogen in the rubber wall are determined to be

$$C_{O_2,\,\text{solid side}}(0) = S \times P_{O_2,\,\text{gas side}}$$
$$= (0.00312\ \text{kmol/m}^3.\text{bar})(3\ \text{bar})$$
$$= \mathbf{0.00936\ kmol/m^3}$$
$$C_{N_2,\,\text{solid side}}(0) = S \times P_{N_2,\,\text{gas side}}$$
$$= (0.00156\ \text{kmol/m}^3.\text{bar})(3\ \text{bar})$$
$$= \mathbf{0.00468\ kmol/m^3}$$



That is, there will be 0.00936 kmol of O_2 and 0.00468 kmol of N_2 gas in each m^3 volume of the rubber wall.

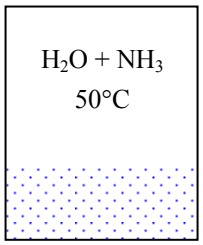
16-79 A liquid-vapor mixture of ammonia and water in equilibrium at a specified temperature is considered. The composition of the vapor phase is given. The composition of the liquid phase is to be determined.

Assumptions The mixture is ideal and thus Raoult's law is applicable.

Properties At 50°C, $P_{\text{sat},H_2O} = 12.352\ \text{kPa}$ and $P_{\text{sat},NH_3} = 2033.5\ \text{kPa}$.

Analysis We have $y_{g,H_2O} = 1\%$ and $y_{g,NH_3} = 99\%$. For an ideal two-phase mixture we have

$$y_{g,H_2O} P_m = y_{f,H_2O} P_{\text{sat},H_2O}(T)$$
$$y_{g,NH_3} P_m = y_{f,NH_3} P_{\text{sat},NH_3}(T)$$
$$y_{f,H_2O} + y_{f,NH_3} = 1$$



Solving for $y_{f,H_2O,}$

$$y_{f,H_2O} = \frac{y_{g,H_2O} P_{\text{sat},NH_3}}{y_{g,NH_3} P_{\text{sat},H_2O}}(1 - y_{f,H_2O}) = \frac{(0.01)(2033.5\ \text{kPa})}{(0.99)(12.352\ \text{kPa})}(1 - y_{f,H_2O})$$

It yields

$$y_{f,H_2O} = \mathbf{0.624} \quad \text{and} \quad y_{f,NH_3} = \mathbf{0.376}$$

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16-80 A mixture of water and ammonia is considered. The mole fractions of the ammonia in the liquid and vapor phases are to be determined.

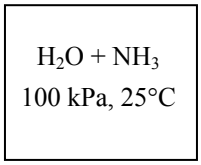
Assumptions The mixture is ideal and thus Raoult's law is applicable.

Properties At 25°C, $P_{\text{sat,H2O}} = 3.1698$ kPa and $P_{\text{sat,NH3}} = 1003.5$ kPa (Tables A-4).

Analysis According to Raoults's law, the partial pressures of ammonia and water in the vapor phase are given by

$$P_{g,\text{NH3}} = y_{f,\text{NH3}} P_{\text{sat,NH3}} = \frac{N_{f,\text{NH3}}}{N_{f,\text{NH3}} + N_{f,\text{H2O}}}(1003.5\,\text{kPa})$$

$$P_{g,\text{H2O}} = y_{f,\text{H2O}} P_{\text{sat,H2O}} = \frac{N_{f,\text{H2O}}}{N_{f,\text{H2O}} + N_{f,\text{H2O}}}(3.1698\,\text{kPa})$$



The sum of these two partial pressures must equal the total pressure of the vapor mixture. In terms of $x = \frac{N_{f,\text{H2O}}}{N_{f,\text{NH3}}}$, this sum is

$$\frac{1003.5}{x+1} + \frac{3.1698x}{x+1} = 100$$

Solving this expression for x gives

$$x = 9.331 \text{ kmol H2O/kmol NH3}$$

In the vapor phase, the partial pressure of the ammonia vapor is

$$P_{g,\text{NH3}} = \frac{1003.5}{x+1} = \frac{1003.5}{9.331+1} = 97.13\,\text{kPa}$$

The mole fraction of ammonia in the vapor phase is then

$$y_{g,\text{NH3}} = \frac{P_{g,\text{NH3}}}{P} = \frac{97.13\,\text{kPa}}{100\,\text{kPa}} = \mathbf{0.9713}$$

According to Raoult's law,

$$y_{f,\text{NH3}} = \frac{P_{g,\text{NH3}}}{P_{\text{sat,NH3}}} = \frac{97.13\,\text{kPa}}{1003.5\,\text{kPa}} = \mathbf{0.0968}$$

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16-81 An ammonia-water absorption refrigeration unit is considered. The operating pressures in the generator and absorber, and the mole fractions of the ammonia in the strong liquid mixture being pumped from the absorber and the weak liquid solution being drained from the generator are to be determined.

Assumptions The mixture is ideal and thus Raoult's law is applicable.

Properties At 0°C, $P_{\text{sat,H2O}} = 0.6112$ kPa and at 46°C, $P_{\text{sat,H2O}} = 10.10$ kPa (Table A-4). The saturation pressures of ammonia at the same temperatures are given to be 430.6 kPa and 1830.2 kPa, respectively.

Analysis According to Raoults's law, the partial pressures of ammonia and water are given by

$$P_{\text{g,NH3}} = y_{f,\text{NH3}} P_{\text{sat,NH3}}$$
$$P_{\text{g,H2O}} = y_{f,\text{H2O}} P_{\text{sat,H2O}} = (1 - y_{f,\text{NH3}}) P_{\text{sat,H2O}}$$

Using Dalton's partial pressure model for ideal gas mixtures, the mole fraction of the ammonia in the vapor mixture is

$$y_{g,\text{NH3}} = \frac{y_{f,\text{NH3}} P_{\text{sat,NH3}}}{y_{f,\text{NH3}} P_{\text{sat,NH3}} + (1 - y_{f,\text{NH3}} P_{\text{sat,H2O}})}$$
$$0.96 = \frac{430.6 y_{f,\text{NH3}}}{430.6 y_{f,\text{NH3}} + 0.6112(1 - y_{f,\text{NH3}})} \longrightarrow y_{f,\text{NH3}} = \mathbf{0.03294}$$

Then,

$$P = y_{f,\text{NH3}} P_{\text{sat,NH3}} + (1 - y_{f,\text{NH3}}) P_{\text{sat,H2O}}$$
$$= (0.03294)(430.6) + (1 - 0.03294)(0.6112) = \mathbf{14.78\ kPa}$$

Performing the similar calculations for the regenerator,

$$0.96 = \frac{1830.2 y_{f,\text{NH3}}}{1830.2 y_{f,\text{NH3}} + 10.10(1 - y_{f,\text{NH3}})} \longrightarrow y_{f,\text{NH3}} = \mathbf{0.1170}$$

$$P = (0.1170)(1830.2) + (1 - 0.1170)(10.10) = \mathbf{223.1 kPa}$$

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16-82 An ammonia-water absorption refrigeration unit is considered. The operating pressures in the generator and absorber, and the mole fractions of the ammonia in the strong liquid mixture being pumped from the absorber and the weak liquid solution being drained from the generator are to be determined.

Assumptions The mixture is ideal and thus Raoult's law is applicable.

Properties At 6°C, $P_{\text{sat,H2O}} = 0.9353$ kPa and at 40°C, $P_{\text{sat,H2O}} = 7.3851$ kPa (Table A-4 or EES). The saturation pressures of ammonia at the same temperatures are given to be 534.8 kPa and 1556.7 kPa, respectively.

Analysis According to Raoults's law, the partial pressures of ammonia and water are given by

$$P_{\text{g,NH3}} = y_{f,\text{NH3}} P_{\text{sat,NH3}}$$
$$P_{\text{g,H2O}} = y_{f,\text{H2O}} P_{\text{sat,H2O}} = (1 - y_{f,\text{NH3}}) P_{\text{sat,H2O}}$$

Using Dalton's partial pressure model for ideal gas mixtures, the mole fraction of the ammonia in the vapor mixture is

$$y_{g,\text{NH3}} = \frac{y_{f,\text{NH3}} P_{\text{sat,NH3}}}{y_{f,\text{NH3}} P_{\text{sat,NH3}} + (1 - y_{f,\text{NH3}} P_{\text{sat,H2O}})}$$

$$0.96 = \frac{534.8 y_{f,\text{NH3}}}{534.8 y_{f,\text{NH3}} + 0.9353(1 - y_{f,\text{NH3}})} \longrightarrow y_{f,\text{NH3}} = \mathbf{0.04028}$$

Then,

$$P = y_{f,\text{NH3}} P_{\text{sat,NH3}} + (1 - y_{f,\text{NH3}}) P_{\text{sat,H2O}}$$
$$= (0.04028)(534.8) + (1 - 0.04028)(0.9353) = \mathbf{22.44\ kPa}$$

Performing the similar calculations for the regenerator,

$$0.96 = \frac{1556.7 y_{f,\text{NH3}}}{1556.7 y_{f,\text{NH3}} + 7.3851(1 - y_{f,\text{NH3}})} \longrightarrow y_{f,\text{NH3}} = \mathbf{0.1022}$$

$$P = (0.1022)(1556.7) + (1 - 0.1022)(7.3851) = \mathbf{165.7\ kPa}$$

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16-83 A liquid mixture of water and R-134a is considered. The mole fraction of the water and R-134a vapor are to be determined.

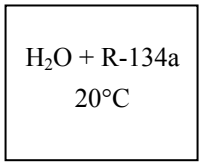
Assumptions The mixture is ideal and thus Raoult's law is applicable.

Properties At 20°C, $P_{\text{sat,H2O}} = 2.3392$ kPa and $P_{\text{sat,R}} = 572.07$ kPa (Tables A-4, A-11). The molar masses of water and R-134a are 18.015 and 102.03 kg/kmol, respectively (Table A-1).

Analysis The mole fraction of the water in the liquid mixture is

$$y_{f,\text{H2O}} = \frac{N_{f,\text{H2O}}}{N_{\text{total}}} = \frac{\text{mf}_{f,\text{H2O}} / M_{\text{H2O}}}{(\text{mf}_{f,\text{H2O}} / M_{\text{H2O}}) + (\text{mf}_{f,\text{R}} / M_{\text{R}})}$$

$$= \frac{0.9/18.015}{(0.9/18.015) + (0.1/102.03)} = 0.9808$$



According to Raoults's law, the partial pressures of R-134a and water in the vapor mixture are

$$P_{g,\text{R}} = y_{f,\text{R}} P_{\text{sat,R}} = (1 - 0.9808)(572.07\,\text{kPa}) = 10.98\,\text{kPa}$$

$$P_{g,\text{H2O}} = y_{f,\text{H2O}} P_{\text{sat,H2O}} = (0.9808)(2.3392\,\text{kPa}) = 2.294\,\text{kPa}$$

The total pressure of the vapor mixture is then

$$P_{\text{total}} = P_{g,\text{R}} + P_{g,\text{H2O}} = 10.98 + 2.294 = 13.274\,\text{kPa}$$

Based on Dalton's partial pressure model for ideal gases, the mole fractions in the vapor phase are

$$y_{g,\text{H2O}} = \frac{P_{g,\text{H2O}}}{P_{\text{total}}} = \frac{2.294\,\text{kPa}}{13.274\,\text{kPa}} = \mathbf{0.1728}$$

$$y_{g,\text{R}} = \frac{P_{g,\text{R}}}{P_{\text{total}}} = \frac{10.98\,\text{kPa}}{13.274\,\text{kPa}} = \mathbf{0.8272}$$

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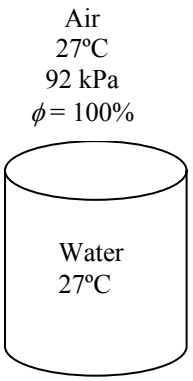
16-84 A glass of water is left in a room. The mole fraction of the water vapor in the air and the mole fraction of air in the water are to be determined when the water and the air are in thermal and phase equilibrium.

Assumptions **1** Both the air and water vapor are ideal gases. **2** Air is saturated since the humidity is 100 percent. **3** Air is weakly soluble in water and thus Henry's law is applicable.

Properties The saturation pressure of water at 27°C is 3.568 kPa (Table A-4). Henry's constant for air dissolved in water at 27ºC (300 K) is given in Table 16-2 to be H = 74,000 bar. Molar masses of dry air and water are 29 and 18 kg/kmol, respectively (Table A-1).

Analysis (*a*) Noting that air is saturated, the partial pressure of water vapor in the air will simply be the saturation pressure of water at 27°C,

$$P_{\text{vapor}} = P_{\text{sat @ 27°C}} = 3.568 \text{ kPa} \quad \text{(Table A-4)}$$



Assuming both the air and vapor to be ideal gases, the mole fraction of water vapor in the air is determined to be

$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{3.568 \text{ kPa}}{92 \text{ kPa}} = \mathbf{0.0388}$$

(*b*) Noting that the total pressure is 92 kPa, the partial pressure of dry air is

$$P_{\text{dry air}} = P - P_{\text{vapor}} = 92 - 3.568 = 88.43 \text{ kPa} = 0.8843 \text{ bar}$$

From Henry's law, the mole fraction of air in the water is determined to be

$$y_{\text{dry air,liquid side}} = \frac{P_{\text{dry air,gas side}}}{H} = \frac{0.8843 \text{ bar}}{74{,}000 \text{ bar}} = \mathbf{1.20 \times 10^{-5}}$$

Discussion The amount of air dissolved in water is very small, as expected.

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16-85 A carbonated drink in a bottle is considered. Assuming the gas space above the liquid consists of a saturated mixture of CO_2 and water vapor and treating the drink as a water, determine the mole fraction of the water vapor in the CO_2 gas and the mass of dissolved CO_2 in a 300 ml drink are to be determined when the water and the CO_2 gas are in thermal and phase equilibrium.

Assumptions **1** The liquid drink can be treated as water. **2** Both the CO_2 and the water vapor are ideal gases. **3** The CO_2 gas and water vapor in the bottle from a saturated mixture. **4** The CO_2 is weakly soluble in water and thus Henry's law is applicable.

Properties The saturation pressure of water at 27°C is 3.568 kPa (Table A-4). Henry's constant for CO_2 dissolved in water at 27ºC (300 K) is given in Table 16-2 to be $H = 1710$ bar. Molar masses of CO_2 and water are 44 and 18 kg/kmol, respectively (Table A-1).

Analysis (*a*) Noting that the CO_2 gas in the bottle is saturated, the partial pressure of water vapor in the air will simply be the saturation pressure of water at 27°C,

$$P_{\text{vapor}} = P_{\text{sat @ 27°C}} = 3.568 \text{ kPa}$$ (more accurate EES value compared to interpolation value from Table A-4)

Assuming both CO_2 and vapor to be ideal gases, the mole fraction of water vapor in the CO_2 gas becomes

$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{3.568 \text{ kPa}}{130 \text{ kPa}} = \mathbf{0.0274}$$

(*b*) Noting that the total pressure is 130 kPa, the partial pressure of CO_2 is

$$P_{\text{CO}_2 \text{ gas}} = P - P_{\text{vapor}} = 130 - 3.568 = 126.4 \text{ kPa} = 1.264 \text{ bar}$$

From Henry's law, the mole fraction of CO_2 in the drink is determined to be

$$y_{\text{CO}_2,\text{liquid side}} = \frac{P_{\text{CO}_2,\text{gas side}}}{H} = \frac{1.264 \text{ bar}}{1710 \text{ bar}} = \mathbf{7.39 \times 10^{-4}}$$

Then the mole fraction of water in the drink becomes

$$y_{\text{water, liquid side}} = 1 - y_{\text{CO}_2\text{, liquid side}} = 1 - 7.39 \times 10^{-4} = 0.9993$$

The mass and mole fractions of a mixture are related to each other by

$$\text{mf}_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \frac{M_i}{M_m}$$

where the apparent molar mass of the drink (liquid water - CO_2 mixture) is

$$M_m = \sum y_i M_i = y_{\text{liquid water}} M_{\text{water}} + y_{\text{CO}_2} M_{\text{CO}_2} = 0.9993 \times 18.0 + (7.39 \times 10^{-4}) \times 44 = 18.02 \text{ kg / kmol}$$

Then the mass fraction of dissolved CO_2 gas in liquid water becomes

$$\text{mf}_{\text{CO}_2,\text{liquid side}} = y_{\text{CO}_2,\text{liquid side}} (0) \frac{M_{\text{CO}_2}}{M_m} = 7.39 \times 10^{-4} \frac{44}{18.02} = 0.00180$$

Therefore, the mass of dissolved CO_2 in a 300 ml ≈ 300 g drink is

$$m_{\text{CO}_2} = \text{mf}_{\text{CO}_2} m_m = (0.00180)(300 \text{ g}) = \mathbf{0.54 \text{ g}}$$

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Review Problems

16-86 The equilibrium constant of the dissociation process $O_2 \leftrightarrow 2O$ is given in Table A-28 at different temperatures. The value at a given temperature is to be verified using Gibbs function data.

Analysis The K_P value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$O_2 \leftrightarrow 2O$
2000 K

$$\begin{aligned}\Delta G^*(T) &= \nu_O \bar{g}^*_O(T) - \nu_{O_2} \bar{g}^*_{O_2}(T) \\ &= \nu_O(\bar{h} - T\bar{s})_O - \nu_{O_2}(\bar{h} - T\bar{s})_{O_2} \\ &= \nu_O[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_O - \nu_{O_2}[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{O_2} \\ &= 2\times(249{,}190 + 42{,}564 - 6852 - 2000\times 201.135) \\ &\quad -1\times(0 + 67{,}881 - 8682 - 2000\times 268.655) \\ &= 243{,}375 \text{ kJ/kmol}\end{aligned}$$

Substituting,

$$\ln K_p = -(243{,}375 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(2000 \text{ K})] = -14.636$$

or

$$K_p = \mathbf{4.4\times 10^{-7}} \qquad \text{(Table A-28: } \ln K_P = \text{-14.622)}$$

16-87 A mixture of H_2 and Ar is heated is heated until 10% of H_2 is dissociated. The final temperature of mixture is to be determined.

Assumptions **1** The constituents of the mixture are ideal gases. **2** Ar in the mixture remains an inert gas.

Analysis The stoichiometric and actual reactions can be written as

Stoichiometric: $H_2 \Leftrightarrow 2H$ (thus $\nu_{H_2} = 1$ and $\nu_H = 2$)

Actual: $H_2 + Ar \longrightarrow \underbrace{0.2H}_{\text{prod}} + \underbrace{0.90H_2}_{\text{react.}} + \underbrace{Ar}_{\text{inert}}$

$H_2 \Leftrightarrow 2H$
Ar
1 atm

The equilibrium constant K_P can be determined from

$$K_p = \frac{N_H^{\nu_H}}{N_{H_2}^{\nu_{H_2}}}\left(\frac{P}{N_{\text{total}}}\right)^{\nu_H - \nu_{H_2}} = \frac{0.2^2}{0.9}\left(\frac{1}{0.9 + 0.2 + 1}\right)^{2-1} = 0.02116$$

and

$$\ln K_p = -3.855$$

From Table A-28, the temperature corresponding to this K_P value is $T =$ **2974 K**.

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16-88 The equilibrium constant for the reaction $CH_4 + 2O_2 \Leftrightarrow CO_2 + 2H_2O$ at 100 kPa and 2000 K is to be determined.

***Assumptions* 1** The constituents of the mixture are ideal gases.

Analysis This is a simultaneous reaction. We can begin with the dissociation of methane and carbon dioxide,

$$CH_4 \Leftrightarrow C + 2H_2 \qquad K_P = e^{-9.685}$$

$$C + O_2 \Leftrightarrow CO_2 \qquad K_P = e^{15.869}$$

$CH_4+2O_2 \Leftrightarrow$
CO_2+2H_2O
3000 K
690 kPa

When these two reactions are summed and the common carbon term cancelled, the result is

$$CH_4 + O_2 \Leftrightarrow CO_2 + 2H_2 \qquad K_P = e^{(15.869-9.685)} = e^{6.184}$$

Next, we include the water dissociation reaction (Table A-28),

$$2H_2 + O_2 \Leftrightarrow 2H_2O \qquad K_P = e^{2(3.086)} = e^{6.172}$$

which when summed with the previous reaction and the common hydrogen term is cancelled yields

$$CH_4 + 2O_2 \Leftrightarrow CO_2 + 2H_2O \qquad K_P = e^{6.184+6.172} = e^{12.356}$$

Then,

$$\ln K_P = \mathbf{12.356}$$

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16-89 A mixture of H_2O, O_2, and N_2 is heated to a high temperature at a constant pressure. The equilibrium composition is to be determined.

Assumptions **1** The equilibrium composition consists of H_2O, O_2, N_2 and H_2. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

1 H_2O
2 O_2
5 N_2
2200 K
5 atm

Stoichiometric: $H_2O \Leftrightarrow H_2 + \frac{1}{2}O_2$ (thus $\nu_{H_2O} = 1, \nu_{H_2} = 1,$ and $\nu_{O_2} = \frac{1}{2}$)

Actual: $H_2O + 2O_2 + 5N_2 \longrightarrow \underbrace{xH_2O}_{\text{react.}} + \underbrace{yH_2 + zO_2}_{\text{products}} + \underbrace{5N_2}_{\text{inert}}$

H balance: $2 = 2x + 2y \longrightarrow y = 1 - x$

O balance: $5 = x + 2z \longrightarrow z = 2.5 - 0.5x$

Total number of moles: $N_{\text{total}} = x + y + z + 5 = 8.5 - 0.5x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2} - \nu_{O_2} - \nu_{H_2O})} = \frac{y \quad z^{0.5}}{x} \left(\frac{P}{N_{\text{total}}}\right)^{1+0.5-1}$$

From Table A-28, $\ln K_P = -6.768$ at 2200 K. Thus $K_P = 0.00115$. Substituting,

$$0.00115 = \frac{(1-x)(1.5-0.5x)^{0.5}}{x} \left(\frac{5}{8.5-0.5x}\right)^{0.5}$$

Solving for x,

$x = 0.9981$

Then,

$y = 1 - x = 0.0019$

$z = 2.5 - 0.5x = 2.00095$

Therefore, the equilibrium composition of the mixture at 2200 K and 5 atm is

$$\mathbf{0.9981H_2O + 0.0019H_2 + 2.00095O_2 + 5N_2}$$

The equilibrium constant for the reaction $H_2O \Leftrightarrow OH + \frac{1}{2}H_2$ is $\ln K_P = -7.148$, which is very close to the K_P value of the reaction considered. Therefore, it is not realistic to assume that no OH will be present in equilibrium mixture.

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16-90 Methane gas is burned with stoichiometric amount of air during a combustion process. The equilibrium composition and the exit temperature are to be determined.

Assumptions **1** The product gases consist of CO_2, H_2O, CO, N_2, and O_2. **2** The constituents of the mixture are ideal gases. **3** This is an adiabatic and steady-flow combustion process.

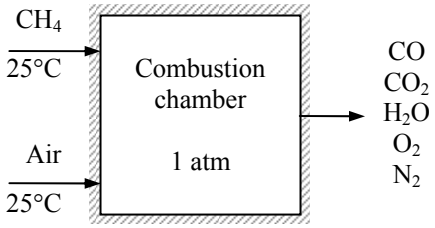
Analysis (*a*) The combustion equation of CH_4 with stoichiometric amount of O_2 can be written as

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow xCO_2 + (1-x)CO + (0.5-0.5x)O_2 + 2H_2O + 7.52N_2$$

After combustion, there will be no CH_4 present in the combustion chamber, and H_2O will act like an inert gas. The equilibrium equation among CO_2, CO, and O_2 can be expressed as

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2}O_2 \quad (\text{thus } \nu_{CO_2} = 1,\ \nu_{CO} = 1,\ \text{and } \nu_{O_2} = \tfrac{1}{2})$$

and $$K_p = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{CO}+\nu_{O_2}-\nu_{CO_2})}$$



where $N_{\text{total}} = x + (1-x) + (1.5-0.5x) + 2 + 7.52 = 12.02 - 0.5x$

Substituting,

$$K_p = \frac{(1-x)(0.5-0.5x)^{0.5}}{x}\left(\frac{1}{12.02-0.5x}\right)^{1.5-1}$$

The value of K_P depends on temperature of the products, which is yet to be determined. A second relation to determine K_P and x is obtained from the steady-flow energy balance expressed as

$$0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow 0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \bar{h}_{f\,R}^\circ$$

since the combustion is adiabatic and the reactants enter the combustion chamber at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h\,(T)$. From the tables,

Substance	$\bar{h}_f^\circ$**, kJ/kmol**	$\bar{h}_{298\,K}$**, kJ/kmol**
$CH_4(g)$	-74,850	--
N_2	0	8669
O_2	0	8682
$H_2O(g)$	-241,820	9904
CO	-110,530	8669
CO_2	-393,520	9364

Substituting,

$$\begin{aligned} 0 = {} & x(-393{,}520 + \bar{h}_{CO_2} - 9364) + (1-x)(-110{,}530 + \bar{h}_{CO} - 8669) \\ & + 2(-241{,}820 + \bar{h}_{H_2O} - 9904) + (0.5-0.5x)(0 + \bar{h}_{O_2} - 8682) \\ & + 7.52(0 + \bar{h}_{N_2} - 8669) - 1(-74{,}850 + h_{298} - h_{298}) - 0 - 0 \end{aligned}$$

which yields

$$x\bar{h}_{CO_2} + (1-x)\bar{h}_{CO} + 2\bar{h}_{H_2O} + (0.5-0.5x)\bar{h}_{O_2} + 7.52\bar{h}_{N_2} - 279{,}344x = 617{,}329$$

Now we have two equations with two unknowns, T_P and x. The solution is obtained by trial and error by assuming a temperature T_P, calculating the equilibrium composition from the first equation, and then checking to see if the second equation is satisfied. A first guess is obtained by assuming there is no CO in the products, i.e., x = 1. It yields $T_P = 2328$ K. The adiabatic combustion temperature with incomplete combustion will be less.

Take $T_p = 2300$ K $\longrightarrow$ $\ln K_p = -4.49$ $\longrightarrow$ $x = 0.870$ $\longrightarrow$ $RHS = 641{,}093$

Take $T_p = 2250$ K $\longrightarrow$ $\ln K_p = -4.805$ $\longrightarrow$ $x = 0.893$ $\longrightarrow$ $RHS = 612{,}755$

By interpolation, $T_p =$ **2258 K** and $x = 0.889$

Thus the composition of the equilibrium mixture is

$$\mathbf{0.889CO_2 + 0.111CO + 0.0555O_2 + 2H_2O + 7.52N_2}$$

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16-91 Problem 16-90 is reconsidered. The effect of excess air on the equilibrium composition and the exit temperature by varying the percent excess air from 0 to 200 percent is to be studied.

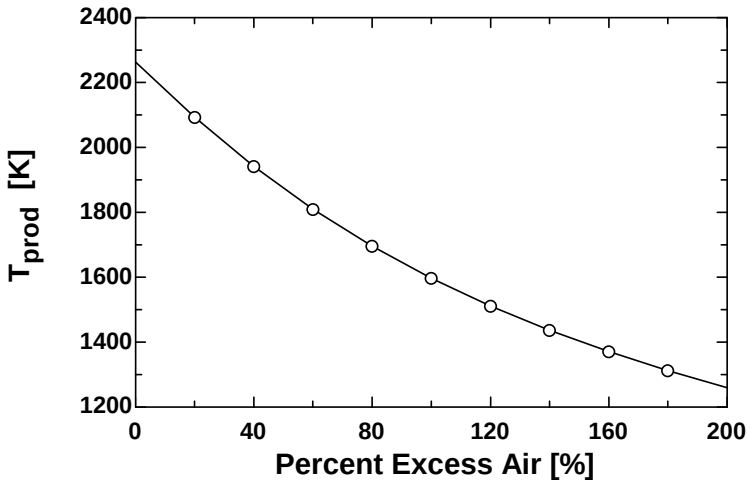
Analysis The problem is solved using EES, and the solution is given below.

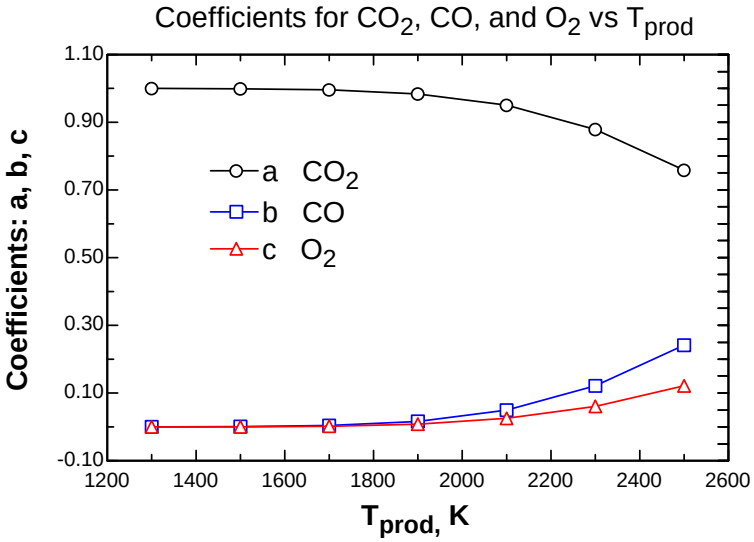
```
"Often, for nonlinear problems such as this one, good gusses are required to start the solution.  First, run the program with zero percent excess air to determine the net heat transfer as a function of T_prod.  Just press F3 or click on the Solve Table icon.  From Plot Window 1, where Q_net is plotted vs T_prod, determnine the value of T_prod for Q_net=0 by holding down the Shift key and move the cross hairs by moving the mouse.  Q_net is approximately zero at T_prod = 2269 K.  From Plot Window 2 at T_prod = 2269 K, a, b, and c are approximately 0.89, 0.10, and 0.056, respectively." "For EES to calculate a, b, c, and T_prod directly for the adiabatic case, remove the '{ }' in the last line of this window to set Q_net = 0.0.  Then from the Options menu select  Variable Info and set the Guess Values of a, b, c, and T_prod to the guess values selected from the Plot Windows. Then press F2 or click on the Calculator icon."

"Input Data"
{PercentEx = 0}
Ex = PercentEX/100
P_prod =101.3 [kPa]
R_u=8.314 [kJ/kmol-K]
T_fuel=298 [K]
T_air=298 [K]
"The combustion equation of CH4 with stoichiometric amount of air is
CH4 + (1+Ex)(2)(O2 + 3.76N2)=CO2 +2H2O+(1+Ex)(2)(3.76)N2"
"For the incomplete combustion process in this problem, the combustion equation is
CH4 + (1+Ex)(2)(O2 + 3.76N2)=aCO2 +bCO + cO2+2H2O+(1+Ex)(2)(3.76)N2"
"Specie balance equations"
"O"
4=a *2+b +c *2+2
"C"
1=a +b
N_tot =a +b +c +2+(1+Ex)*(2)*3.76   "Total kilomoles of products at equilibrium"
"We assume the equilibrium reaction is
CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component as a function of its temperature at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
 "The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2
"The equilibrium constant is given by Eq. 16-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 16-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(b^1*c^0.5)/(a^1)"
sqrt(P/N_tot )*b *sqrt(c )=K_P *a
"Conservation of energy for the reaction, assuming SSSF, neglecting work ,  ke, and pe:"
E_in - E_out = DELTAE_cv
E_in = Q_net + HR
"The enthalpy of the reactant gases is"
HR=enthalpy(CH4,T=T_fuel)+ (1+Ex)*(2) *enthalpy(O2,T=T_air)+(1+Ex)*(2)*3.76 *enthalpy(N2,T=T_air)
E_out = HP
"The enthalpy of the product gases is"
HP=a *enthalpy(CO2,T=T_prod )+b *enthalpy(CO,T=T_prod ) +2*enthalpy(H2O,T=T_prod
)+(1+Ex)*(2)*3.76*enthalpy(N2,T=T_prod ) + c *enthalpy(O2,T=T_prod )
DELTAE_cv = 0  "Steady-flow requirement"
Q_net=0 "For an adiabatic reaction the net heat added is zero."
```

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PercentEx	T_{prod} [K]
0	2260
20	2091
40	1940
60	1809
80	1695
100	1597
120	1511
140	1437
160	1370
180	1312
200	1259





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16-92 The equilibrium partial pressure of the carbon dioxide for the reaction $CH_4 + 2O_2 \Leftrightarrow CO_2 + 2H_2O$ at 700 kPa and 3000 K is to be determined.

Assumptions **1** The equilibrium composition consists of CH_4, O_2, CO_2, and H_2O. **2** The constituents of the mixture are ideal gases.

Analysis This is a simultaneous reaction. We can begin with the dissociation of methane and carbon dioxide,

$$CH_4 \Leftrightarrow C + 2H_2 \qquad K_P = e^{-9.685}$$

$$C + O_2 \Leftrightarrow CO_2 \qquad K_P = e^{15.869}$$

$CH_4+2O_2 \Leftrightarrow CO_2+2H_2O$
3000 K
700 kPa

When these two reactions are summed and the common carbon term cancelled, the result is

$$CH_4 + O_2 \Leftrightarrow CO_2 + 2H_2 \qquad K_P = e^{(15.869-9.685)} = e^{6.184}$$

Next, we include the water dissociation reaction,

$$2H_2 + O_2 \Leftrightarrow 2H_2O \qquad K_P = e^{2(3.086)} = e^{6.172}$$

which when summed with the previous reaction and the common hydrogen term is cancelled yields

$$CH_4 + 2O_2 \Leftrightarrow CO_2 + 2H_2O \qquad K_P = e^{6.184+6.172} = e^{12.356}$$

Then,

$$\ln K_P = 12.356$$

Actual reeaction: $CH_4 + 2O_2 \longrightarrow \underbrace{xCH_4 + yO_2}_{\text{react.}} + \underbrace{zCO_2 + mH_2O}_{\text{products}}$

C balance: $1 = x + z \longrightarrow z = 1 - x$

H balance: $4 = 4x + 2m \longrightarrow m = 2 - 2x$

O balance: $4 = 2y + 2z + m \longrightarrow y = 2x$

Total number of moles: $N_{\text{total}} = x + y + z + m = 3$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO2}^{\nu_{CO2}} N_{H2O}^{\nu_{H2O}}}{N_{CH4}^{\nu_{CH4}} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}}\right)^{\nu_{CO2}+\nu_{H2O}-\nu_{CH4}-\nu_{O2}}$$

Substituting,

$$e^{12.356} = \frac{(1-x)(2-2x)^2}{x(2x)^2} \left(\frac{700/101.325}{3}\right)^{1+2-1-2}$$

Solving for x,

$$x = 0.01601$$

Then,

$$y = 2x = 0.03202$$

$$z = 1 - x = 0.98399$$

$$m = 2 - 2x = 1.96798$$

Therefore, the equilibrium composition of the mixture at 3000 K and 700 kPa is

$$0.01601\,CH_4 + 0.03202\,O_2 + 0.98399\,CO_2 + 1.96798\,H_2O$$

The mole fraction of carbon dioxide is

$$y_{CO2} = \frac{0.98399}{3} = 0.3280$$

and the partial pressure of the carbon dioxide in the product mixture is

$$P_{CO2} = y_{CO2} P = (0.3280)(700\text{ kPa}) = \mathbf{230\ kPa}$$

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16-93 Methane is heated from a specified state to another state. The amount of heat required is to be determined without and with dissociation cases.

Properties The molar mass and gas constant of methane are 16.043 kg/kmol and 0.5182 kJ/kg·K (Table A-1).

Assumptions **1** The equilibrium composition consists of O_2 and O. **2** The constituents of the mixture are ideal gases.

Analysis (*a*) An energy balance for the process gives

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = N(\bar{u}_2 - \bar{u}_1)$$
$$= N\left[\bar{h}_2 - \bar{h}_1 - R_u(T_2 - T_1)\right]$$

CH_4
1000 K
1 atm

Using the empirical coefficients of Table A-2c,

$$\bar{h}_2 - \bar{h}_1 = \int_1^2 c_p\,dT = a(T_2 - T_1) + \frac{b}{2}(T_2^2 - T_1^2) + \frac{c}{3}(T_2^3 - T_1^3) + \frac{d}{4}(T_2^4 - T_1^4)$$

$$= 19.89(1000 - 298) + \frac{0.05024}{2}(1000^2 - 298^2) + \frac{1.269\times10^{-5}}{3}(1000^3 - 298^3)$$
$$+ \frac{-11.01\times10^{-9}}{4}(1000^4 - 298^4)$$
$$= 38{,}239 \text{ kJ/kmol}$$

Substituting,

$$Q_{\text{in}} = (10 \text{ kmol})\left[38{,}239 \text{ kJ/kmol} - (8.314 \text{ kJ/kmol}\cdot\text{K})(1000 - 298)\text{K}\right] = \mathbf{324{,}000 \text{ kJ}}$$

(*b*) The stoichiometric and actual reactions in this case are

Stoichiometric: $\text{CH}_4 \Leftrightarrow \text{C} + 2\text{H}_2$ (thus $\nu_{\text{CH4}} = 1$, $\nu_{\text{C}} = 1$ and $\nu_{\text{H2}} = 2$)

Actual: $\text{CH}_4 \longrightarrow \underbrace{x\text{CH}_4}_{\text{react.}} + \underbrace{y\text{C} + z\text{H}_2}_{\text{products}}$

C balance: $1 = x + y \longrightarrow y = 1 - x$

H balance: $4 = 4x + 2z \longrightarrow z = 2 - 2x$

Total number of moles: $N_{\text{total}} = x + y + z = 3 - 2x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{\text{C}}^{\nu_{\text{C}}} N_{\text{H2}}^{\nu_{\text{H2}}}}{N_{\text{CH4}}^{\nu_{\text{CH4}}}} \left(\frac{P}{N_{\text{total}}}\right)^{\nu_{\text{C}} + \nu_{\text{H2}} - \nu_{\text{CH4}}}$$

From the problem statement, at 1000 K, $\ln K_p = -2.328$. Then,

$$K_P = e^{-2.328} = 0.09749$$

Substituting,

$$0.09749 = \frac{(1-x)(2-2x)^2}{x}\left(\frac{1}{3-2x}\right)^{1+2-1}$$

Solving for x,

$$x = 0.6414$$

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Then,

$$y = 1 - x = 0.3586$$

$$z = 2 - 2x = 0.7172$$

Therefore, the equilibrium composition of the mixture at 1000 K and 1 atm is

$$0.6414\,\mathrm{CH_4} + 0.3586\,\mathrm{C} + 0.7172\,\mathrm{H_2}$$

The mole fractions are

$$y_{\mathrm{CH4}} = \frac{N_{\mathrm{CH4}}}{N_{\mathrm{total}}} = \frac{0.6414}{0.6414 + 0.3586 + 0.7172} = \frac{0.6414}{1.7172} = 0.3735$$

$$y_{\mathrm{C}} = \frac{N_{\mathrm{C}}}{N_{\mathrm{total}}} = \frac{0.3586}{1.7172} = 0.2088$$

$$y_{\mathrm{H2}} = \frac{N_{\mathrm{H2}}}{N_{\mathrm{total}}} = \frac{0.7172}{1.7172} = 0.4177$$

The heat transfer can be determined from

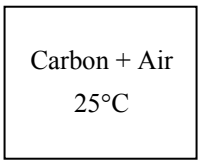
$$\begin{aligned} Q_{\mathrm{in}} &= N(y_{\mathrm{CH4}} c_{v,\mathrm{CH4}} T_2 + y_{\mathrm{H2}} c_{v,\mathrm{H2}} T_2 + y_{\mathrm{C}} c_{v,\mathrm{C}} T_2) - N c_{v,\mathrm{CH4}} T_1 \\ &= (10)\big[(0.3735)(63.3)(1000) + (0.4177)(21.7)(1000) + (0.2088)(0.711)(1000)\big] - (10)(27.8)(298) \\ &= \mathbf{245{,}700\ kJ} \end{aligned}$$

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16-94 Solid carbon is burned with a stoichiometric amount of air. The number of moles of CO_2 formed per mole of carbon is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O_2, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis Inspection of Table A-28 reveals that the dissociation equilibrium constants of CO_2, O_2, and N_2 are quite small and therefore may be neglected. (We learned from another source that the equilibrium constant for CO is also small). The combustion is then complete and the reaction is described by



$$C + (O_2 + 3.76N_2) \longrightarrow CO_2 + 3.76N_2$$

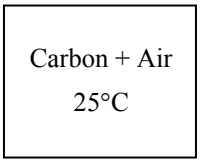
The number of moles of CO_2 in the products is then

$$\frac{N_{CO2}}{N_C} = \mathbf{1}$$

16-95 Solid carbon is burned with a stoichiometric amount of air. The amount of heat released per kilogram of carbon is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O_2, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis Inspection of Table A-28 reveals that the dissociation equilibrium constants of CO_2, O_2, and N_2 are quite small and therefore may be neglected. (We learned from another source that the equilibrium constant for CO is also small). The combustion is then complete and the reaction is described by



$$C + (O_2 + 3.76N_2) \longrightarrow CO_2 + 3.76N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\overline{h}_f^\circ$	$\overline{h}_{298K}$	$\overline{h}_{1240K}$
	kJ/kmol	kJ/kmol	kJ/kmol
N_2	0	8669	38,129
CO_2	-393,520	9364	56,108

Substituting,

$$\begin{aligned} -Q_{out} &= (1)(-393{,}520 + 56{,}108 - 9364) + (3.76)(0 + 38{,}129 - 8669) \\ &= -236{,}000 \text{ kJ/kmol C} \end{aligned}$$

or $$Q_{out} = \frac{236{,}000 \text{ kJ/kmol}}{12 \text{ kg/kmol}} = \mathbf{19{,}670 \text{ kJ/kg C}}$$

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16-96 Methane gas is burned with 30 percent excess air. The equilibrium composition of the products of combustion and the amount of heat released by this combustion are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, H_2O, O_2, NO, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis Inspection of the equilibrium constants of the possible reactions indicate that only the formation of NO need to be considered in addition to other complete combustion products. Then, the stoichiometric and actual reactions in this case are

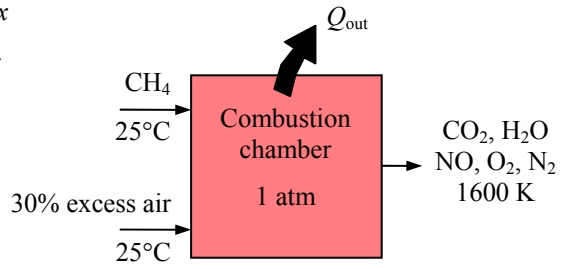
Stoichiometric: $\quad N_2 + O_2 \Leftrightarrow 2NO \quad (\text{thus } \nu_{N2} = 1,\ \nu_{O2} = 1,\ \text{and } \nu_{NO} = 2)$

Actual: $\quad CH_4 + 2.6(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + xNO + yO_2 + zN_2$

N balance: $\quad 2 \times 9.776 = x + 2z \longrightarrow z = 9.776 - 0.5x$

O balance: $\quad 5.2 = 2 + 2 + x + 2y \longrightarrow y = 0.6 - 0.5x$

Total number of moles: $\quad N_{\text{total}} = 1 + 2 + x + y + z = 13.38$



The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N2}^{\nu_{N2}} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{NO} - \nu_{N2} - \nu_{O2})}$$

From Table A-28, at 1600 K, $\ln K_p = -5.294$. Since the stoichiometric reaction being considered is double this reaction,

$$K_p = \exp(-2 \times 5.294) = 2.522 \times 10^{-5}$$

Substituting,

$$2.522 \times 10^{-5} = \frac{x^2}{(0.6 - 0.5x)(9.766 - 0.5x)} \left(\frac{1}{13.38}\right)^{2-1-1}$$

Solving for x,

$$x = 0.0121$$

Then,

$$y = 0.6 - 0.5x = 0.594$$

$$z = 9.776 - 0.5x = 9.77$$

Therefore, the equilibrium composition of the products mixture at 1600 K and 1 atm is

$$\mathbf{CH_4 + 2.6(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 0.0121NO + 0.594O_2 + 9.77N_2}$$

The heat transfer for this combustion process is determined from the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{\text{out}} = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298K}$ kJ/kmol	$\bar{h}_{1600K}$ kJ/kmol
CH_4	-74,850	---	---
O_2	0	8682	52,961
N_2	0	8669	50,571
H_2O	-241,820	9904	62,748
CO_2	-393,520	9364	76,944

Neglecting the effect of NO in the energy balance and substituting,

$$\begin{aligned} -Q_{\text{out}} &= (1)(-393{,}520 + 76{,}944 - 9364) + (2)(-241{,}820 + 62{,}748 - 9904) + 0.594(52{,}961 - 8682) \\ &\quad + (9.77)(50{,}571 - 8669) - (-74{,}850) \\ &= -193{,}500 \text{ kJ/kmol } CH_4 \end{aligned}$$

or $\quad Q_{\text{out}} = \mathbf{193{,}500\ kJ/kmol\ CH_4}$

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16-97E Gaseous octane gas is burned with 40% excess air. The equilibrium composition of the products of combustion is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, H_2O, O_2, NO, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis The stoichiometric and actual reactions in this case are

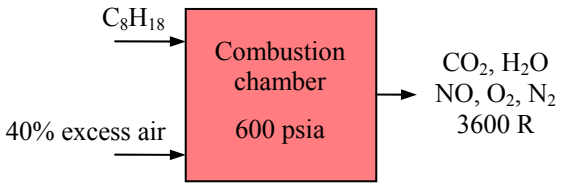
Stoichiometric: $\mathrm{N_2 + O_2 \Leftrightarrow 2NO}$ (thus $\nu_{N2} = 1$, $\nu_{O2} = 1$, and $\nu_{NO} = 2$)

Actual: $$\mathrm{C_8H_{18} + 1.4 \times 12.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + xNO + yO_2 + zN_2}$$

N balance: $131.6 = x + 2z \longrightarrow z = 65.8 - 0.5x$

O balance: $35 = 16 + 9 + x + 2y \longrightarrow y = 5 - 0.5x$

Total number of moles: $N_{total} = 8 + 9 + x + y + z = 87.8$



The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N2}^{\nu_{N2}} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{total}} \right)^{(\nu_{NO} - \nu_{N2} - \nu_{O2})}$$

From Table A-28, at 2000 K (3600 R), $\ln K_p = -3.931$. Since the stoichiometric reaction being considered is double this reaction,

$$K_p = \exp(-2 \times 3.931) = 3.851 \times 10^{-4}$$

Substituting,

$$3.851 \times 10^{-4} = \frac{x^2}{(5 - 0.5x)(65.8 - 0.5x)} \left(\frac{600/14.7}{87.8} \right)^{2-1-1}$$

Solving for x,

$$x = 0.3492$$

Then,

$$y = 5 - 0.5x = 4.825$$

$$z = 65.8 - 0.5x = 65.63$$

Therefore, the equilibrium composition of the products mixture at 2000 K and 4 MPa is

$$\mathbf{C_8H_{18} + 17.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 0.3492NO + 4.825O_2 + 65.63N_2}$$

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16-98 Propane gas is burned with 20% excess air. The equilibrium composition of the products of combustion on a mass basis and the amount of heat released by this combustion are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, H_2O, O_2, NO, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis (*a*) The stoichiometric and actual reactions in this case are

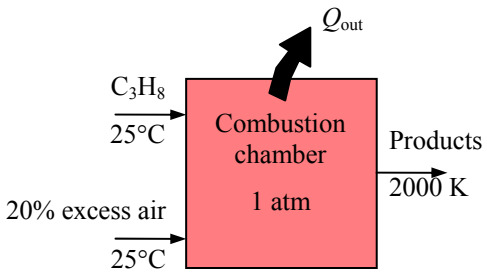
Stoichiometric: $\quad N_2 + O_2 \Leftrightarrow 2NO \quad (\text{thus } \nu_{N2} = 1,\ \nu_{O2} = 1,\ \text{and } \nu_{NO} = 2)$

Actual: $\quad C_3H_8 + 1.2 \times 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + xNO + yO_2 + zN_2$

N balance: $\quad 45.12 = x + 2z \longrightarrow z = 22.56 - 0.5x$

O balance: $\quad 12 = 6 + 4 + x + 2y \longrightarrow y = 1 - 0.5x$

Total number of moles: $\quad N_{\text{total}} = 3 + 4 + x + y + z = 30.56$



The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N2}^{\nu_{N2}} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{NO} - \nu_{N2} - \nu_{O2})}$$

From Table A-28, at 2000 K, $\ln K_p = -3.931$. Since the stoichiometric reaction being considered is double this reaction,

$$K_p = \exp(-2 \times 3.931) = 3.851 \times 10^{-4}$$

Substituting,

$$3.851 \times 10^{-4} = \frac{x^2}{(1 - 0.5x)(22.56 - 0.5x)} \left(\frac{1}{30.56}\right)^{2-1-1}$$

Solving for x,

$$x = 0.09097$$

Then,

$$y = 1 - 0.5x = 0.9545$$

$$z = 22.56 - 0.5x = 22.51$$

Therefore, the equilibrium composition of the products mixture at 2000 K and 1 atm is

$$C_3H_8 + 6(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 0.09097NO + 0.9545O_2 + 22.51N_2$$

The mass of each product and the total mass of the products is

$$m_{CO2} = N_{CO2} M_{CO2} = (3\ \text{kmol})(44\ \text{kg/kmol}) = 132\ \text{kg}$$
$$m_{H2O} = N_{H2O} M_{H2O} = (4\ \text{kmol})(18\ \text{kg/kmol}) = 72\ \text{kg}$$
$$m_{NO} = N_{NO} M_{NO} = (0.09097\ \text{kmol})(30\ \text{kg/kmol}) = 2.73\ \text{kg}$$
$$m_{O2} = N_{O2} M_{O2} = (0.9545\ \text{kmol})(32\ \text{kg/kmol}) = 30.54\ \text{kg}$$
$$m_{N2} = N_{N2} M_{N2} = (22.51\ \text{kmol})(28\ \text{kg/kmol}) = 630.28\ \text{kg}$$
$$m_{\text{total}} = 132 + 72 + 2.73 + 30.54 + 630.28 = 867.55\ \text{kg}$$

The mass fractions of the products are

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$$\text{mf}_{\text{CO2}} = \frac{m_{\text{CO2}}}{m_{\text{total}}} = \frac{132\ \text{kg}}{867.55\ \text{kg}} = \mathbf{0.1522}$$

$$\text{mf}_{\text{H2O}} = \frac{m_{\text{H2O}}}{m_{\text{total}}} = \frac{72\ \text{kg}}{867.55\ \text{kg}} = \mathbf{0.0830}$$

$$\text{mf}_{\text{NO}} = \frac{m_{\text{NO}}}{m_{\text{total}}} = \frac{2.73\ \text{kg}}{867.55\ \text{kg}} = \mathbf{0.0031}$$

$$\text{mf}_{\text{O2}} = \frac{m_{\text{O2}}}{m_{\text{total}}} = \frac{30.54\ \text{kg}}{867.55\ \text{kg}} = \mathbf{0.0352}$$

$$\text{mf}_{\text{N2}} = \frac{m_{\text{N2}}}{m_{\text{total}}} = \frac{630.28\ \text{kg}}{867.55\ \text{kg}} = \mathbf{0.7265}$$

(*b*) The heat transfer for this combustion process is determined from the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{\text{out}} = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\overline{h}_f^\circ$ kJ/kmol	$\overline{h}_{298\text{K}}$ kJ/kmol	$\overline{h}_{1600\text{K}}$ kJ/kmol
C_3H_8	-103,850	---	---
O_2	0	8682	67,881
N_2	0	8669	64,810
H_2O	-241,820	9904	82,593
CO_2	-393,520	9364	100,804

Neglecting the effect of NO in the energy balance and substituting,

$$\begin{aligned} -Q_{\text{out}} &= (3)(-393{,}520 + 100{,}804 - 9364) + (4)(-241{,}820 + 82{,}593 - 9904) + 0.9545(67{,}881 - 8682) \\ &\quad + (22.51)(64{,}810 - 8669) - (-103{,}850) \\ &= -158{,}675\ \text{kJ/kmol}\ C_3H_8 \end{aligned}$$

or $$Q_{\text{out}} = \frac{158{,}675\ \text{kJ/kmol}}{44\ \text{kg/kmol}} = \mathbf{3606\ kJ/kg\ C_3H_8}$$

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16-99 Propane gas is burned with stoichiometric air in an adiabatic manner. The temperature of the products and the equilibrium composition of the products are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, H_2O, O_2, NO, and N_2. **2** The constituents of the mixture are ideal gases.

Analysis (*a*) The stoichiometric and actual reactions in this case are

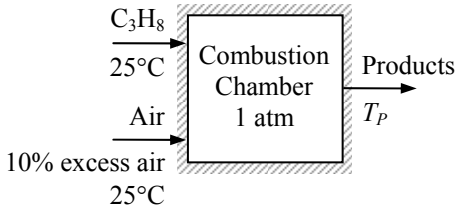
Stoichiometric: $N_2 + O_2 \Leftrightarrow 2NO$ (thus $\nu_{N2} = 1,\ \nu_{O2} = 1,$ and $\nu_{NO} = 2$)

Actual: $C_3H_8 + 1.1\times 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + xNO + yO_2 + zN_2$

N balance: $41.36 = x + 2z \longrightarrow z = 20.68 - 0.5x$

O balance: $11 = 6 + 4 + x + 2y \longrightarrow y = 0.5 - 0.5x$

Total number of moles: $N_{\text{total}} = 3 + 4 + x + y + z = 21.18$



The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N2}^{\nu_{N2}} N_{O2}^{\nu_{O2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{NO}-\nu_{N2}-\nu_{O2})}$$

We assume that the products will be at 2000 K. Then from Table A-28, at 2000 K, $\ln K_p = -3.931$. Since the stoichiometric reaction being considered is double this reaction,

$$K_p = \exp(-2\times 3.931) = 3.851\times 10^{-4}$$

Substituting,

$$3.851\times 10^{-4} = \frac{x^2}{(0.5-0.5x)(20.68-0.5x)}\left(\frac{1}{21.18}\right)^{2-1-1}$$

Solving for x,

$$x = 0.0611$$

Then,

$$y = 0.5 - 0.5x = 0.4695$$

$$z = 20.68 - 0.5x = 20.65$$

Therefore, the equilibrium composition of the products mixture at 2000 K and 1 atm is

$$\mathbf{C_3H_8 + 6(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 0.0611NO + 0.4695O_2 + 20.65N_2}$$

(*b*) From the tables,

Substance	$\bar{h}_f^\circ$, kJ/kmol	$\bar{h}_{298K}$, kJ/kmol
C_3H_8 (*g*)	-103,850	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO_2	-393,520	9364

Thus,

$$(3)\left(-393{,}520 + \bar{h}_{CO2} - 9364\right) + (4)\left(-241{,}820 + \bar{h}_{H2O} - 9904\right) + (0.0611)\left(39{,}460 + \bar{h}_{OH} - 9188\right)$$
$$+ (0.4695)\left(0 + \bar{h}_{O2} - 8682\right) + (20.65)\left(0 + \bar{h}_{N2} - 8669\right) = (1)\left(-103{,}850\right) + 0 + 0$$

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It yields

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 0.0611\bar{h}_{OH} + 0.4695\bar{h}_{O2} + 20.65\bar{h}_{N2} = 2{,}292{,}940 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,292,940/(3+4+0.0611+0.4695+20.65) = 81,366 kJ/kmol. This enthalpy value corresponds to about 2450 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2450 K, but somewhat under it because of the higher specific heat of H_2O.

At 2200 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 0.0611\bar{h}_{OH} + 0.4695\bar{h}_{O2} + 20.65\bar{h}_{N2} = 3(112{,}939) + 4(92{,}940) + 0.0611(69{,}932)$$
$$+ 0.4695(75{,}484) + 20.65(64{,}810) = 2{,}088{,}620 \text{ kJ} \quad \text{(Lower than 2,292,940)}$$

At 2400 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 0.0611\bar{h}_{OH} + 0.4695\bar{h}_{O2} + 20.65\bar{h}_{N2} = 3(125{,}152) + 4(103{,}508) + 0.0611(77{,}015)$$
$$+ 0.4695(83{,}174) + 20.65(79{,}320) = 2{,}471{,}200 \text{ kJ} \quad \text{(Higher than 2,292,940)}$$

By interpolation of the two results,

T_P = **2307 K = 2034°C**

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16-100 A mixture of H_2O and O_2 is heated to a high temperature. The equilibrium composition is to be determined.

Assumptions **1** The equilibrium composition consists of H_2O, OH, O_2, and H_2. **2** The constituents of the mixture are ideal gases.

Analysis The reaction equation during this process can be expressed as

H_2O, OH, H_2, O_2
3600 K
8 atm

$$2H_2O + 3O_2 \longrightarrow x\,H_2O + y\,H_2 + z\,O_2 + w\,OH$$

Mass balances for hydrogen and oxygen yield

H balance: $4 = 2x + 2y + w$ (1)

O balance: $8 = x + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the K_P relations) to determine the equilibrium composition of the mixture. They are

$$H_2O \Leftrightarrow H_2 + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$H_2O \Leftrightarrow \tfrac{1}{2}H_2 + OH \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3600 K are determined from Table A-28 to be

$$\ln K_{P1} = -1.392 \longrightarrow K_{P1} = 0.24858$$

$$\ln K_{P2} = -1.088 \longrightarrow K_{P2} = 0.33689$$

The K_P relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{O_2}-\nu_{H_2O})}$$

$$K_{P2} = \frac{N_{H_2}^{\nu_{H_2}} N_{OH}^{\nu_{OH}}}{N_{H_2O}^{\nu_{H_2O}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{OH}-\nu_{H_2O})}$$

where

$$N_{\text{total}} = N_{H_2O} + N_{H_2} + N_{O_2} + N_{OH} = x + y + z + w$$

Substituting,

$$0.24858 = \frac{(y)(z)^{1/2}}{x}\left(\frac{8}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$0.33689 = \frac{(w)(y)^{1/2}}{x}\left(\frac{8}{x+y+z+w}\right)^{1/2} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns x, y, z, and w yields

$x = 1.371$ $y = 0.1646$ $z = 2.85$ $w = 0.928$

Therefore, the equilibrium composition becomes

$$\mathbf{1.371H_2O + 0.165H_2 + 2.85O_2 + 0.928OH}$$

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16-101 A mixture of CO_2 and O_2 is heated to a high temperature. The equilibrium composition is to be determined.

Assumptions **1** The equilibrium composition consists of CO_2, CO, O_2, and O. **2** The constituents of the mixture are ideal gases.

Analysis The reaction equation during this process can be expressed as

$$3CO_2 + 3O_2 \longrightarrow xCO_2 + yCO + zO_2 + wO$$

CO_2, CO, O_2, O
2600 K
1.5 atm

Mass balances for carbon and oxygen yield

C balance: $3 = x + y$ (1)

O balance: $12 = 2x + y + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the K_P relations) to determine the equilibrium composition of the mixture. They are

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$O_2 \Leftrightarrow 2O \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 2600 K are determined from Table A-28 to be

$$\ln K_{P1} = -2.801 \longrightarrow K_{P1} = 0.06075$$

$$\ln K_{P2} = -7.521 \longrightarrow K_{P2} = 0.0005416$$

The K_P relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{CO}+\nu_{O_2}-\nu_{CO_2})}$$

$$K_{P2} = \frac{N_{O}^{\nu_{O}}}{N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{\text{total}}}\right)^{\nu_{O}-\nu_{O_2}}$$

where

$$N_{\text{total}} = N_{CO_2} + N_{O_2} + N_{CO} + N_{O} = x + y + z + w$$

Substituting,

$$0.06075 = \frac{(y)(z)^{1/2}}{x}\left(\frac{1.5}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$0.0005416 = \frac{w^2}{z}\left(\frac{1.5}{x+y+z+w}\right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns x, y, z, and w yields

$x = 2.803$ $y = 0.197$ $z = 3.057$ $w = 0.08233$

Thus the equilibrium composition is

$$\mathbf{2.803CO_2 + 0.197CO + 3.057O_2 + 0.0823O}$$

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16-102 Problem 16-101 is reconsidered. The effect of pressure on the equilibrium composition by varying pressure from 1 atm to 10 atm is to be studied.

Analysis The problem is solved using EES, and the solution is given below.

```
"Given"
T=2600 [K]
P=1.5 [atm]

"The equilibrium constant for these two reactions at 2600 K are determined from Table A-28"
K_p1=exp(-2.801)
K_p2=exp(-7.521)

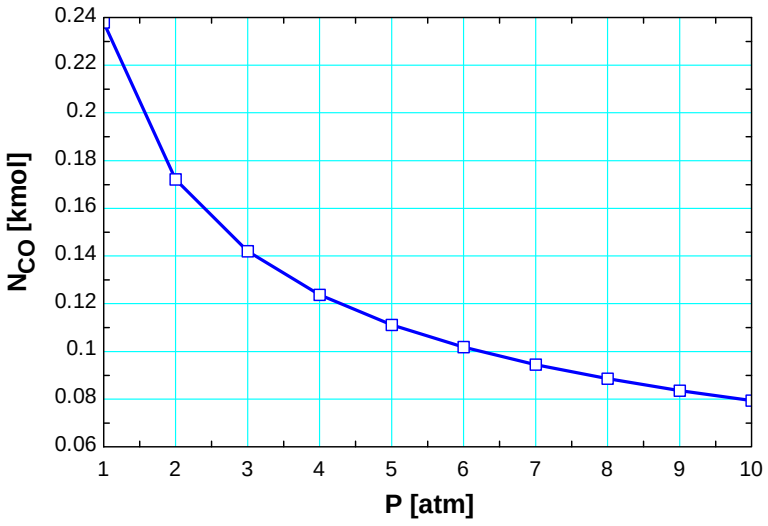
"Analysis"
"Actual reaction:  3 CO2 + 3 O2 = N_CO2 CO2 + N_CO CO + N_O2 O2 + N_O O"
3=N_CO2+N_CO "C balance"
12=2*N_CO2+N_CO+2*N_O2+N_O "O balance"
N_total=N_CO2+N_CO+N_O2+N_O

"Stoichiometric reaction 1:  CO2 = CO + 1/2 O2"
"Stoichiometric coefficients for reaction 1"
nu_CO2_1=1
nu_CO_1=1
nu_O2_1=1/2

"Stoichiometric reaction 2:  O2 = 2 O"
"Stoichiometric coefficients for reaction 2"
nu_O2_2=1
nu_O_2=2

"K_p relations are"
K_p1=(N_CO^nu_CO_1*N_O2^nu_O2_1)/N_CO2^nu_CO2_1*(P/N_total)^(nu_CO_1+nu_O2_1-nu_CO2_1)
K_p2=N_O^nu_O_2/N_O2^nu_O2_2*(P/N_total)^(nu_O_2-nu_O2_2)
```

P_{atm} [atm]	b [$kmol_{CO}$]
1	0.2379
2	0.1721
3	0.1419
4	0.1237
5	0.1111
6	0.1017
7	0.09442
8	0.0885
9	0.08357
10	0.0794



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16-103 The $\bar{h}_R$ at a specified temperature is to be determined using enthalpy and K_p data.

Assumptions Both the reactants and products are ideal gases.

Analysis (*a*) The complete combustion equation of H_2 can be expressed as

$$H_2 + \tfrac{1}{2} O_2 \Leftrightarrow H_2O$$

The $\bar{h}_R$ of the combustion process of H_2 at 2400 K is the amount of energy released as one kmol of H_2 is burned in a steady-flow combustion chamber at a temperature of 2400 K, and can be determined from

$$\bar{h}_R = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the H_2O, H_2, and O_2 to be ideal gases, we have $h = h\,(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\,K}$ kJ/kmol	$\bar{h}_{2400\,K}$ kJ/kmol
H_2O	-241,820	9904	103,508
H_2	0	8468	75,383
O_2	0	8682	83,174

Substituting,

$$\begin{aligned}\bar{h}_R &= 1(-241{,}820 + 103{,}508 - 9904) \\ &\quad - 1(0 + 75{,}383 - 8468) \\ &\quad - 0.5(0 + 83{,}174 - 8682) \\ &= \mathbf{-252{,}377\ kJ\ /\ kmol}\end{aligned}$$

(*b*) The $\bar{h}_R$ value at 2400 K can be estimated by using K_P values at 2200 K and 2600 K (the closest two temperatures to 2400 K for which K_P data are available) from Table A-28,

$$\ln \frac{K_{P2}}{K_{P1}} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$4.648 - 6.768 \cong \frac{\bar{h}_R}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{2200\ \text{K}} - \frac{1}{2600\ \text{K}}\right)$$

$$\bar{h}_R \cong \mathbf{-252{,}047\ kJ/kmol}$$

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16-104 Problem 16-103 is reconsidered. The effect of temperature on the enthalpy of reaction using both methods by varying the temperature from 2000 to 3000 K is to be investigated.

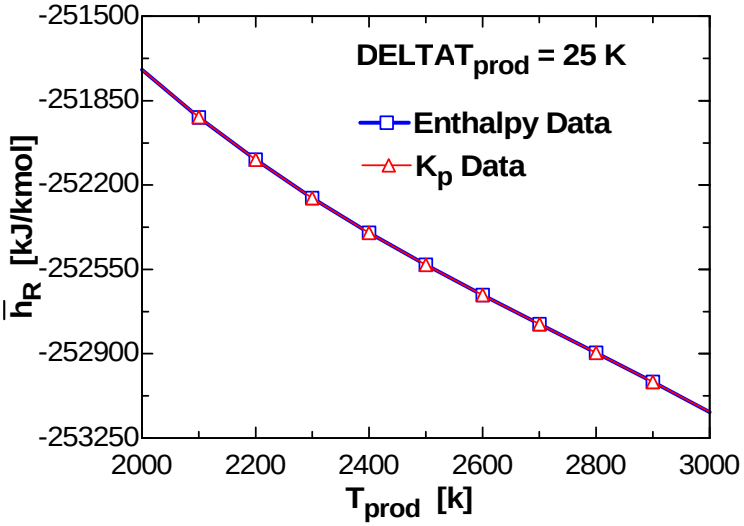
Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
T_prod=2400 [K]
DELTAT_prod =25 [K]
R_u=8.314 [kJ/kmol-K]
T_prod_1 = T_prod - DELTAT_prod
T_prod_2 = T_prod + DELTAT_prod

"The combustion equation is   1 H2 + 0.5 O2 =>1 H2O"
"The enthalpy of reaction H_bar_R using enthalpy data is:"
h_bar_R_Enthalpy = HP - HR
HP = 1*Enthalpy(H2O,T=T_prod )
HR = 1*Enthalpy(H2,T=T_prod ) + 0.5*Enthalpy(O2,T=T_prod )

"The enthalpy of reaction H_bar_R using enthalpy data is found using the following equilibruim data:" "The following equations provide the specific Gibbs function (g=h-Ts) for
each component as a function of its temperature at 1 atm pressure, 101.3 kPa"
g_H2O_1=Enthalpy(H2O,T=T_prod_1 )-T_prod_1 *Entropy(H2O,T=T_prod_1 ,P=101.3)
g_H2_1=Enthalpy(H2,T=T_prod_1 )-T_prod_1 *Entropy(H2,T=T_prod_1 ,P=101.3)
g_O2_1=Enthalpy(O2,T=T_prod_1 )-T_prod_1 *Entropy(O2,T=T_prod_1 ,P=101.3)
g_H2O_2=Enthalpy(H2O,T=T_prod_2 )-T_prod_2 *Entropy(H2O,T=T_prod_2 ,P=101.3)
g_H2_2=Enthalpy(H2,T=T_prod_2 )-T_prod_2 *Entropy(H2,T=T_prod_2 ,P=101.3)
g_O2_2=Enthalpy(O2,T=T_prod_2 )-T_prod_2 *Entropy(O2,T=T_prod_2 ,P=101.3)
"The standard-state (at 1 atm) Gibbs functions are"
DELTAG_1 =1*g_H2O_1-0.5*g_O2_1-1*g_H2_1
DELTAG_2 =1*g_H2O_2-0.5*g_O2_2-1*g_H2_2
"The equilibrium constants are given by Eq. 15-14."
K_p_1 = exp(-DELTAG_1/(R_u*T_prod_1)) "From EES data"
K_P_2 = exp(-DELTAG_2/(R_u*T_prod_2)) "From EES data"
"the entahlpy of reaction is estimated from the equilibrium constant K_p by using EQ 15-18 as:"
ln(K_P_2/K_P_1)=h_bar_R_Kp/R_u*(1/T_prod_1 - 1/T_prod_2)
PercentError = ABS((h_bar_R_enthalpy - h_bar_R_Kp)/h_bar_R_enthalpy)*Convert(, %)
```

Percent Error [%]	T_{prod} [K]	$h_{REnthalpy}$ [kJ/kmol]	h_{RKp} [kJ/kmol]
0.0002739	2000	-251723	-251722
0.0002333	2100	-251920	-251919
0.000198	2200	-252096	-252095
0.0001673	2300	-252254	-252254
0.0001405	2400	-252398	-252398
0.0001173	2500	-252532	-252531
0.00009706	2600	-252657	-252657
0.00007957	2700	-252778	-252777
0.00006448	2800	-252897	-252896
0.00005154	2900	-253017	-253017
0.0000405	3000	-253142	-253142



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16-105 The K_P value of the dissociation process $O_2 \Leftrightarrow 2O$ at a specified temperature is to be determined using the $\bar{h}_R$ data and K_P value at a specified temperature.

Assumptions Both the reactants and products are ideal gases.

Analysis The $\bar{h}_R$ and K_P data are related to each other by

$$\ln \frac{K_{P2}}{K_{P1}} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1}-\frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1}-\frac{1}{T_2}\right)$$

The $\bar{h}_R$ of the specified reaction at 2800 K is the amount of energy released as one kmol of O_2 dissociates in a steady-flow combustion chamber at a temperature of 2800 K, and can be determined from

$$\bar{h}_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the O_2 and O to be ideal gases, we have $h = h\,(T)$. From the tables,

Substance	$\bar{h}_f^\circ$ kJ/kmol	$\bar{h}_{298\,K}$ kJ/kmol	$\bar{h}_{2800\,K}$ kJ/kmol
O	249,190	6852	59,241
O_2	0	8682	98,826

Substituting,

$$\bar{h}_R = 2(249{,}190 + 59{,}241 - 6852) - 1(0 + 98{,}826 - 8682)$$
$$= 513{,}014 \text{ kJ / kmol}$$

The K_P value at 3000 K can be estimated from the equation above by using this $\bar{h}_R$ value and the K_P value at 2600 K which is $\ln K_{P1} = -7.521$,

$$\ln K_{P2} - (-7.521) = \frac{513{,}014 \text{ kJ/kmol}}{8.314 \text{ kJ/kmol} \cdot \text{K}}\left(\frac{1}{2600 \text{ K}} - \frac{1}{3000 \text{ K}}\right)$$

$$\ln K_{P2} = -4.357 \quad \text{(Table A - 28 : } \ln K_{P2} = -4.357\text{)}$$

or

$$K_{P2} = \mathbf{0.0128}$$

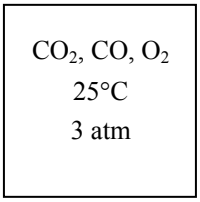
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16-106 A mixture of CO and O_2 contained in a tank is ignited. The final pressure in the tank and the amount of heat transfer are to be determined.

Assumptions **1** The equilibrium composition consists of CO_2 and O_2. **2** Both the reactants and the products are ideal gases.

Analysis The combustion equation can be written as

$$CO + 3O_2 \longrightarrow CO_2 + 2.5O_2$$



The heat transfer can be determined from

$$-Q_{\text{out}} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ - P\overline{v}\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ - P\overline{v}\right)_R$$

Both the reactants and the products are assumed to be ideal gases, and thus all the internal energy and enthalpies depend on temperature only, and the $P\overline{v}$ terms in this equation can be replaced by R_uT. It yields

$$-Q_{\text{out}} = \sum N_P\left(\overline{h}_f^\circ + \overline{h}_{800\text{ K}} - \overline{h}_{298\text{ K}} - R_uT\right)_P - \sum N_R\left(\overline{h}_f^\circ - R_uT\right)_R$$

since reactants are at the standard reference temperature of 25°C. From the tables,

Substance	$\overline{h}_f^\circ$ kJ/kmol	$\overline{h}_{298\text{ K}}$ kJ/kmol	$\overline{h}_{800\text{K}}$ kJ/kmol
CO	-110,530	8669	23,844
O_2	0	8682	24,523
CO_2	-393,520	9364	32,179

Substituting,

$$\begin{aligned}-Q_{\text{out}} &= 1(-393{,}520 + 32{,}179 - 9364 - 8.314\times 800)\\ &\quad + 2.5(0 + 24{,}523 - 8682 - 8.314\times 800)\\ &\quad - 3(0 - 8.314\times 298)\\ &\quad - 1(-110{,}530 - 8.314\times 298)\\ &= \mathbf{-233{,}940\ kJ/kmol\ CO}\end{aligned}$$

or

$$Q_{\text{out}} = \mathbf{233{,}940\ kJ/kmol\ CO}$$

The final pressure in the tank is determined from

$$\frac{P_1\boldsymbol{V}}{P_2\boldsymbol{V}} = \frac{N_1R_uT_1}{N_2R_uT_2} \longrightarrow P_2 = \frac{N_2T_2}{N_1T_1}P_1 = \frac{3.5}{4}\times\frac{800\text{ K}}{298\text{ K}}(3\text{ atm}) = \mathbf{7.05\ atm}$$

The equilibrium constant for the reaction $CO + \frac{1}{2}O_2 \Leftrightarrow CO_2$ at 800 K is $\ln K_P = 37.2$ (by interpolation), which is much greater than 7.05. Therefore, it is not realistic to assume that no CO will be present in equilibrium mixture.

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16-107 A 2-L bottle is filled with carbonated drink that is fully charged (saturated) with CO_2 gas. The volume that the CO_2 gas would occupy if it is released and stored in a container at room conditions is to be determined.

Assumptions **1** The liquid drink can be treated as water. **2** Both the CO_2 gas and the water vapor are ideal gases. **3** The CO_2 gas is weakly soluble in water and thus Henry's law is applicable.

Properties The saturation pressure of water at 17°C is 1.938 kPa (Table A-4). Henry's constant for CO_2 dissolved in water at 17ºC (290 K) is H = 1280 bar (Table 16-2). Molar masses of CO_2 and water are 44.01 and 18.015 kg/kmol, respectively (Table A-1). The gas constant of CO_2 is 0.1889 kPa.m^3/kg.K. Also, 1 bar = 100 kPa.

Analysis In the charging station, the CO_2 gas and water vapor mixture above the liquid will form a saturated mixture. Noting that the saturation pressure of water at 17°C is 1.938 kPa, the partial pressure of the CO_2 gas is

$$P_{\mathrm{CO_2,\,gas\,side}} = P - P_{\mathrm{vapor}} = P - P_{\mathrm{sat\,@\,17°C}} = 600 - 1.938 = 598.06\ \mathrm{kPa} = 5.9806\ \mathrm{bar}$$

From Henry's law, the mole fraction of CO_2 in the liquid drink is determined to be

$$y_{\mathrm{CO_2,liquid\,side}} = \frac{P_{\mathrm{CO_2,gas\,side}}}{H} = \frac{5.9806\ \mathrm{bar}}{1280\ \mathrm{bar}} = 0.00467$$

Then the mole fraction of water in the drink becomes

$$y_{\mathrm{water,\,liquid\,side}} = 1 - y_{\mathrm{CO_2,\,liquid\,side}} = 1 - 0.00467 = 0.99533$$

The mass and mole fractions of a mixture are related to each other by

$$w_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \frac{M_i}{M_m}$$

where the apparent molar mass of the drink (liquid water - CO_2 mixture) is

$$\begin{aligned} M_m &= \sum y_i M_i = y_{\mathrm{liquid\,water}} M_{\mathrm{water}} + y_{\mathrm{CO_2}} M_{\mathrm{CO_2}} \\ &= 0.99533 \times 18.015 + 0.00467 \times 44.01 = 18.14\ \mathrm{kg/kmol} \end{aligned}$$

Then the mass fraction of dissolved CO_2 in liquid drink becomes

$$w_{\mathrm{CO_2,\,liquid\,side}} = y_{\mathrm{CO_2,\,liquid\,side}}(0)\frac{M_{\mathrm{CO_2}}}{M_m} = 0.00467\frac{44.01}{18.14} = 0.0113$$

Therefore, the mass of dissolved CO_2 in a 2 L ≈ 2 kg drink is

$$m_{\mathrm{CO_2}} = w_{\mathrm{CO_2}} m_m = 0.0113(2\ \mathrm{kg}) = 0.0226\ \mathrm{kg}$$

Then the volume occupied by this CO_2 at the room conditions of 20°C and 100 kPa becomes

$$V = \frac{mRT}{P} = \frac{(0.0226\ \mathrm{kg})(0.1889\ \mathrm{kPa \cdot m^3/kg \cdot K})(293\ \mathrm{K})}{100\ \mathrm{kPa}} = \mathbf{0.0125\ m^3 = 12.5\ L}$$

Discussion Note that the amount of dissolved CO_2 in a 2-L pressurized drink is large enough to fill 6 such bottles at room temperature and pressure. Also, we could simplify the calculations by assuming the molar mass of carbonated drink to be the same as that of water, and take it to be 18 kg/kmol because of the very low mole fraction of CO_2 in the drink.

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16-108 Ethyl alcohol C_2H_5OH (gas) is burned in a steady-flow adiabatic combustion chamber with 90 percent excess air. The adiabatic flame temperature of the products is to be determined and the adiabatic flame temperature as a function of the percent excess air is to be plotted.

Analysis The complete combustion reaction in this case can be written as

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{\text{th}}\left[O_2 + 3.76N_2\right] \longrightarrow 2\,CO_2 + 3\,H_2O + (Ex)(a_{\text{th}})\,O_2 + f\,N_2$$

where a_{th} is the stoichiometric coefficient for air. The oxygen balance gives

$$1 + (1+Ex)a_{\text{th}} \times 2 = 2\times 2 + 3\times 1 + (Ex)(a_{\text{th}})\times 2$$

The reaction equation with products in equilibrium is

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{\text{th}}\left[O_2 + 3.76N_2\right] \longrightarrow a\,CO_2 + b\,CO + d\,H_2O + e\,O_2 + f\,N_2$$

The coefficients are determined from the mass balances

Carbon balance: $2 = a + b$

Hydrogen balance: $6 = 2d \longrightarrow d = 3$

Oxygen balance: $1 + (1+Ex)a_{\text{th}} \times 2 = a\times 2 + b + d + e\times 2$

Nitrogen balance: $(1+Ex)a_{\text{th}} \times 3.76 = f$

Solving the above equations, we find the coefficients to be

$$Ex = 0.9,\ a_{\text{th}} = 3,\ a = 2,\ b = 0.00008644,\ d = 3,\ e = 2.7,\ f = 21.43$$

Then, we write the balanced reaction equation as

$$C_2H_5OH\,(\text{gas}) + 5.7\left[O_2 + 3.76N_2\right] \longrightarrow 2\,CO_2 + 0.00008644\,CO + 3\,H_2O + 2.7\,O_2 + 21.43\,N_2$$

Total moles of products at equilibrium are

$$N_{\text{tot}} = 2 + 0.00008644 + 3 + 2.7 + 21.43 = 29.13$$

The assumed equilibrium reaction is

$$CO_2 \longleftrightarrow CO + 0.5O_2$$

The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{CO}\,\overline{g}^*_{CO}(T_{\text{prod}}) + \nu_{O2}\,\overline{g}^*_{O2}(T_{\text{prod}}) - \nu_{CO2}\,\overline{g}^*_{CO2}(T_{\text{prod}})$$

and the Gibbs functions are defined as

$$\overline{g}^*_{CO}(T_{\text{prod}}) = (\overline{h} - T_{\text{prod}}\overline{s})_{CO}$$
$$\overline{g}^*_{O2}(T_{\text{prod}}) = (\overline{h} - T_{\text{prod}}\overline{s})_{O2}$$
$$\overline{g}^*_{CO2}(T_{\text{prod}}) = (\overline{h} - T_{\text{prod}}\overline{s})_{CO2}$$

The equilibrium constant is also given by

$$K_p = \frac{be^{0.5}}{a}\left(\frac{P}{N_{\text{tot}}}\right)^{1+0.5-1} = \frac{(0.00008644)(2.7)^{0.5}}{2}\left(\frac{1}{29.13}\right)^{0.5} = 0.00001316$$

A steady flow energy balance gives

$$H_R = H_P$$

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where

$$H_R = \bar{h}^o_{f\ \text{fuel@25°C}} + 5.7\bar{h}_{\text{O2@25°C}} + 21.43\bar{h}_{\text{N2@25°C}}$$
$$= (-235{,}310\ \text{kJ/kmol}) + 5.7(0) + 21.43(0) = -235{,}310\ \text{kJ/kmol}$$

$$H_P = 2\bar{h}_{\text{CO2@}T_{\text{prod}}} + 0.00008644\bar{h}_{\text{CO@}T_{\text{prod}}} + 3h_{\text{H2O@}T_{\text{prod}}} + 2.7\bar{h}_{\text{O2@}T_{\text{prod}}} + 21.43\bar{h}_{\text{N2@}T_{\text{prod}}}$$

Solving the energy balance equation using EES, we obtain the adiabatic flame temperature to be

$$T_{\text{prod}} = \mathbf{1569\ K}$$

The copy of entire EES solution including parametric studies is given next:

```
"The product temperature isT_prod"
"The reactant temperature is:"
T_reac= 25+273.15 "[K]"

"For adiabatic combustion of 1 kmol of fuel: "
Q_out = 0 "[kJ]"
PercentEx = 90  "Percent excess air"
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3"[kPa]"
R_u=8.314 "[kJ/kmol-K]"

"The complete combustion reaction equation for excess air  is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=2 CO2 +  3 H2O +Ex*A_th O2 + f N2"
"Oxygen Balance for complete combustion:"
1 + (1+Ex)*A_th*2=2*2+3*1 + Ex*A_th*2

"The reaction equation for excess air and products in equilibrium is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=a CO2 + b CO+ d H2O + e O2 + f N2"
"Carbon Balance:"
2=a + b
"Hydrogen Balance:"
6=2*d
"Oxygen Balance:"
1 + (1+Ex)*A_th*2=a*2+b + d + e*2
"Nitrogen Balance:"
(1+Ex)*A_th*3.76 = f
N_tot =a +b + d + e + f  "Total kilomoles of products at equilibrium"

"The assumed equilibrium reaction is CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)

"The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2

"The equilibrium constant is given by Eq. 15-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(b^1*e^0.5)/(a^1)"

sqrt(P/N_tot )*b *sqrt(e )=K_P *a
"The steady-flow energy balance is:"
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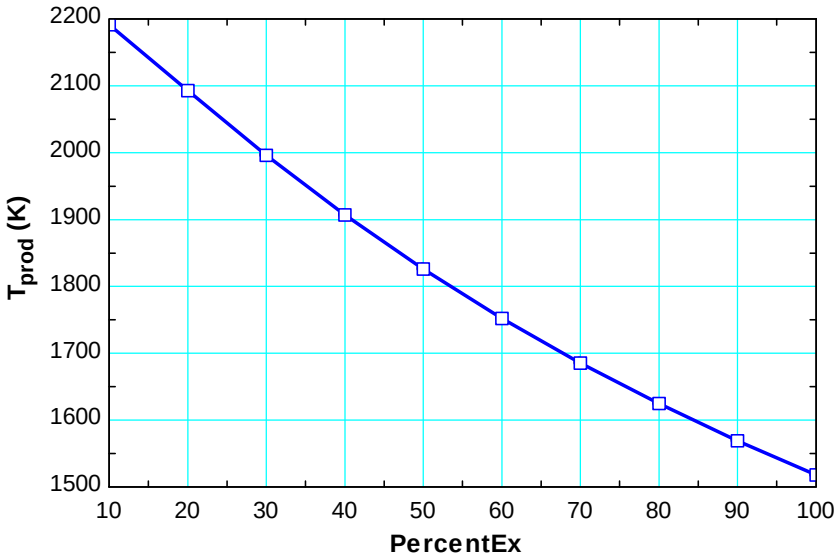
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H_R = Q_out+H_P

h_bar_f_C2H5OHgas=-235310 "[kJ/kmol]"

H_R=1*(h_bar_f_C2H5OHgas)
+(1+Ex)*A_th*ENTHALPY(O2,T=T_reac)+(1+Ex)*A_th*3.76*ENTHALPY(N2,T=T_reac) "[kJ/kmol]"
H_P=a*ENTHALPY(CO2,T=T_prod)+b*ENTHALPY(CO,T=T_prod)+d*ENTHALPY(H2O,T=T_prod)
+e*ENTHALPY(O2,T=T_prod)+f*ENTHALPY(N2,T=T_prod) "[kJ/kmol]"

PercentEx [%]	a	a_{th}	b	d	e	f	T_{prod} [K]
10	1.921	3	0.07868	3	0.3393	12.41	2191
20	1.97	3	0.03043	3	0.6152	13.54	2093
30	1.988	3	0.01212	3	0.9061	14.66	1996
40	1.995	3	0.004983	3	1.202	15.79	1907
50	1.998	3	0.002111	3	1.501	16.92	1826
60	1.999	3	0.0009184	3	1.8	18.05	1752
70	2	3	0.0004093	3	2.1	19.18	1685
80	2	3	0.0001863	3	2.4	20.3	1625
90	2	3	0.00008644	3	2.7	21.43	1569
100	2	3	0.00004081	3	3	22.56	1518



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16-109 The percent theoretical air required for the combustion of octane such that the volume fraction of CO in the products is less than 0.1% and the heat transfer are to be determined. Also, the percent theoretical air required for 0.1% CO in the products as a function of product pressure is to be plotted.

Analysis The complete combustion reaction equation for excess air is

$$C_8H_{18} + P_{th}a_{th}[O_2 + 3.76N_2] \longrightarrow 8\,CO_2 + 9\,H_2O + (P_{th} - 1)a_{th}\,O_2 + f\,N_2$$

The oxygen balance is

$$P_{th}a_{th} \times 2 = 8 \times 2 + 9 \times 1 + (P_{th} - 1)a_{th} \times 2$$

The reaction equation for excess air and products in equilibrium is

$$C_8H_{18} + P_{th}a_{th}[O_2 + 3.76N_2] \longrightarrow a\,CO_2 + b\,CO + d\,H_2O + e\,O_2 + f\,N_2$$

The coefficients are to be determined from the mass balances

Carbon balance: $8 = a + b$

Hydrogen balance: $18 = 2d \longrightarrow d = 9$

Oxygen balance: $P_{th}a_{th} \times 2 = a \times 2 + b + d + e \times 2$

Nitrogen balance: $P_{th}a_{th} \times 3.76 = f$

Volume fraction of CO must be less than 0.1%. That is,

$$y_{CO} = \frac{b}{N_{tot}} = \frac{b}{a+b+d+e+f} = 0.001$$

The assumed equilibrium reaction is

$$CO_2 \longleftrightarrow CO + 0.5O_2$$

The K_p value of a reaction at a specified temperature can be determined from the Gibbs function data:

$$\bar{g}^*_{CO}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO} = (-53{,}826) - (2000)(258.48) = -570{,}781\ \text{kJ/kmol}$$
$$\bar{g}^*_{O2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{O2} = (59{,}193) - (2000)(268.53) = -477{,}876\ \text{kJ/kmol}$$
$$\bar{g}^*_{CO2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO2} = (-302{,}128) - (2000)(309.00) = -920{,}121\ \text{kJ/kmol}$$

The enthalpies at 2000 K and entropies at 2000 K and 101.3 kPa are obtained from EES. Substituting,

$$\Delta G^*(T_{prod}) = \nu_{CO}\bar{g}^*_{CO}(T_{prod}) + \nu_{O2}\bar{g}^*_{O2}(T_{prod}) - \nu_{CO2}\bar{g}^*_{CO2}(T_{prod})$$
$$= 1(-570{,}781) + 0.5(-477{,}876) - (-920{,}121) = 110{,}402\ \text{kJ/kmol}$$

$$K_p = \exp\left(\frac{-\Delta G^*(T_{prod})}{R_u T_{prod}}\right) = \exp\left(\frac{-110{,}402}{(8.314)(2000)}\right) = 0.001308$$

The equilibrium constant is also given by

$$K_p = \frac{be^{0.5}}{a}\left(\frac{P}{N_{tot}}\right)^{1+0.5-1} = \frac{be^{0.5}}{a}\left(\frac{P_{prod}/101.3}{a+b+d+e+f}\right)^{1+0.5-1}$$

The steady flow energy balance gives

$$H_R = Q_{out} + H_P$$

where

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$$H_R = 1\bar{h}_{\text{C8H18 @ 298 K}} + P_{\text{th}} a_{\text{th}} \bar{h}_{\text{O2 @ 298 K}} + (P_{\text{th}} a_{\text{th}} \times 3.76)\bar{h}_{\text{N2 @ 298 K}}$$
$$= (-208{,}459) + P_{\text{th}} a_{\text{th}} (0) + (P_{\text{th}} a_{\text{th}} \times 3.76)(0) = -208{,}459 \text{ kJ/kmol}$$
$$H_P = a\bar{h}_{\text{CO2 @ 2000 K}} + b\bar{h}_{\text{CO @ 2000 K}} + d\bar{h}_{\text{H2O @ 2000 K}} + e\bar{h}_{\text{O2 @ 2000 K}} + f\,\bar{h}_{\text{N2 @ 2000 K}}$$
$$= a(-302{,}128) + b(-53{,}826) + d(-169{,}171) + e(59{,}193) + f(56{,}115)$$

The enthalpies are obtained from EES. Solving all the equations simultaneously using EES, we obtain

$$P_{\text{th}} = 1.024,\ a_{\text{th}} = 12.5,\ a = 7.935,\ b = 0.06544,\ d = 9,\ e = 0.3289,\ f = 48.11$$
$$\text{PercentTh} = P_{\text{th}} \times 100 = 1.024 \times 100 = \mathbf{102.4\%}$$
$$Q_{\text{out}} = \mathbf{995{,}500\ kJ/kmol\ C_8H_{18}}$$

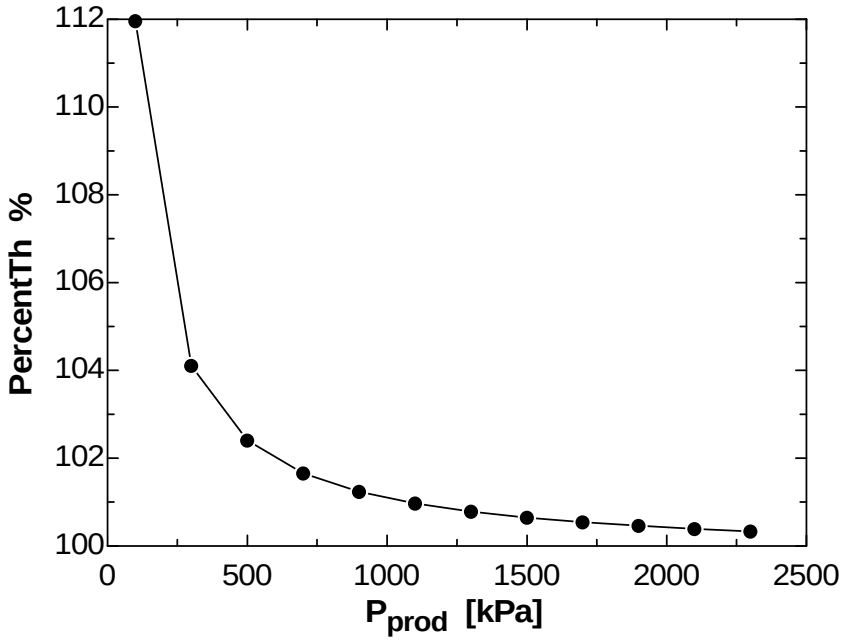
The copy of entire EES solution including parametric studies is given next:

```
"The product temperature is:"
T_prod = 2000 "[K]"
"The reactant temperature is:"
T_reac= 25+273 "[K]"
"PercentTH is  Percent theoretical air"
Pth= PercentTh/100  "Pth = % theoretical air/100"
P_prod = 5 "[atm]" *convert(atm,kPa)"[kPa]"
R_u=8.314 "[kJ/kmol-K]"
"The complete combustion reaction equation for excess air  is:"
"C8H18+ Pth*A_th (O2 +3.76N2)=8 CO2 +  9 H2O +(Pth-1)*A_th O2 + f N2"
"Oxygen Balance for complete combustion:"
 Pth*A_th*2=8*2+9*1 + (Pth-1)*A_th*2
"The reaction equation for excess air and products in equilibrium is:"
"C8H18+ Pth*A_th (O2 +3.76N2)=a CO2 + b CO+ d H2O + e O2 + f N2"
"Carbon Balance:"
8=a + b
"Hydrogen Balance:"
18=2*d
"Oxygen Balance:"
Pth*A_th*2=a*2+b + d + e*2
"Nitrogen Balance:"
Pth*A_th*3.76 = f
N_tot =a +b + d + e + f  "Total kilomoles of products at equilibrium"
"The volume faction of CO in the products is to be less than 0.1%.  For ideal gas mixtures volume fractions equal
mole fractions."
"The mole fraction of CO in the product gases is:"
y_CO = 0.001
y_CO = b/N_tot
"The assumed equilibrium reaction is CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2
"The equilibrium constant is given by Eq. 15-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(b^1*e^0.5)/(a^1)"
sqrt(P/N_tot )*b *sqrt(e )=K_P *a
"The steady-flow energy balance is:"
```

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H_R = Q_out+H_P
H_R=1*ENTHALPY(C8H18,T=T_reac)+Pth*A_th*ENTHALPY(O2,T=T_reac)+Pth*A_th*3.76*ENTHALPY(N2,T=T_reac) "[kJ/kmol]"
H_P=a*ENTHALPY(CO2,T=T_prod)+b*ENTHALPY(CO,T=T_prod)+d*ENTHALPY(H2O,T=T_prod)
+e*ENTHALPY(O2,T=T_prod)+f*ENTHALPY(N2,T=T_prod) "[kJ/kmol]"

P_{prod} [kPa]	PercentTh [%]
100	112
300	104.1
500	102.4
700	101.7
900	101.2
1100	101
1300	100.8
1500	100.6
1700	100.5
1900	100.5
2100	100.4
2300	100.3



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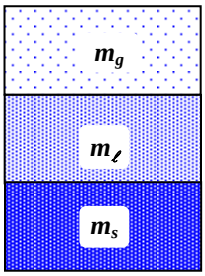
16-110 It is to be shown that when the three phases of a pure substance are in equilibrium, the specific Gibbs function of each phase is the same.

Analysis The total Gibbs function of the three phase mixture of a pure substance can be expressed as

$$G = m_s g_s + m_\ell g_\ell + m_g g_g$$

where the subscripts s, ℓ, and g indicate solid, liquid and gaseous phases. Differentiating by holding the temperature and pressure (thus the Gibbs functions, g) constant yields

$$dG = g_s dm_s + g_\ell dm_\ell + g_g dm_g$$



From conservation of mass,

$$dm_s + dm_\ell + dm_g = 0 \quad \longrightarrow \quad dm_s = -dm_\ell - dm_g$$

Substituting,

$$dG = -g_s(dm_\ell + dm_g) + g_\ell dm_\ell + g_g dm_g$$

Rearranging,

$$dG = (g_\ell - g_s)dm_\ell + (g_g - g_s)dm_g$$

For equilibrium, $dG = 0$. Also dm_ℓ and dm_g can be varied independently. Thus each term on the right hand side must be zero to satisfy the equilibrium criteria. It yields

$$g_\ell = g_s \quad \text{and} \quad g_g = g_s$$

Combining these two conditions gives the desired result,

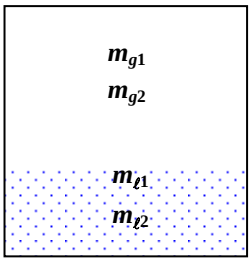
$$g_\ell = g_s = g_s$$

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16-111 It is to be shown that when the two phases of a two-component system are in equilibrium, the specific Gibbs function of each phase of each component is the same.

Analysis The total Gibbs function of the two phase mixture can be expressed as

$$G = (m_{\ell 1} g_{\ell 1} + m_{g1} g_{g1}) + (m_{\ell 2} g_{\ell 2} + m_{g2} g_{g2})$$



where the subscripts ℓ and g indicate liquid and gaseous phases. Differentiating by holding the temperature and pressure (thus the Gibbs functions) constant yields

$$dG = g_{\ell 1} dm_{\ell 1} + g_{g1} dm_{g1} + g_{\ell 2} dm_{\ell 2} + g_{g2} dm_{g2}$$

From conservation of mass,

$$dm_{g1} = -dm_{\ell 1} \quad \text{and} \quad dm_{g2} = -dm_{\ell 2}$$

Substituting,

$$dG = (g_{\ell 1} - g_{g1}) dm_{\ell 1} + (g_{\ell 2} - g_{g2}) dm_{\ell 2}$$

For equilibrium, $dG = 0$. Also $dm_{\ell 1}$ and $dm_{\ell 2}$ can be varied independently. Thus each term on the right hand side must be zero to satisfy the equilibrium criteria. Then we have

$$g_{\ell 1} = g_{g1} \quad \text{and} \quad g_{\ell 2} = g_{g2}$$

which is the desired result.

16-112 Using Henry's law, it is to be shown that the dissolved gases in a liquid can be driven off by heating the liquid.

Analysis Henry's law is expressed as

$$y_{\text{i, liquid side}}(0) = \frac{P_{\text{i, gas side}}(0)}{H}$$

Henry's constant H increases with temperature, and thus the fraction of gas i in the liquid $y_{\text{i,liquid side}}$ decreases. Therefore, heating a liquid will drive off the dissolved gases in a liquid.

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Fundamentals of Engineering (FE) Exam Problems

16-113 If the equilibrium constant for the reaction $H_2 + ½O_2 \rightarrow H_2O$ is K, the equilibrium constant for the reaction $2H_2O \rightarrow 2H_2 + O_2$ at the same temperature is

(a) $1/K$ (b) $1/(2K)$ (c) $2K$ (d) K^2 (e) $1/K^2$

Answer (e) $1/K^2$

16-114 If the equilibrium constant for the reaction $CO + ½O_2 \rightarrow CO_2$ is K, the equilibrium constant for the reaction $CO_2 + 3N_2 \rightarrow CO + ½O_2 + 3N_2$ at the same temperature is

(a) $1/K$ (b) $1/(K + 3)$ (c) $4K$ (d) K (e) $1/K^2$

Answer (a) $1/K$

16-115 The equilibrium constant for the reaction $H_2 + ½O_2 \rightarrow H_2O$ at 1 atm and 1500°C is given to be K. Of the reactions given below, all at 1500°C, the reaction that has a different equilibrium constant is

(a) $H_2 + ½O_2 \rightarrow H_2O$ at 5 atm,
(b) $2H_2 + O_2 \rightarrow 2H_2O$ at 1 atm,
(c) $H_2 + O_2 \rightarrow H_2O + ½O_2$ at 2 atm,
(d) $H_2 + ½O_2 + 3N_2 \rightarrow H_2O + 3N_2$ at 5 atm,
(e) $H_2 + ½O_2 + 3N_2 \rightarrow H_2O + 3N_2$ at 1 atm,

Answer (b) $2H_2 + O_2 \rightarrow 2H_2O$ at 1 atm,

16-116 Of the reactions given below, the reaction whose equilibrium composition at a specified temperature is not affected by pressure is

(a) $H_2 + ½O_2 \rightarrow H_2O$
(b) $CO + ½O_2 \rightarrow CO_2$
(c) $N_2 + O_2 \rightarrow 2NO$
(d) $N_2 \rightarrow 2N$
(e) all of the above.

Answer (c) $N_2 + O_2 \rightarrow 2NO$

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16-117 Of the reactions given below, the reaction whose number of moles of products increases by the addition of inert gases into the reaction chamber at constant pressure and temperature is

(a) $H_2 + ½O_2 \rightarrow H_2O$
(b) $CO + ½O_2 \rightarrow CO_2$
(c) $N_2 + O_2 \rightarrow 2NO$
(d) $N_2 \rightarrow 2N$
(e) none of the above.

Answer (d) $N_2 \rightarrow 2N$

16-118 Moist air is heated to a very high temperature. If the equilibrium composition consists of H_2O, O_2, N_2, OH, H_2, and NO, the number of equilibrium constant relations needed to determine the equilibrium composition of the mixture is

(a) 1 (b) 2 (c) 3 (d) 4 (e) 5

Answer (c) 3

16-119 Propane C_3H_8 is burned with air, and the combustion products consist of CO_2, CO, H_2O, O_2, N_2, OH, H_2, and NO. The number of equilibrium constant relations needed to determine the equilibrium composition of the mixture is

(a) 1 (b) 2 (c) 3 (d) 4 (e) 5

Answer (d) 4

16-120 Consider a gas mixture that consists of three components. The number of independent variables that need to be specified to fix the state of the mixture is

(a) 1 (b) 2 (c) 3 (d) 4 (e) 5

Answer (d) 4

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16-121 The value of Henry's constant for CO_2 gas dissolved in water at 290 K is 12.8 MPa. Consider water exposed to air at 100 kPa that contains 3 percent CO_2 by volume. Under phase equilibrium conditions, the mole fraction of CO_2 gas dissolved in water at 290 K is

(a) 2.3×10^{-4} (b) 3.0×10^{-4} (c) 0.80×10^{-4} (d) 2.2×10^{-4} (e) 5.6×10^{-4}

Answer (a) 2.3×10^{-4}

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
H=12.8 "MPa"
P=0.1 "MPa"
y_CO2_air=0.03
P_CO2_air=y_CO2_air*P
y_CO2_liquid=P_CO2_air/H

"Some Wrong Solutions with Common Mistakes:"
W1_yCO2=P_CO2_air*H "Multiplying by H instead of dividing by it"
W2_yCO2=P_CO2_air "Taking partial pressure in air"
```

16-122 The solubility of nitrogen gas in rubber at 25°C is 0.00156 kmol/m^3·bar. When phase equilibrium is established, the density of nitrogen in a rubber piece placed in a nitrogen gas chamber at 300 kPa is

(a) 0.005 kg/m^3 (b) 0.018 kg/m^3 (c) 0.047 kg/m^3 (d) 0.13 kg/m^3 (e) 0.28 kg/m^3

Answer (d) 0.13 kg/m^3

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=25 "C"
S=0.00156 "kmol/bar.m^3"
MM_N2=28 "kg/kmol"
S_mass=S*MM_N2 "kg/bar.m^3"
P_N2=3 "bar"
rho_solid=S_mass*P_N2

"Some Wrong Solutions with Common Mistakes:"
W1_density=S*P_N2 "Using solubility per kmol"
```

16-123 … 16-125 Design and Essay Problems

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Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 17
COMPRESSIBLE FLOW

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Stagnation Properties

17-1C **No, there is not significant error**, because the velocities encountered in air-conditioning applications are very low, and thus the static and the stagnation temperatures are practically identical.

Discussion If the air stream were supersonic, however, the error would indeed be significant.

17-2C *Stagnation enthalpy* **combines the ordinary enthalpy and the kinetic energy of a fluid**, and offers convenience when analyzing high-speed flows. **It differs from the ordinary enthalpy by the kinetic energy term**.

Discussion Most of the time, we mean *specific enthalpy*, i.e., enthalpy per unit mass, when we use the term *enthalpy*.

17-3C *Dynamic temperature* is **the temperature rise of a fluid during a stagnation process**.

Discussion When a gas decelerates from high speed to zero speed at a stagnation point, the temperature of the gas rises.

17-4C The temperature of the air **rises as it approaches the nose because of the stagnation process**.

Discussion In the frame of reference moving with the aircraft, the air decelerates from high speed to zero at the nose (stagnation point), and this causes the air temperature to rise.

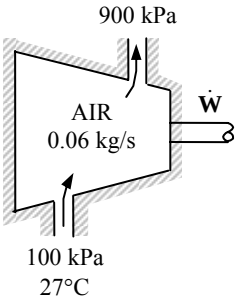
17-5 The inlet stagnation temperature and pressure and the exit stagnation pressure of air flowing through a compressor are specified. The power input to the compressor is to be determined.

Assumptions **1** The compressor is isentropic. **2** Air is an ideal gas.

Properties The properties of air at room temperature are $c_p = 1.005$ kJ/kg·K and $k = 1.4$.

Analysis The exit stagnation temperature of air T_{02} is determined from

$$T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{(k-1)/k} = (300.2\text{ K})\left(\frac{900}{100}\right)^{(1.4-1)/1.4} = 562.4\text{ K}$$



From the energy balance on the compressor,

$$\dot{W}_{\text{in}} = \dot{m}(h_{20} - h_{01})$$

or,

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_{02} - T_{01}) = (0.06\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(562.4 - 300.2)\text{K} = \mathbf{15.8\ kW}$$

Discussion Note that the stagnation properties can be used conveniently in the energy equation.

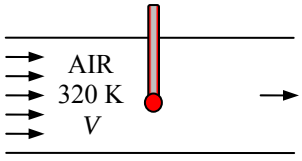
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17-6 Air at 320 K is flowing in a duct. The temperature that a stationary probe inserted into the duct will read is to be determined for different air velocities.

Assumptions The stagnation process is isentropic.

Properties The specific heat of air at room temperature is c_p = 1.005 kJ/kg·K.

Analysis The air which strikes the probe will be brought to a complete stop, and thus it will undergo a stagnation process. The thermometer will sense the temperature of this stagnated air, which is the stagnation temperature, T_0. It is determined from $T_0 = T + \dfrac{V^2}{2c_p}$. The results for each case are calculated below:



(*a*) $$T_0 = 320\,\text{K} + \frac{(1\,\text{m/s})^2}{2\times 1.005\,\text{kJ/kg}\cdot\text{K}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{320.0\ K}$$

(*b*) $$T_0 = 320\,\text{K} + \frac{(10\,\text{m/s})^2}{2\times 1.005\,\text{kJ/kg}\cdot\text{K}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{320.1 K}$$

(*c*) $$T_0 = 320\,\text{K} + \frac{(100\,\text{m/s})^2}{2\times 1.005\,\text{kJ/kg}\cdot\text{K}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{325.0\ K}$$

(*d*) $$T_0 = 320\,\text{K} + \frac{(1000\,\text{m/s})^2}{2\times 1.005\,\text{kJ/kg}\cdot\text{K}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{817.5\ K}$$

Discussion Note that the stagnation temperature is nearly identical to the thermodynamic temperature at low velocities, but the difference between the two is significant at high velocities.

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17-7 The states of different substances and their velocities are specified. The stagnation temperature and stagnation pressures are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Helium and nitrogen are ideal gases.

Analysis (*a*) Helium can be treated as an ideal gas with c_p = 5.1926 kJ/kg·K and k = 1.667. Then the stagnation temperature and pressure of helium are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 50°\text{C} + \frac{(240\ \text{m/s})^2}{2\times 5.1926\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{55.5°C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.25\ \text{MPa})\left(\frac{328.7\ \text{K}}{323.2\ \text{K}}\right)^{1.667/(1.667-1)} = \mathbf{0.261\ MPa}$$

(*b*) Nitrogen can be treated as an ideal gas with c_p = 1.039 kJ/kg·K and k =1.400. Then the stagnation temperature and pressure of nitrogen are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 50°\text{C} + \frac{(300\ \text{m/s})^2}{2\times 1.039\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{93.3°C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.15\ \text{MPa})\left(\frac{366.5\ \text{K}}{323.2\ \text{K}}\right)^{1.4/(1.4-1)} = \mathbf{0.233\ MPa}$$

(*c*) Steam can be treated as an ideal gas with c_p = 1.865 kJ/kg·K and k =1.329. Then the stagnation temperature and pressure of steam are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 350°\text{C} + \frac{(480\ \text{m/s})^2}{2\times 1.865\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{411.8°C = 685\ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.1\ \text{MPa})\left(\frac{685\ \text{K}}{623.2\ \text{K}}\right)^{1.329/(1.329-1)} = \mathbf{0.147\ MPa}$$

Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

17-8 The state of air and its velocity are specified. The stagnation temperature and stagnation pressure of air are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Air is an ideal gas.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4.

Analysis The stagnation temperature of air is determined from

$$T_0 = T + \frac{V^2}{2c_p} = 238\ \text{K} + \frac{(470\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 347.9\ \text{K} \cong \mathbf{348\ K}$$

Other stagnation properties at the specified state are determined by considering an isentropic process between the specified state and the stagnation state,

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (36\ \text{kPa})\left(\frac{347.9\ \text{K}}{238\ \text{K}}\right)^{1.4/(1.4-1)} = 135.9\ \text{kPa} \cong \mathbf{136\ kPa}$$

Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

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17-9E Steam flows through a device. The stagnation temperature and pressure of steam and its velocity are specified. The static pressure and temperature of the steam are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Steam is an ideal gas.

Properties Steam can be treated as an ideal gas with c_p = 0.4455 Btu/lbm·R and k =1.329.

Analysis The static temperature and pressure of steam are determined from

$$T = T_0 - \frac{V^2}{2c_p} = 700°\text{F} - \frac{(900\ \text{ft/s})^2}{2\times 0.4455\ \text{Btu/lbm}\cdot°\text{F}}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right) = \mathbf{663.7°F}$$

$$P = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (120\ \text{psia})\left(\frac{1123.7\ \text{R}}{1160\ \text{R}}\right)^{1.329/(1.329-1)} = \mathbf{105.5\ psia}$$

Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

17-10 Air flows through a device. The stagnation temperature and pressure of air and its velocity are specified. The static pressure and temperature of air are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Air is an ideal gas.

Properties The properties of air at an anticipated average temperature of 600 K are c_p = 1.051 kJ/kg·K and k = 1.376.

Analysis The static temperature and pressure of air are determined from

$$T = T_0 - \frac{V^2}{2c_p} = 673.2 - \frac{(570\ \text{m/s})^2}{2\times 1.051\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{518.6\ K}$$

and

$$P_2 = P_{02}\left(\frac{T_2}{T_{02}}\right)^{k/(k-1)} = (0.6\ \text{MPa})\left(\frac{518.6\ \text{K}}{673.2\ \text{K}}\right)^{1.376/(1.376-1)} = \mathbf{0.23\ MPa}$$

Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

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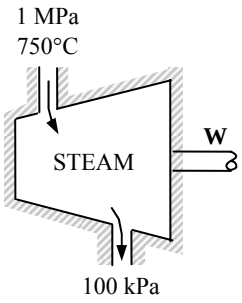
17-11 The inlet stagnation temperature and pressure and the exit stagnation pressure of products of combustion flowing through a gas turbine are specified. The power output of the turbine is to be determined.

Assumptions **1** The expansion process is isentropic. **2** Products of combustion are ideal gases.

Properties The properties of products of combustion are c_p = 1.157 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.33.

Analysis The exit stagnation temperature T_{02} is determined to be

$$T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{(k-1)/k} = (1023.2\ \text{K})\left(\frac{0.1}{1}\right)^{(1.33-1)/1.33} = 577.9\ \text{K}$$



Also,

$$c_p = kc_v = k\left(c_p - R\right) \quad \longrightarrow \quad c_p = \frac{kR}{k-1} = \frac{1.33\left(0.287\ \text{kJ/kg}\cdot\text{K}\right)}{1.33-1} = 1.157\ \text{kJ/kg}\cdot\text{K}$$

From the energy balance on the turbine,

$$-w_{\text{out}} = (h_{20} - h_{01})$$

or, $$w_{\text{out}} = c_p\left(T_{01} - T_{02}\right) = \left(1.157\ \text{kJ/kg}\cdot\text{K}\right)(1023.2 - 577.9)\ \text{K} = 515.2\ \text{kJ/kg} \cong \mathbf{515\ kJ/kg}$$

Discussion Note that the stagnation properties can be used conveniently in the energy equation.

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Speed of Sound and Mach Number

17-12C *Sound* is an **infinitesimally small pressure wave**. It is **generated by a small disturbance in a medium**. It **travels by wave propagation**. **Sound waves cannot travel in a vacuum**.

Discussion Electromagnetic waves, like light and radio waves, can travel in a vacuum, but sound cannot.

17-13C **Yes, the propagation of sound waves is nearly isentropic**. Because the amplitude of an ordinary sound wave is very small, and it does not cause any significant change in temperature and pressure.

Discussion No process is truly isentropic, but the increase of entropy due to sound propagation is negligibly small.

17-14C The *sonic speed* in a medium **depends on the properties of the medium, and it changes as the properties of the medium change**.

Discussion The most common example is the change in speed of sound due to temperature change.

17-15C Sound travels faster in **warm (higher temperature) air** since $c = \sqrt{kRT}$.

Discussion On the microscopic scale, we can imagine the air molecules moving around at higher speed in warmer air, leading to higher propagation of disturbances.

17-16C **Sound travels fastest in helium**, since $c = \sqrt{kRT}$ and helium has the highest kR value. It is about 0.40 for air, 0.35 for argon, and 3.46 for helium.

Discussion We are assuming, of course, that these gases behave as ideal gases – a good approximation at room temperature.

17-17C Air at specified conditions will behave like an ideal gas, and *the speed of sound in an ideal gas depends on temperature only*. Therefore, **the speed of sound is the same in both mediums**.

Discussion If the temperature were different, however, the speed of sound would be different.

17-18C In general, **no**, because the Mach number also depends on the speed of sound in gas, which depends on the temperature of the gas. **The Mach number remains constant only if the temperature *and* the velocity are constant**.

Discussion It turns out that the speed of sound is not a strong function of pressure. In fact, it is not a function of pressure at all for an ideal gas.

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17-19 The Mach number of an aircraft and the speed of sound in air are to be determined at two specified temperatures.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K. Its specific heat ratio at room temperature is $k = 1.4$.

Analysis From the definitions of the speed of sound and the Mach number,

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\,\text{kJ/kg}\cdot\text{K})(300\,\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = \mathbf{347\ m/s}$$

and $$\text{Ma} = \frac{V}{c} = \frac{240\,\text{m/s}}{347\,\text{m/s}} = \mathbf{0.692}$$

(*b*) At 1000 K,

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\,\text{kJ/kg}\cdot\text{K})(1000\,\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = \mathbf{634\ m/s}$$

and $$\text{Ma} = \frac{V}{c} = \frac{240\,\text{m/s}}{634\,\text{m/s}} = \mathbf{0.379}$$

Discussion Note that a constant Mach number does not necessarily indicate constant speed. The Mach number of a rocket, for example, will be increasing even when it ascends at constant speed. Also, the specific heat ratio k changes with temperature, and the accuracy of the result at 1000 K can be improved by using the k value at that temperature (it would give $k = 1.386$, $c = 619$ m/s, and Ma = 0.388).

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17-20 Carbon dioxide flows through a nozzle. The inlet temperature and velocity and the exit temperature of CO_2 are specified. The Mach number is to be determined at the inlet and exit of the nozzle.

Assumptions **1** CO_2 is an ideal gas with constant specific heats at room temperature. **2** This is a steady-flow process.

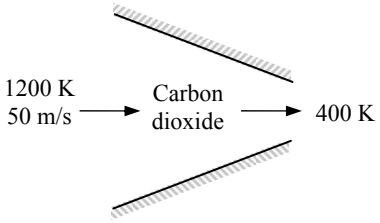
Properties The gas constant of carbon dioxide is $R = 0.1889$ kJ/kg·K. Its constant pressure specific heat and specific heat ratio at room temperature are $c_p = 0.8439$ kJ/kg·K and $k = 1.288$.

Analysis (*a*) At the inlet

$$c_1 = \sqrt{k_1 RT_1} = \sqrt{(1.288)(0.1889\,\text{kJ/kg}\cdot\text{K})(1200\,\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 540.3\,\text{m/s}$$

Thus,

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{50\,\text{m/s}}{540.3\,\text{m/s}} = \mathbf{0.0925}$$



(*b*) At the exit,

$$c_2 = \sqrt{k_2 RT_2} = \sqrt{(1.288)(0.1889\,\text{kJ/kg}\cdot\text{K})(400\,\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 312.0\ \text{m/s}$$

The nozzle exit velocity is determined from the steady-flow energy balance relation,

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \quad \rightarrow \quad 0 = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

$$0 = (0.8439\,\text{kJ/kg}\cdot\text{K})(400 - 1200\,\text{K}) + \frac{V_2^2 - (50\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) \longrightarrow V_2 = 1163\,\text{m/s}$$

Thus,

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{1163\,\text{m/s}}{312\,\text{m/s}} = \mathbf{3.73}$$

Discussion The specific heats and their ratio *k* change with temperature, and the accuracy of the results can be improved by accounting for this variation. Using EES (or another property database):

At 1200 K: $c_p = 1.278$ kJ/kg·K, $k = 1.173$ $\rightarrow$ $c_1 = 516$ m/s, $V_1 = 50$ m/s, $\text{Ma}_1 = 0.0969$

At 400 K: $c_p = 0.9383$ kJ/kg·K, $k = 1.252$ $\rightarrow$ $c_2 = 308$ m/s, $V_2 = 1356$ m/s, $\text{Ma}_2 = 4.41$

Therefore, the constant specific heat assumption results in an error of **4.5%** at the inlet and **15.5%** at the exit in the Mach number, which are significant.

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17-21 Nitrogen flows through a heat exchanger. The inlet temperature, pressure, and velocity and the exit pressure and velocity are specified. The Mach number is to be determined at the inlet and exit of the heat exchanger.

Assumptions **1** N_2 is an ideal gas. **2** This is a steady-flow process. **3** The potential energy change is negligible.

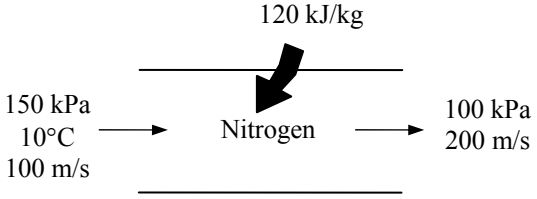
Properties The gas constant of N_2 is $R = 0.2968$ kJ/kg·K. Its constant pressure specific heat and specific heat ratio at room temperature are $c_p = 1.040$ kJ/kg·K and $k = 1.4$.

Analysis

$$c_1 = \sqrt{k_1 RT_1} = \sqrt{(1.400)(0.2968\ \text{kJ/kg}\cdot\text{K})(283\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 342.9\ \text{m/s}$$

Thus,

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{100\ \text{m/s}}{342.9\ \text{m/s}} = \mathbf{0.292}$$



From the energy balance on the heat exchanger,

$$q_{\text{in}} = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

$$120\ \text{kJ/kg} = (1.040\ \text{kJ/kg.°C})(T_2 - 10°\text{C}) + \frac{(200\ \text{m/s})^2 - (100\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

It yields

$$T_2 = 111°\text{C} = 384\ \text{K}$$

$$c_2 = \sqrt{k_2 RT_2} = \sqrt{(1.4)(0.2968\ \text{kJ/kg}\cdot\text{K})(384\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 399\ \text{m/s}$$

Thus,

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{200\ \text{m/s}}{399\ \text{m/s}} = \mathbf{0.501}$$

Discussion The specific heats and their ratio k change with temperature, and the accuracy of the results can be improved by accounting for this variation. Using EES (or another property database):

At 10°C : $c_p = 1.038$ kJ/kg·K, $k = 1.400$ → $c_1 = 343$ m/s, $V_1 = 100$ m/s, $\text{Ma}_1 = 0.292$

At 111°C $c_p = 1.041$ kJ/kg·K, $k = 1.399$ → $c_2 = 399$ m/s, $V_2 = 200$ m/s, $\text{Ma}_2 = 0.501$

Therefore, the constant specific heat assumption results in an error of **4.5%** at the inlet and **15.5%** at the exit in the Mach number, which are almost identical to the values obtained assuming constant specific heats.

17-22 The speed of sound in refrigerant-134a at a specified state is to be determined.

Assumptions R-134a is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of R-134a is $R = 0.08149$ kJ/kg·K. Its specific heat ratio at room temperature is $k = 1.108$.

Analysis From the ideal-gas speed of sound relation,

$$c = \sqrt{kRT} = \sqrt{(1.108)(0.08149\ \text{kJ/kg}\cdot\text{K})(60 + 273\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{173\ m/s}$$

Discusion Note that the speed of sound is independent of pressure for ideal gases.

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17-23 The Mach number of a passenger plane for specified limiting operating conditions is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K. Its specific heat ratio at room temperature is $k = 1.4$.

Analysis From the speed of sound relation

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(-60+273\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 293\ \text{m/s}$$

Thus, the Mach number corresponding to the maximum cruising speed of the plane is

$$\text{Ma} = \frac{V_{\max}}{c} = \frac{(945/3.6)\ \text{m/s}}{293\ \text{m/s}} = \mathbf{0.897}$$

Discussion Note that this is a subsonic flight since Ma < 1. Also, using a k value at -60°C would give practically the same result.

17-24E Steam flows through a device at a specified state and velocity. The Mach number of steam is to be determined assuming ideal gas behavior.

Assumptions Steam is an ideal gas with constant specific heats.

Properties The gas constant of steam is $R = 0.1102$ Btu/lbm·R. Its specific heat ratio is given to be $k = 1.3$.

Analysis From the ideal-gas speed of sound relation,

$$c = \sqrt{kRT} = \sqrt{(1.3)(0.1102\ \text{Btu/lbm}\cdot\text{R})(1160\ \text{R})\left(\frac{25{,}037\ \text{ft}^2/\text{s}^2}{1\ \text{Btu/lbm}}\right)} = 2040\ \text{ft/s}$$

Thus,

$$\text{Ma} = \frac{V}{c} = \frac{900\ \text{ft/s}}{2040\ \text{ft/s}} = \mathbf{0.441}$$

Discussion Using property data from steam tables and not assuming ideal gas behavior, it can be shown that the Mach number in steam at the specified state is 0.446, which is sufficiently close to the ideal-gas value of 0.441. Therefore, the ideal gas approximation is a reasonable one in this case.

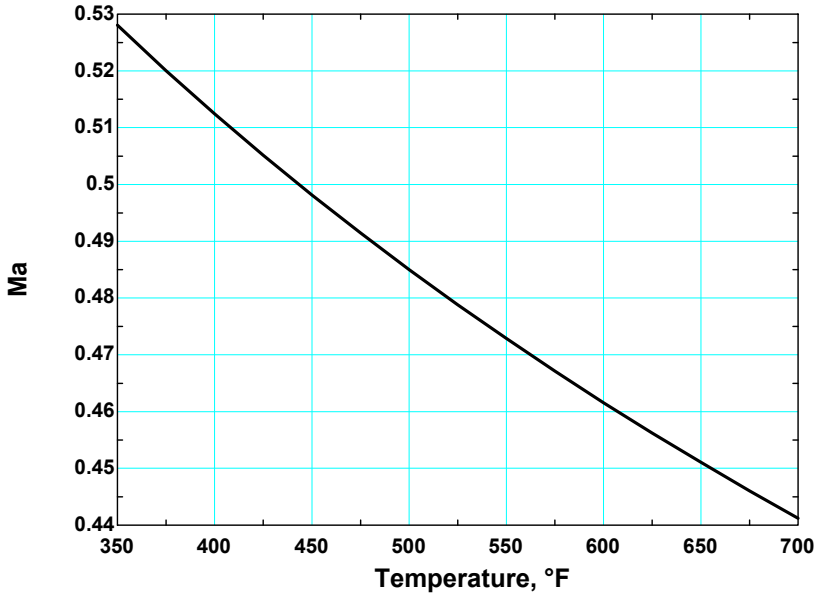
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17-25E Problem 17-24E is reconsidered. The variation of Mach number with temperature as the temperature changes between 350° and 700°F is to be investigated, and the results are to be plotted.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
T=Temperature+460
R=0.1102
V=900
k=1.3
c=SQRT(k*R*T*25037)
Ma=V/c
```

Temperature, T, °F	Mach number Ma
350	0.528
375	0.520
400	0.512
425	0.505
450	0.498
475	0.491
500	0.485
525	0.479
550	0.473
575	0.467
600	0.462
625	0.456
650	0.451
675	0.446
700	0.441



Discussion Note that for a specified flow speed, the Mach number decreases with increasing temperature, as expected.

17-26 The expression for the speed of sound for an ideal gas is to be obtained using the isentropic process equation and the definition of the speed of sound.

Analysis The isentropic relation $Pv^k = A$ where A is a constant can also be expressed as

$$P = A\left(\frac{1}{v}\right)^k = A\rho^k$$

Substituting it into the relation for the speed of sound,

$$c^2 = \left(\frac{\partial P}{\partial \rho}\right)_s = \left(\frac{\partial (A\rho)^k}{\partial \rho}\right)_s = kA\rho^{k-1} = k(A\rho^k)/\rho = k(P/\rho) = kRT$$

since for an ideal gas $P = \rho RT$ or $RT = P/\rho$. Therefore, $\boxed{c = \sqrt{kRT}}$, which is the desired relation.

Discussion Notice that pressure has dropped out; the speed of sound in an ideal gas is *not* a function of pressure.

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17-27 The inlet state and the exit pressure of air are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are $R = 0.287$ kJ/kg·K and $k = 1.4$. The specific heat ratio k varies with temperature, but in our case this change is very small and can be disregarded.

Analysis The final temperature of air is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (333.2\text{ K})\left(\frac{0.4\text{ MPa}}{1.5\text{ MPa}}\right)^{(1.4-1)/1.4} = 228.4\text{ K}$$

Treating k as a constant, the ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1 RT_1}}{\sqrt{k_2 RT_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{333.2}}{\sqrt{228.4}} = \mathbf{1.21}$$

Discussion Note that the speed of sound is proportional to the square root of thermodynamic temperature.

17-28 The inlet state and the exit pressure of helium are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

Assumptions Helium is an ideal gas with constant specific heats at room temperature.

Properties The properties of helium are $R = 2.0769$ kJ/kg·K and $k = 1.667$.

Analysis The final temperature of helium is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (333.2\text{ K})\left(\frac{0.4}{1.5}\right)^{(1.667-1)/1.667} = 196.3\text{ K}$$

The ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1 RT_1}}{\sqrt{k_2 RT_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{333.2}}{\sqrt{196.3}} = \mathbf{1.30}$$

Discussion Note that the speed of sound is proportional to the square root of thermodynamic temperature.

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17-29E The inlet state and the exit pressure of air are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are $R = 0.06855$ Btu/lbm·R and $k = 1.4$. The specific heat ratio k varies with temperature, but in our case this change is very small and can be disregarded.

Analysis The final temperature of air is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (659.7\text{ R})\left(\frac{60}{170}\right)^{(1.4-1)/1.4} = 489.9\text{ R}$$

Treating k as a constant, the ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1 R T_1}}{\sqrt{k_2 R T_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{659.7}}{\sqrt{489.9}} = \mathbf{1.16}$$

Discussion Note that the speed of sound is proportional to the square root of thermodynamic temperature.

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One Dimensional Isentropic Flow

17-30C (*a*) The **velocity increases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid decrease**.

Discussion The velocity increase is opposite to what happens in supersonic flow.

17-31C (*a*) The **velocity decreases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid increase**.

Discussion The velocity decrease is opposite to what happens in supersonic flow.

17-32C (*a*) The **exit velocity remains constant at sonic speed**, (*b*) the **mass flow rate through the nozzle decreases because of the reduced flow area**.

Discussion Without a diverging portion of the nozzle, a converging nozzle is limited to sonic velocity at the exit.

17-33C (*a*) The **velocity decreases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid increase**.

Discussion The velocity decrease is opposite to what happens in subsonic flow.

17-34C (*a*) The **velocity increases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid decrease**.

Discussion The velocity increase is opposite to what happens in subsonic flow.

17-35C **The pressures at the two throats are identical**.

Discussion Since the gas has the same stagnation conditions, it also has the same sonic conditions at the throat.

17-36C **No**, it is not possible.

Discussion The only way to do it is to have first a converging nozzle, and then a diverging nozzle.

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17-37 The Mach number of scramjet and the air temperature are given. The speed of the engine is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K. Its specific heat ratio at room temperature is $k = 1.4$.

Analysis The temperature is -20 + 273.15 = 253.15 K. The speed of sound is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(253.15\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 318.93\ \text{m/s}$$

and

$$V = c\text{Ma} = (318.93\ \text{m/s})(7)\left(\frac{3.6\ \text{km/h}}{1\ \text{m/s}}\right) = 8037\ \text{km/h} \cong \mathbf{8040\ km/h}$$

Discussion Note that extremely high speed can be achieved with scramjet engines. We cannot justify more than three significant digits in a problem like this.

17-38E The Mach number of scramjet and the air temperature are given. The speed of the engine is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $R = 0.06855$ Btu/lbm·R. Its specific heat ratio at room temperature is $k = 1.4$.

Analysis The temperature is 0 + 459.67 = 459.67 R. The speed of sound is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.06855\ \text{Btu/lbm}\cdot\text{R})(459.67\ \text{R})\left(\frac{25{,}037\ \text{ft}^2/\text{s}^2}{1\ \text{Btu/lbm}}\right)} = 1050.95\ \text{ft/s}$$

and

$$V = c\text{Ma} = (1050.95\ \text{ft/s})(7)\left(\frac{1\ \text{mi/h}}{1.46667\ \text{ft/s}}\right) = 5015.9\ \text{mi/h} \cong \mathbf{5020\ mi/h}$$

Discussion Note that extremely high speed can be achieved with scramjet engines. We cannot justify more than three significant digits in a problem like this.

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17-39 The speed of an airplane and the air temperature are give. It is to be determined if the speed of this airplane is subsonic or supersonic.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K. Its specific heat ratio at room temperature is $k = 1.4$.

Analysis The temperature is -50 + 273.15 = 223.15 K. The speed of sound is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(223.15\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)\left(\frac{3.6\ \text{km/h}}{1\ \text{m/s}}\right)} = 1077.97\ \text{km/h}$$

and

$$\text{Ma} = \frac{V}{c} = \frac{920\ \text{km/h}}{1077.97\ \text{km/h}} = 0.85346 \cong \mathbf{0.853}$$

The speed of the airplane is **subsonic** since the Mach number is less than 1.

Discussion Subsonic airplanes stay sufficiently far from the Mach number of 1 to avoid the instabilities associated with transonic flights.

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17-40 The critical temperature, pressure, and density of air and helium are to be determined at specified conditions.

Assumptions Air and Helium are ideal gases with constant specific heats at room temperature.

Properties The properties of air at room temperature are $R = 0.287$ kJ/kg·K, $k = 1.4$, and $c_p = 1.005$ kJ/kg·K. The properties of helium at room temperature are $R = 2.0769$ kJ/kg·K, $k = 1.667$, and $c_p = 5.1926$ kJ/kg·K.

Analysis (*a*) Before we calculate the critical temperature T^*, pressure P^*, and density ρ^*, we need to determine the stagnation temperature T_0, pressure P_0, and density ρ_0.

$$T_0 = 100°\text{C} + \frac{V^2}{2c_p} = 100 + \frac{(250\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 131.1°\text{C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\ \text{kPa})\left(\frac{404.3\ \text{K}}{373.2\ \text{K}}\right)^{1.4/(1.4-1)} = 264.7\ \text{kPa}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{264.7\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(404.3\ \text{K})} = 2.281\ \text{kg/m}^3$$

Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (404.3\ \text{K})\left(\frac{2}{1.4+1}\right) = \mathbf{337\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (264.7\ \text{kPa})\left(\frac{2}{1.4+1}\right)^{1.4/(1.4-1)} = \mathbf{140\ kPa}$$

$$\rho^* = \rho_0\left(\frac{2}{k+1}\right)^{1/(k-1)} = (2.281\ \text{kg/m}^3)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)} = \mathbf{1.45\ kg/m^3}$$

(*b*) For helium, $T_0 = T + \frac{V^2}{2c_p} = 40 + \frac{(300\ \text{m/s})^2}{2\times 5.1926\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 48.7°\text{C}$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\ \text{kPa})\left(\frac{321.9\ \text{K}}{313.2\ \text{K}}\right)^{1.667/(1.667-1)} = 214.2\ \text{kPa}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{214.2\ \text{kPa}}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(321.9\ \text{K})} = 0.320\ \text{kg/m}^3$$

Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (321.9\ \text{K})\left(\frac{2}{1.667+1}\right) = \mathbf{241\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (214.2\ \text{kPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{104.3\ kPa}$$

$$\rho^* = \rho_0\left(\frac{2}{k+1}\right)^{1/(k-1)} = (0.320\ \text{kg/m}^3)\left(\frac{2}{1.667+1}\right)^{1/(1.667-1)} = \mathbf{0.208\ kg/m^3}$$

Discussion These are the temperature, pressure, and density values that will occur at the throat when the flow past the throat is supersonic.

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17-41 Quiescent carbon dioxide at a given state is accelerated isentropically to a specified Mach number. The temperature and pressure of the carbon dioxide after acceleration are to be determined.

Assumptions Carbon dioxide is an ideal gas with constant specific heats at room temperature.

Properties The specific heat ratio of the carbon dioxide at room temperature is $k = 1.288$.

Analysis The inlet temperature and pressure in this case is equivalent to the stagnation temperature and pressure since the inlet velocity of the carbon dioxide is said to be negligible. That is, $T_0 = T_i = 400$ K and $P_0 = P_i = 1200$ kPa. Then,

$$T = T_0\left(\frac{2}{2+(k-1)\text{Ma}^2}\right) = (600\text{ K})\left(\frac{2}{2+(1.288\text{-}1)(0.6)^2}\right) = 570.43\text{ K} \cong \mathbf{570\ K}$$

and

$$P = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (1200\text{ kPa})\left(\frac{570.43\text{ K}}{600\text{ K}}\right)^{1.288/(1.288-1)} = 957.23\text{ K} \cong \mathbf{957\ kPa}$$

Discussion Note that both the pressure and temperature drop as the gas is accelerated as part of the internal energy of the gas is converted to kinetic energy.

17-42 Air enters a converging-diverging nozzle at specified conditions. The lowest pressure that can be obtained at the throat of the nozzle is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio of air at room temperature is $k = 1.4$.

Analysis The lowest pressure that can be obtained at the throat is the critical pressure P^*, which is determined from

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (800\text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/(1.4-1)} = \mathbf{423\ kPa}$$

Discussion This is the pressure that occurs at the throat when the flow past the throat is supersonic.

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17-43 Helium enters a converging-diverging nozzle at specified conditions. The lowest temperature and pressure that can be obtained at the throat of the nozzle are to be determined.

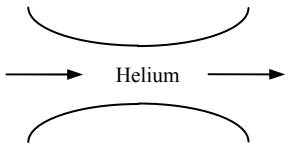
Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of helium are $k = 1.667$ and $c_p = 5.1926$ kJ/kg·K.

Analysis The lowest temperature and pressure that can be obtained at the throat are the critical temperature T^* and critical pressure P^*. First we determine the stagnation temperature T_0 and stagnation pressure P_0,

$$T_0 = T + \frac{V^2}{2c_p} = 800\,\text{K} + \frac{(100\,\text{m/s})^2}{2\times 5.1926\,\text{kJ/kg}\cdot\text{°C}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 801\,\text{K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.7\,\text{MPa})\left(\frac{801\,\text{K}}{800\,\text{K}}\right)^{1.667/(1.667-1)} = 0.702\,\text{MPa}$$



Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (801\,\text{K})\left(\frac{2}{1.667+1}\right) = \mathbf{601\ K}$$

and

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.702\,\text{MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.342\ MPa}$$

Discussion These are the temperature and pressure that will occur at the throat when the flow past the throat is supersonic.

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17-44 Air flows through a duct. The state of the air and its Mach number are specified. The velocity and the stagnation pressure, temperature, and density of the air are to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air at room temperature are $R = 0.287$ kPa.m^3/kg.K and $k = 1.4$.

Analysis The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(373.2\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 387.2\ \text{m/s}$$

Thus,

$$V = \text{Ma} \times c = (0.8)(387.2\ \text{m/s}) = \mathbf{310\ m/s}$$



Also,

$$\rho = \frac{P}{RT} = \frac{200\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(373.2\ \text{K})} = 1.867\ \text{kg/m}^3$$

Then the stagnation properties are determined from

$$T_0 = T\left(1 + \frac{(k-1)\text{Ma}^2}{2}\right) = (373.2\ \text{K})\left(1 + \frac{(1.4\text{-}1)(0.8)^2}{2}\right) = \mathbf{421\ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\ \text{kPa})\left(\frac{421.0\ \text{K}}{373.2\ \text{K}}\right)^{1.4/(1.4-1)} = \mathbf{305\ kPa}$$

$$\rho_0 = \rho\left(\frac{T_0}{T}\right)^{1/(k-1)} = (1.867\ \text{kg/m}^3)\left(\frac{421.0\ \text{K}}{373.2\ \text{K}}\right)^{1/(1.4-1)} = \mathbf{2.52\ kg/m^3}$$

Discussion Note that both the pressure and temperature drop as the gas is accelerated as part of the internal energy of the gas is converted to kinetic energy.

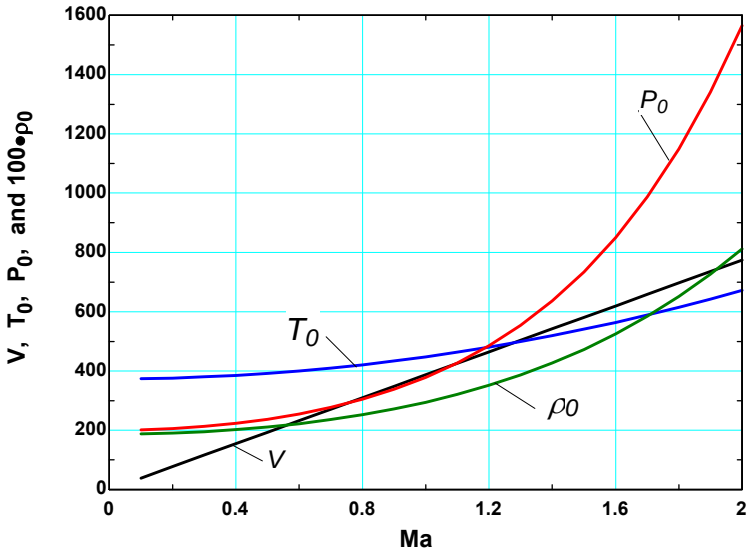
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17-45 Problem 17-44 is reconsidered. The effect of Mach number on the velocity and stagnation properties as the Ma is varied from 0.1 to 2 are to be investigated, and the results are to be plotted.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
P=200
T=100+273.15
R=0.287
k=1.4
c=SQRT(k*R*T*1000)
Ma=V/c
rho=P/(R*T)

"Stagnation properties"
T0=T*(1+(k-1)*Ma^2/2)
P0=P*(T0/T)^(k/(k-1))
rho0=rho*(T0/T)^(1/(k-1))
```



Mach num. Ma	Velocity, V, m/s	Stag. Temp, T_0, K	Stag. Press, P_0, kPa	Stag. Density, ρ_0, kg/m^3
0.1	38.7	373.9	201.4	1.877
0.2	77.4	376.1	205.7	1.905
0.3	116.2	379.9	212.9	1.953
0.4	154.9	385.1	223.3	2.021
0.5	193.6	391.8	237.2	2.110
0.6	232.3	400.0	255.1	2.222
0.7	271.0	409.7	277.4	2.359
0.8	309.8	420.9	304.9	2.524
0.9	348.5	433.6	338.3	2.718
1.0	387.2	447.8	378.6	2.946
1.1	425.9	463.5	427.0	3.210
1.2	464.7	480.6	485.0	3.516
1.3	503.4	499.3	554.1	3.867
1.4	542.1	519.4	636.5	4.269
1.5	580.8	541.1	734.2	4.728
1.6	619.5	564.2	850.1	5.250
1.7	658.3	588.8	987.2	5.842
1.8	697.0	615.0	1149.2	6.511
1.9	735.7	642.6	1340.1	7.267
2.0	774.4	671.7	1564.9	8.118

Discussion Note that as Mach number increases, so does the flow velocity and stagnation temperature, pressure, and density.

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17-46 An aircraft is designed to cruise at a given Mach number, elevation, and the atmospheric temperature. The stagnation temperature on the leading edge of the wing is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are $R = 0.287$ kPa.m^3/kg.K, $c_p = 1.005$ kJ/kg·K, and $k = 1.4$.

Analysis The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(236.15\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 308.0\ \text{m/s}$$

Thus,

$$V = \text{Ma} \times c = (1.4)(308.0\ \text{m/s}) = 431.2\ \text{m/s}$$

Then,

$$T_0 = T + \frac{V^2}{2c_p} = 236.15 + \frac{(431.2\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{329\ K}$$

Discussion Note that the temperature of a gas increases during a stagnation process as the kinetic energy is converted to enthalpy.

17-47E Air flows through a duct at a specified state and Mach number. The velocity and the stagnation pressure, temperature, and density of the air are to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are $R = 0.06855$ Btu/lbm·R = 0.3704 psia·ft^3/lbm·R and $k = 1.4$.

Analysis First, $T = 320 + 459.67 = 779.67$ K. The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.06855\ \text{Btu/1bm}\cdot\text{R})(779.67\ \text{R})\left(\frac{25{,}037\ \text{ft}^2/\text{s}^2}{1\ \text{Btu/1bm}}\right)} = 1368.72\ \text{ft/s}$$

Thus,

$$V = \text{Ma} \times c = (0.7)(1368.72\ \text{ft/s}) = 958.10 \cong \mathbf{958\ ft/s}$$

Also,

$$\rho = \frac{P}{RT} = \frac{25\ \text{psia}}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(779.67\ \text{R})} = 0.086568\ \text{1bm/ft}^3$$

Then the stagnation properties are determined from

$$T_0 = T\left(1 + \frac{(k-1)\text{Ma}^2}{2}\right) = (779.67\ \text{R})\left(1 + \frac{(1.4\text{-}1)(0.7)^2}{2}\right) = 856.08\ \text{R} \cong \mathbf{856\ R}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (25\ \text{psia})\left(\frac{856.08\ \text{R}}{779.67\ \text{R}}\right)^{1.4/(1.4-1)} = 34.678\ \text{psia} \cong \mathbf{34.7\ psia}$$

$$\rho_0 = \rho\left(\frac{T_0}{T}\right)^{1/(k-1)} = (0.08656\ \ \text{1bm/ft}^3)\left(\frac{856.08\ \text{R}}{779.67\ \text{R}}\right)^{1/(1.4-1)} = 0.10936\ \text{lbm/ft}^3 \cong \mathbf{0.109\ lbm/ft^3}$$

Discussion Note that the temperature, pressure, and density of a gas increases during a stagnation process.

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Isentropic Flow Through Nozzles

17-48C The fluid would **accelerate even further** instead of decelerating.

Discussion This is the opposite of what would happen in subsonic flow.

17-49C The fluid would **accelerate even further**, as desired.

Discussion This is the opposite of what would happen in subsonic flow.

17-50C (*a*) The **exit velocity reaches the sonic speed**, (*b*) the **exit pressure equals the critical pressure**, and (*c*) the **mass flow rate reaches the maximum value**.

Discussion In such a case, we say that the flow is *choked*.

17-51C (*a*) **No effect on velocity**. (*b*) **No effect on pressure**. (*c*) **No effect on mass flow rate**.

Discussion In this situation, the flow is already choked initially, so further lowering of the back pressure does not change anything upstream of the nozzle exit plane.

17-52C **If the back pressure is low enough so that sonic conditions exist at the throats, the mass flow rates in the two nozzles would be identical**. However, **if the flow is not sonic at the throat, the mass flow rate through the nozzle with the diverging section would be greater**, because it acts like a subsonic diffuser.

Discussion Once the flow is choked at the throat, whatever happens downstream is irrelevant to the flow upstream of the throat.

17-53C Maximum flow rate through a converging nozzle is achieved when Ma = 1 at the exit of a nozzle. For all other Ma values the mass flow rate decreases. Therefore, **the mass flow rate would decrease if hypersonic velocities were achieved at the throat of a converging nozzle**.

Discussion Note that this is not possible unless the flow upstream of the converging nozzle is already hypersonic.

17-54C Ma* is the **local velocity non-dimensionalized with respect to the sonic speed at the throat**, whereas Ma is the **local velocity non-dimensionalized with respect to the local sonic speed**.

Discussion The two are identical at the throat when the flow is choked.

17-55C (*a*) The **velocity decreases**, (*b*) the **pressure increases**, and (*c*) the **mass flow rate remains the same**.

Discussion Qualitatively, this is the same as what we are used to (in previous chapters) for incompressible flow.

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17-56C No, if the flow in the throat is subsonic. If the velocity at the throat is subsonic, the diverging section would act like a diffuser and decelerate the flow. **Yes, if the flow in the throat is already supersonic**, the diverging section would accelerate the flow to even higher Mach number.

Discussion In duct flow, the latter situation is not possible unless a second converging-diverging portion of the duct is located upstream, and there is sufficient pressure difference to choke the flow in the upstream throat.

17-57 It is to be explained why the maximum flow rate per unit area for a given ideal gas depends only on $P_0 / \sqrt{T_0}$. Also for an ideal gas, a relation is to be obtained for the constant a in $\dot{m}_{\max} / A^* = a\left(P_0 / \sqrt{T_0}\right)$.

Properties The properties of the ideal gas considered are $R = 0.287$ kPa.m^3/kg·K and $k = 1.4$.

Analysis The maximum flow rate is given by

$$\dot{m}_{\max} = A^* P_0 \sqrt{k/RT_0}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)} \qquad \text{or} \qquad \dot{m}_{\max} / A^* = \left(P_0 / \sqrt{T_0}\right)\sqrt{k/R}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)}$$

For a given gas, k and R are fixed, and thus the mass flow rate depends on the parameter $P_0 / \sqrt{T_0}$. Thus, $\dot{m}_{\max} / A^*$ can be expressed as $\dot{m}_{\max} / A^* = a\left(P_0 / \sqrt{T_0}\right)$ where

$$a = \sqrt{k/R}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)} = \sqrt{\frac{1.4}{(0.287\ \text{kJ/kg.K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)}}\left(\frac{2}{1.4+1}\right)^{2.4/0.8} = \mathbf{0.0404\ (m/s)\sqrt{K}}$$

Discussion Note that when sonic conditions exist at a throat of known cross-sectional area, the mass flow rate is fixed by the stagnation conditions.

17-58 For an ideal gas, an expression is to be obtained for the ratio of the speed of sound where Ma = 1 to the speed of sound based on the stagnation temperature, c^*/c_0.

Analysis For an ideal gas the speed of sound is expressed as $c = \sqrt{kRT}$. Thus,

$$\frac{c^*}{c_0} = \frac{\sqrt{kRT^*}}{\sqrt{kRT_0}} = \left(\frac{T^*}{T_0}\right)^{1/2} = \left(\frac{\mathbf{2}}{\boldsymbol{k+1}}\right)^{\mathbf{1/2}}$$

Discussion Note that a speed of sound changes the flow as the temperature changes.

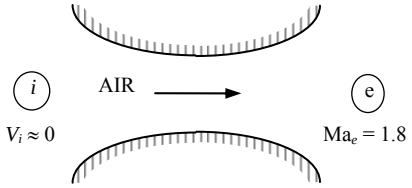
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17-59 Air enters a converging-diverging nozzle at a specified pressure. The back pressure that will result in a specified exit Mach number is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio of air is $k = 1.4$.

Analysis The stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. It remains constant throughout the nozzle since the flow is isentropic,



$$P_0 = P_i = 1.2 \text{ MPa}$$

From Table A-32 at $Ma_e = 1.8$, we read $P_e / P_0 = 0.1740$.

Thus,

$$P = 0.1740 P_0 = 0.1740(1.2 \text{ MPa}) = \mathbf{0.209 \text{ MPa}}$$

Discussion If we solve this problem using the relations for compressible isentropic flow, the results would be identical.

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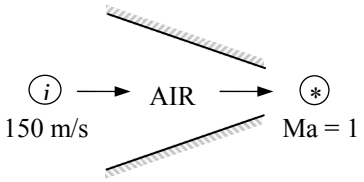
17-60 Air enters a nozzle at specified temperature, pressure, and velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air are $k = 1.4$ and $c_p = 1.005$ kJ/kg·K.

Analysis The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T_i + \frac{V_i^2}{2c_p} = 420\ \text{K} + \frac{(150\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 431.194\ \text{K}$$



and

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (0.6\ \text{MPa})\left(\frac{431.194\ \text{K}}{420\ \text{K}}\right)^{1.4/(1.4-1)} = 0.65786\ \text{MPa}$$

From Table A-32 (or from Eqs. 17-18 and 17-19) at Ma = 1, we read $T/T_0 = 0.8333$, $P/P_0 = 0.5283$. Thus,

$$T = 0.8333T_0 = 0.8333(431.194\ \text{K}) = 359.31\ \text{K} \approx \mathbf{359\ K}$$

and

$$P = 0.5283P_0 = 0.5283(0.65786\ \text{MPa}) = 0.34754\ \text{MPa} \approx \mathbf{0.348\ MPa = 348\ kPa}$$

Also,

$$c_i = \sqrt{kRT}_i = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(420\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 410.799\ \text{m/s}$$

and

$$\text{Ma}_i = \frac{V_i}{c_i} = \frac{150\ \text{m/s}}{410.799\ \text{m/s}} = 0.3651$$

From Table A-32 at this Mach number we read $A_i/A^* = 1.7452$. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A_i} = \frac{1}{1.7452} = 0.57300 \cong \mathbf{0.573}$$

Discussion We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

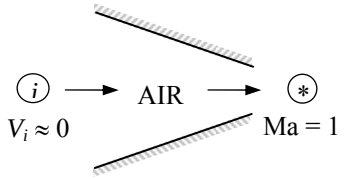
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17-61 Air enters a nozzle at specified temperature and pressure with low velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio of air is $k = 1.4$.

Analysis The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. The stagnation temperature and pressure in this case are identical to the inlet temperature and pressure since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.



$$T_0 = T_i = 350 \text{ K} \quad \text{and} \quad P_0 = P_i = 0.2 \text{ MPa}$$

From Table A-32 (or from Eqs. 17-18 and 17-19) at Ma =1, we read $T/T_0 = 0.8333$, $P/P_0 = 0.5283$.

Thus,

$$T = 0.8333T_0 = 0.8333(350 \text{ K}) = \mathbf{292\ K}$$

and

$$P = 0.5283P_0 = 0.5283(0.2 \text{ MPa}) = \mathbf{0.106\ MPa}$$

The Mach number at the nozzle inlet is Ma = 0 since $V_i \cong 0$. From Table A-32 at this Mach number we read $A_i/A^* = \infty$.

Thus the ratio of the throat area to the nozzle inlet area is $\dfrac{A^*}{A_i} = \dfrac{1}{\infty} = \mathbf{0}$.

Discussion If we solve this problem using the relations for compressible isentropic flow, the results would be identical.

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17-62E Air enters a nozzle at specified temperature, pressure, and velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

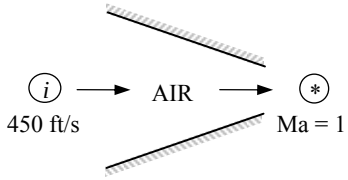
Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air are $k = 1.4$ and $c_p = 0.240$ Btu/lbm·R (Table A-2Ea).

Analysis The properties of the fluid at the location where Ma =1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T + \frac{V_i^2}{2c_p} = 630\ \text{R} + \frac{(450\ \text{ft/s})^2}{2 \times 0.240\ \text{Btu/lbm} \cdot \text{R}} \left(\frac{1\ \text{Btu/1bm}}{25{,}037\ \text{ft}^2/\text{s}^2} \right) = 646.9\ \text{R}$$

$$P_0 = P_i \left(\frac{T_0}{T_i} \right)^{k/(k-1)} = (30\ \text{psia}) \left(\frac{646.9\ \text{K}}{630\ \text{K}} \right)^{1.4/(1.4-1)} = 32.9\ \text{psia}$$



From Table A-32 (or from Eqs. 17-18 and 17-19) at Ma =1, we read $T/T_0 = 0.8333$, $P/P_0 = 0.5283$.

Thus,

$$T = 0.8333T_0 = 0.8333(646.9\ \text{R}) = \mathbf{539\ R}$$

and

$$P = 0.5283P_0 = 0.5283(32.9\ \text{psia}) = \mathbf{17.4\ psia}$$

Also,

$$c_i = \sqrt{kRT}_i = \sqrt{(1.4)(0.06855\ \text{Btu/1bm} \cdot \text{R})(630\ \text{R}) \left(\frac{25{,}037\ \text{ft}^2/\text{s}^2}{1\ \text{Btu/1bm}} \right)} = 1230\ \text{ft/s}$$

and

$$\text{Ma}_i = \frac{V_i}{c_i} = \frac{450\ \text{ft/s}}{1230\ \text{ft/s}} = 0.3657$$

From Table A-32 at this Mach number we read $A_i/A^* = 1.7426$. Thus the ratio of the throat area to the nozzle inlet area is

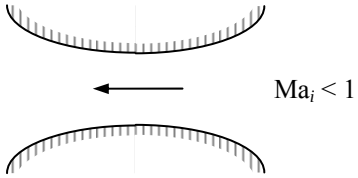
$$\frac{A^*}{A_i} = \frac{1}{1.7426} = \mathbf{0.574}$$

Discussion If we solve this problem using the relations for compressible isentropic flow, the results would be identical.

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17-63 For subsonic flow at the inlet, the variation of pressure, velocity, and Mach number along the length of the nozzle are to be sketched for an ideal gas under specified conditions.

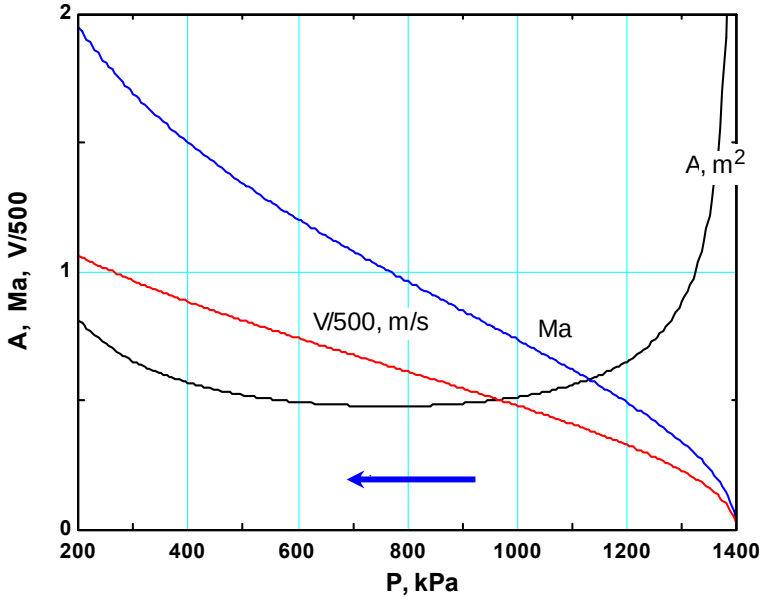
Assumptions **1** The gas is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The flow is choked at the throat.



Analysis Using EES and CO_2 as the gas, we calculate and plot flow area *A*, velocity *V*, and Mach number Ma as the pressure drops from a stagnation value of 1400 kPa to 200 kPa. Note that the curve for *A* is related to the shape of the nozzle, with horizontal axis serving as the centerline. The EES equation window and the plot are shown below.

```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
P0=1400 "kPa"

T0=473 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
rho_norm=rho/rho_0 "Normalized density"
T=T0*(P/P0)^((k-1)/k)
Tnorm=T/T0  "Normalized temperature"
V=SQRT(2*Cp*(T0-T)*1000)
V_norm=V/500
A=m/(rho*V)*500
C=SQRT(k*R*T*1000)
Ma=V/C
```



Discussion We are assuming that the back pressure is sufficiently low that the flow is choked at the throat, and the flow downstream of the throat is supersonic without any shock waves. Mach number and velocity continue to rise right through the throat into the diverging portion of the nozzle, since the flow becomes supersonic.

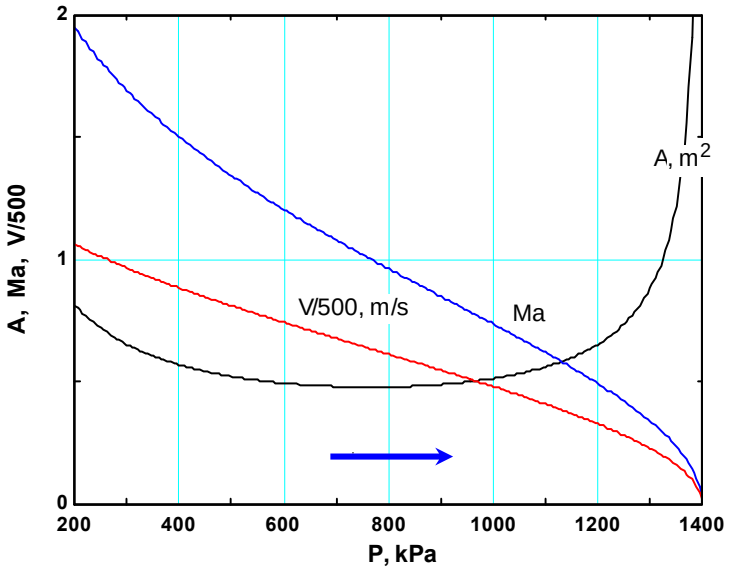
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17-64 We repeat the previous problem, but for supersonic flow at the inlet. The variation of pressure, velocity, and Mach number along the length of the nozzle are to be sketched for an ideal gas under specified conditions.

Analysis Using EES and CO_2 as the gas, we calculate and plot flow area *A*, velocity *V*, and Mach number Ma as the pressure rises from 200 kPa at a very high velocity to the stagnation value of 1400 kPa. Note that the curve for *A* is related to the shape of the nozzle, with horizontal axis serving as the centerline.

```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
P0=1400 "kPa"

T0=473 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
rho_norm=rho/rho_0 "Normalized density"
T=T0*(P/P0)^((k-1)/k)
Tnorm=T/T0  "Normalized temperature"
V=SQRT(2*Cp*(T0-T)*1000)
V_norm=V/500
A=m/(rho*V)*500
C=SQRT(k*R*T*1000)
Ma=V/C
```



Discussion Note that this problem is identical to the proceeding one, except the flow direction is reversed. In fact, when plotted like this, the plots are identical.

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17-65 Nitrogen enters a converging-diverging nozzle at a given pressure. The critical velocity, pressure, temperature, and density in the nozzle are to be determined.

Assumptions **1** Nitrogen is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

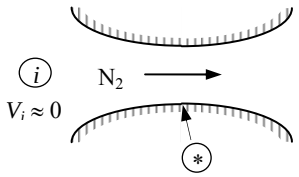
Properties The properties of nitrogen are $k = 1.4$ and $R = 0.2968$ kJ/kg·K.

Analysis The stagnation pressure in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle,

$$P_0 = P_i = 700 \text{ kPa}$$

$$T_0 = T_i = 400 \text{ K}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{700 \text{ kPa}}{(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(400 \text{ K})} = 5.896 \text{ kg/m}^3$$



Critical properties are those at a location where the Mach number is Ma = 1. From Table A-32 at Ma =1, we read T/T_0 =0.8333, $P/P_0 = 0.5283$, and $\rho/\rho_0 = 0.6339$.

Then the critical properties become

$$T^* = 0.8333T_0 = 0.8333(400 \text{ K}) = \mathbf{333 \text{ K}}$$

$$P^* = 0.5283P_0 = 0.5283(700 \text{ kPa}) = \mathbf{370 \text{ MPa}}$$

$$\rho^* = 0.6339\rho_0 = 0.6339(5.896 \text{ kg/m}^3) = \mathbf{3.74 \text{ kg/m}^3}$$

Also,

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.4)(0.2968 \text{ kJ/kg} \cdot \text{K})(333 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{372 \text{ m/s}}$$

Discussion We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

17-66 An ideal gas is flowing through a nozzle. The flow area at a location where Ma = 2.4 is specified. The flow area where Ma = 1.2 is to be determined.

Assumptions Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio is given to be $k = 1.4$.

Analysis The flow is assumed to be isentropic, and thus the stagnation and critical properties remain constant throughout the nozzle. The flow area at a location where $\text{Ma}_2 = 1.2$ is determined using A/A^* data from Table A-32 to be

$$\text{Ma}_1 = 2.4 : \frac{A_1}{A^*} = 2.4031 \longrightarrow A^* = \frac{A_1}{2.4031} = \frac{36 \text{ cm}^2}{2.4031} = 14.98 \text{ cm}^2$$

$$\text{Ma}_2 = 1.2 : \frac{A_2}{A^*} = 1.0304 \longrightarrow A_2 = (1.0304)A^* = (1.0304)(14.98 \text{ cm}^2) = \mathbf{15.4 \text{ cm}^2}$$

Discussion We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

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17-67 An ideal gas is flowing through a nozzle. The flow area at a location where Ma = 2.4 is specified. The flow area where Ma = 1.2 is to be determined.

Assumptions Flow through the nozzle is steady, one-dimensional, and isentropic.

Analysis The flow is assumed to be isentropic, and thus the stagnation and critical properties remain constant throughout the nozzle. The flow area at a location where $\text{Ma}_2 = 1.2$ is determined using the A/A^* relation,

$$\frac{A}{A^*} = \frac{1}{\text{Ma}}\left\{\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}^2\right)\right\}^{(k+1)/2(k-1)}$$

For $k = 1.33$ and $\text{Ma}_1 = 2.4$:

$$\frac{A_1}{A^*} = \frac{1}{2.4}\left\{\left(\frac{2}{1.33+1}\right)\left(1+\frac{1.33-1}{2}2.4^2\right)\right\}^{2.33/2\times 0.33} = 2.570$$

and

$$A^* = \frac{A_1}{2.570} = \frac{36\,\text{cm}^2}{2.570} = 14.01\ \text{cm}^2$$

For $k = 1.33$ and $\text{Ma}_2 = 1.2$:

$$\frac{A_2}{A^*} = \frac{1}{1.2}\left\{\left(\frac{2}{1.33+1}\right)\left(1+\frac{1.33-1}{2}1.2^2\right)\right\}^{2.33/2\times 0.33} = 1.0316$$

and

$$A_2 = (1.0316)A^* = (1.0316)(14.01\,\text{cm}^2) = \mathbf{14.45\ cm^2}$$

Discussion Note that the compressible flow functions in Table A-32 are prepared for $k = 1.4$, and thus they cannot be used to solve this problem.

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17-68E Air enters a converging-diverging nozzle at a specified temperature and pressure with low velocity. The pressure, temperature, velocity, and mass flow rate are to be calculated in the specified test section.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

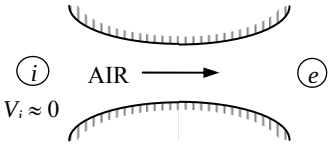
Properties The properties of air are $k = 1.4$ and $R = 0.06855$ Btu/lbm·R = 0.3704 psia·ft^3/lbm·R.

Analysis The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

$$P_0 = P_i = 150 \text{ psia}$$

and

$$T_0 = T_i = 100°\text{F} \approx 560 \text{ R}$$



Then,

$$T_e = T_0\left(\frac{2}{2+(k-1)\text{Ma}^2}\right) = (560 \text{ R})\left(\frac{2}{2+(1.4-1)2^2}\right) = \mathbf{311R}$$

$$P_e = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (150 \text{ psia})\left(\frac{311}{560}\right)^{1.4/0.4} = \mathbf{19.1 \text{ psia}}$$

$$\rho_e = \frac{P_e}{RT_e} = \frac{19.1 \text{ psia}}{(0.3704 \text{ psia.ft}^3/\text{lbm}\cdot\text{R})(311 \text{ R})} = 0.166 \text{ lbm/ft}^3$$

The nozzle exit velocity can be determined from $V_e = \text{Ma}_e c_e$, where c_e is the speed of sound at the exit conditions,

$$V_e = \text{Ma}_e c_e = \text{Ma}_e\sqrt{kRT_e} = (2)\sqrt{(1.4)(0.06855 \text{ Btu/lbm}\cdot\text{R})(311 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = 1729 \text{ ft/s} \cong \mathbf{1730 \text{ ft/s}}$$

Finally,

$$\dot{m} = \rho_e A_e V_e = (0.166 \text{ lbm/ft}^3)(5 \text{ ft}^2)(1729 \text{ ft/s}) = 1435 \text{ lbm/s} \cong \mathbf{1440 \text{ lbm/s}}$$

Discussion Air must be very dry in this application because the exit temperature of air is extremely low, and any moisture in the air will turn to ice particles.

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17-69 Air enters a converging nozzle at a specified temperature and pressure with low velocity. The exit pressure, the exit velocity, and the mass flow rate versus the back pressure are to be calculated and plotted.

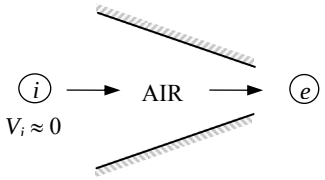
Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air are $k = 1.4$, $R = 0.287$ kJ/kg·K, and $c_p = 1.005$ kJ/kg·K.

Analysis The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.,

$$P_0 = P_i = 900 \text{ kPa}$$

$$T_0 = T_i = 400 \text{ K}$$



The critical pressure is determined to be

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (900 \text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/0.4} = 475.5 \text{ kPa}$$

Then the pressure at the exit plane (throat) will be

$$P_e = P_b \qquad \text{for} \qquad P_b \geq 475.5 \text{ kPa}$$

$$P_e = P^* = 475.5 \text{ kPa} \qquad \text{for} \qquad P_b < 475.5 \text{ kPa (choked flow)}$$

Thus the back pressure will not affect the flow when $100 < P_b < 475.5$ kPa. For a specified exit pressure P_e, the temperature, the velocity and the mass flow rate can be determined from

Temperature $$T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (400 \text{ K})\left(\frac{P_e}{900}\right)^{0.4/1.4}$$

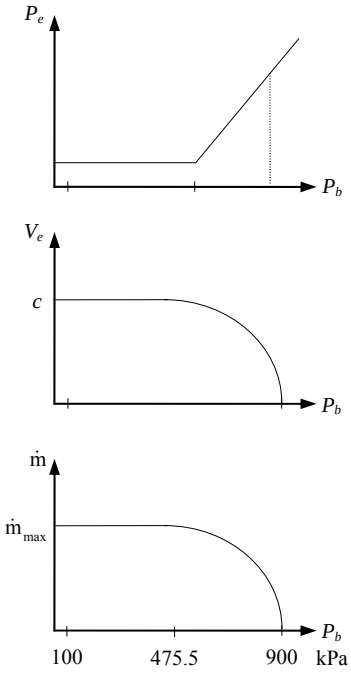
Velocity $$V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.005 \text{ kJ/kg}\cdot\text{K})(400 - T_e)\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)}$$

Density $$\rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.287 \text{ kPa}\cdot\text{m}^3/kg\cdot K)T_e}$$

Mass flow rate $$\dot{m} = \rho_e V_e A_e = \rho_e V_e (0.001 \text{ m}^2)$$

The results of the calculations are tabulated as

P_b, kPa	P_e, kPa	T_e, K	V_e, m/s	ρ_e, kg/m^3	$\dot{m}$, kg/s
900	900	400	0	7.840	0
800	800	386.8	162.9	7.206	1.174
700	700	372.3	236.0	6.551	1.546
600	600	356.2	296.7	5.869	1.741
500	500	338.2	352.4	5.151	1.815
475.5	475.5	333.3	366.2	4.971	1.820
400	475.5	333.3	366.2	4.971	1.820
300	475.5	333.3	366.2	4.971	1.820
200	475.5	333.3	366.2	4.971	1.820
100	475.5	333.3	366.2	4.971	1.820



Discussion We see from the plots that once the flow is choked at a back pressure of 475.5 kPa, the mass flow rate remains constant regardless of how low the back pressure gets.

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17-70 We are to reconsider the previous problem. Using EES (or other) software, we are to solve the problem for the inlet conditions of 0.8 MPa and 1200 K.

Analysis Air at 800 kPa, 1200 K enters a converging nozzle with a negligible velocity. The throat area of the nozzle is 10 cm2. Assuming isentropic flow, calculate and plot the exit pressure, the exit velocity, and the mass flow rate versus the back pressure P_b for 0.8>= P_b >=0.1 MPa.

```
Procedure ExitPress(P_back,P_crit : P_exit, Condition$)
If (P_back>=P_crit) then
        P_exit:=P_back   "Unchoked Flow Condition"
        Condition$:='unchoked'
else
        P_exit:=P_crit    "Choked Flow Condition"
        Condition$:='choked'
Endif
End

Gas$='Air'
A_cm2=10  "Throat area, cm2"
P_inlet =800"kPa"
T_inlet= 1200"K"
"P_back =422.7"  "kPa"

A_exit = A_cm2*Convert(cm^2,m^2)
C_p=specheat(Gas$,T=T_inlet)
C_p-C_v=R
k=C_p/C_v
M=MOLARMASS(Gas$) "Molar mass of  Gas$"
R= 8.314/M        "Gas constant for Gas$"

"Since the inlet velocity is negligible, the stagnation temperature = T_inlet;
and, since the nozzle is isentropic, the stagnation pressure = P_inlet."

P_o=P_inlet       "Stagnation pressure"
T_o=T_inlet       "Stagnation temperature"
P_crit /P_o=(2/(k+1))^(k/(k-1))      "Critical pressure from Eq. 16-22"
Call ExitPress(P_back,P_crit : P_exit, Condition$)

T_exit /T_o=(P_exit/P_o)^((k-1)/k) "Exit temperature for isentopic flow, K"

V_exit ^2/2=C_p*(T_o-T_exit)*1000          "Exit velocity, m/s"

Rho_exit=P_exit/(R*T_exit)          "Exit density, kg/m3"

m_dot=Rho_exit*V_exit*A_exit    "Nozzle mass flow rate, kg/s"

"If you wish to redo the plots, hide the diagram window and remove the { } from
the first 4 variables just under the procedure.  Next set the desired range of
back pressure in the parametric table.  Finally, solve the table (F3).  "
```

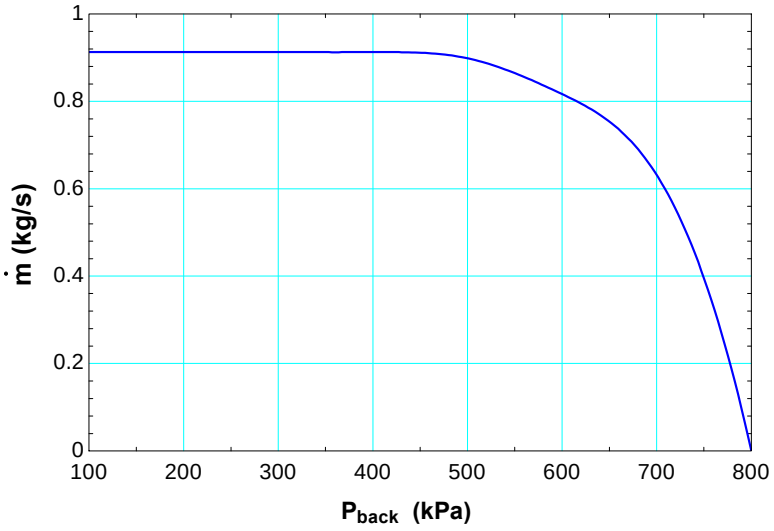
The table of results and the corresponding plot are provided below.

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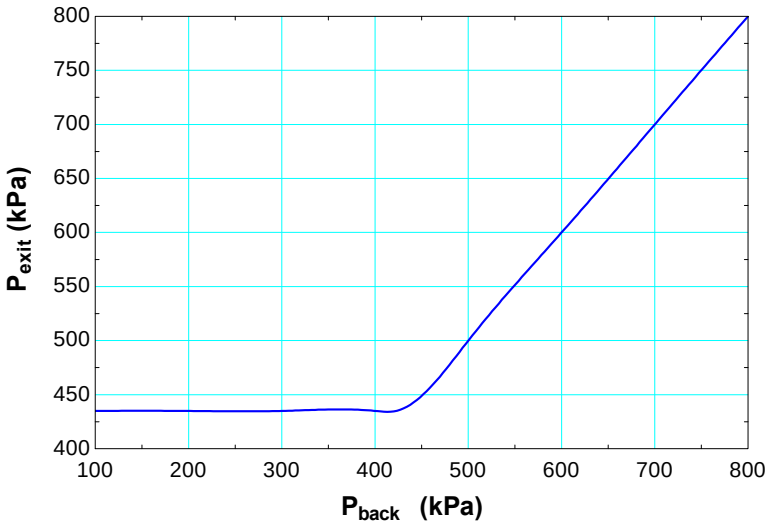
EES SOLUTION

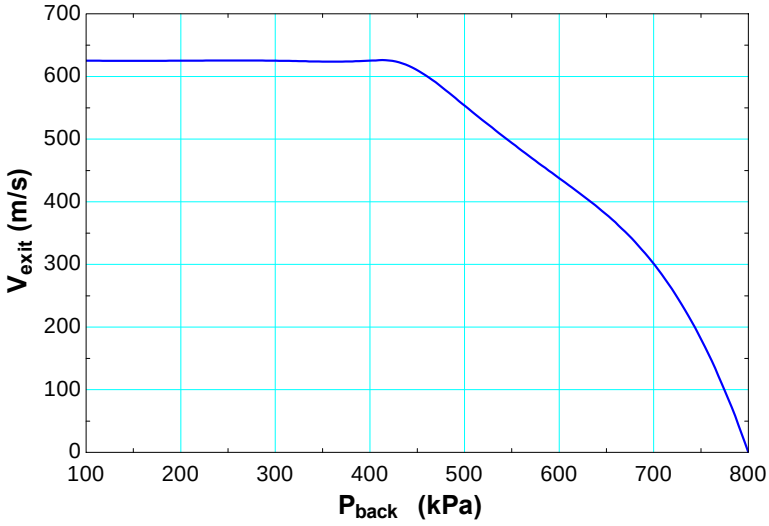
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A_exit=0.001
Condition$='choked'
C_p=1.208
C_v=0.9211
Gas$='Air'
k=1.312
M=28.97
m_dot=0.9124
P_back=422.7
P_crit=434.9
P_exit=434.9
P_inlet=800
P_o=800
R=0.287
Rho_exit=1.459
T_exit=1038
T_inlet=1200
T_o=1200
V_exit=625.2

P_{back}[kPa]	P_{exit} [kPa]	V_{exit} [m/s]	m [kg/s]	T_{exit} [K]	ρ_{exit} [kg/m^3]
100	434.9	625.2	0.9124	1038	1.459
200	434.9	625.2	0.9124	1038	1.459
300	434.9	625.2	0.9124	1038	1.459
400	434.9	625.2	0.9124	1038	1.459
422.7	434.9	625.2	0.9124	1038	1.459
500	500	553.5	0.8984	1073	1.623
600	600	437.7	0.8164	1121	1.865
700	700	300.9	0.6313	1163	2.098
800	800	0.001523	0.000003538	1200	2.323



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Discussion We see from the plot that once the flow is choked at a back pressure of 422.7 kPa, the mass flow rate remains constant regardless of how low the back pressure gets.

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Shock Waves and Expansion Waves

17-71C **No**, because the flow must be supersonic before a shock wave can occur. The flow in the converging section of a nozzle is always subsonic.

Discussion A normal shock (if it is to occur) would occur in the supersonic (diverging) section of the nozzle.

17-72C The *Fanno line* represents the **states that satisfy the conservation of mass and energy equations**. The *Rayleigh line* represents the **states that satisfy the conservation of mass and momentum equations**. The *intersections* points of these lines represent the **states that satisfy the conservation of mass, energy, and momentum equations**.

Discussion *T-s* diagrams are quite helpful in understanding these kinds of flows.

17-73C **No**, the second law of thermodynamics requires the flow after the shock to be subsonic.

Discussion A normal shock wave always goes from supersonic to subsonic in the flow direction.

17-74C (*a*) **velocity decreases**, (*b*) **static temperature increases**, (*c*) **stagnation temperature remains the same**, (*d*) **static pressure increases**, and (*e*) **stagnation pressure decreases**.

Discussion In addition, the Mach number goes from supersonic (Ma > 1) to subsonic (Ma < 1).

17-75C *Oblique shocks* occur when **a gas flowing at supersonic speeds strikes a flat or inclined surface**. Normal shock waves are perpendicular to flow whereas inclined shock waves, as the name implies, are typically **inclined relative to the flow direction**. Also, normal shocks form a straight line whereas **oblique shocks can be straight or curved**, depending on the surface geometry.

Discussion In addition, while a normal shock must go from supersonic (Ma > 1) to subsonic (Ma < 1), the Mach number downstream of an oblique shock can be either supersonic or subsonic.

17-76C **Yes**, the *upstream flow* has to be supersonic for an oblique shock to occur. **No**, the flow *downstream* of an oblique shock can be subsonic, sonic, and even supersonic.

Discussion The latter is not true for normal shocks. For a normal shock, the flow must always go from supersonic (Ma > 1) to subsonic (Ma < 1).

17-77C **Yes, the claim is correct**. Conversely, normal shocks can be thought of as special oblique shocks in which the shock angle is $\beta = \pi/2$, or 90^o.

Discussion The component of flow in the direction normal to the oblique shock acts exactly like a normal shock. We can think of the flow parallel to the oblique shock as "going along for the ride" – it does not affect anything.

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17-78C When the wedge half-angle δ is greater than the maximum deflection angle θ_{max}, the shock becomes curved and detaches from the nose of the wedge, forming what is called a *detached oblique shock* or a *bow wave*. The numerical value of the shock angle at the nose is $\beta = 90^o$.

Discussion When δ is less than θ_{max}, the oblique shock is attached to the nose.

17-79C When supersonic flow impinges on a blunt body like the rounded nose of an aircraft, the wedge half-angle δ at the nose is 90^o, and an attached oblique shock cannot exist, regardless of Mach number. Therefore, **a detached oblique shock must occur in front of *all* such blunt-nosed bodies**, whether two-dimensional, axisymmetric, or fully three-dimensional.

Discussion Since $\delta = 90^o$ at the nose, δ is always greater than θ_{max}, regardless of Ma or the shape of the rest of the body.

17-80C The isentropic relations of ideal gases are ***not* applicable for flows across (*a*) normal shock waves and (*b*) oblique shock waves**, but they ***are* applicable for flows across (*c*) Prandtl-Meyer expansion waves**.

Discussion Flow across any kind of shock wave involves irreversible losses – hence, it cannot be isentropic.

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17-81 Air flowing through a nozzle experiences a normal shock. Various properties are to be calculated before and after the shock.

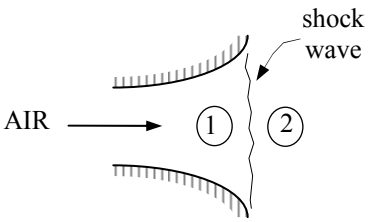
Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air at room temperature are k= 1.4, R = 0.287 kJ/kg·K, and c_p = 1.005 kJ/kg·K.

Analysis The stagnation temperature and pressure before the shock are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 205 + \frac{(740\ \text{m/s})^2}{2(1.005\ \text{kJ/kg}\cdot\text{K})}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 477.4\ \text{K}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (18\ \text{kPa})\left(\frac{477.4\ \text{K}}{205\ \text{K}}\right)^{1.4/(1.4-1)} = 347.0\ \text{kPa}$$



The velocity and the Mach number before the shock are determined from

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(205\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{287.0\ m/s}$$

and

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{740\ \text{m/s}}{287.0\ \text{m/s}} = \mathbf{2.578}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-33. For Ma_1 = 2.578 we read (We obtained the following values using analytical relations in Table A-33.)

$$\text{Ma}_2 = \mathbf{0.5058},\quad \frac{P_{02}}{P_1} = 9.0349,\quad \frac{P_2}{P_1} = 7.5871,\quad \text{and}\quad \frac{T_2}{T_1} = 2.2158$$

Then the stagnation pressure P_{02} , static pressure P_2 , and static temperature T_2 , are determined to be

$P_{02} = 9.0349P_1 = (9.0349)(18\ \text{kPa}) = \mathbf{162.6\ kPa}$

$P_2 = 7.5871P_1 = (7.5871)(18\ \text{kPa}) = \mathbf{136.6\ kPa}$

$T_2 = 2.2158T_1 = (2.2158)(205\ \text{K}) = \mathbf{454.2\ K}$

The air velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.5058)\sqrt{(1.4)(0.287\ \text{kJ/kg.K})(454.2\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{216.1\ m/s}$$

Discussion This problem could also be solved using the relations for compressible flow and normal shock functions. The results would be identical.

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17-82 Air flowing through a nozzle experiences a normal shock. The entropy change of air across the normal shock wave is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air at room temperature are $R = 0.287$ kJ/kg·K and $c_p = 1.005$ kJ/kg·K.

Analysis The entropy change across the shock is determined to be

$$\begin{aligned} s_2 - s_1 &= c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \\ &= (1.005\ \text{kJ/kg} \cdot \text{K}) \ln(2.2158) - (0.287\ \text{kJ/kg} \cdot \text{K}) \ln(7.5871) \\ &= \mathbf{0.2180\ \ kJ/kg \cdot K} \end{aligned}$$

Discussion A shock wave is a highly dissipative process, and the entropy generation is large during shock waves.

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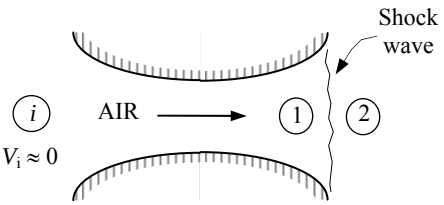
17-83 Air flowing through a converging-diverging nozzle experiences a normal shock at the exit. The effect of the shock wave on various properties is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs. **3** The shock wave occurs at the exit plane.

Properties The properties of air are $k = 1.4$ and $R = 0.287$ kJ/kg·K.

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Then,

$$P_{01} = P_i = 1 \text{ MPa}$$
$$T_{01} = T_i = 300 \text{ K}$$



Then,

$$T_1 = T_{01}\left(\frac{2}{2+(k-1)\text{Ma}_1^2}\right) = (300 \text{ K})\left(\frac{2}{2+(1.4-1)2.4^2}\right) = 139.4 \text{ K}$$

and

$$P_1 = P_{01}\left(\frac{T_1}{T_0}\right)^{k/(k-1)} = (1 \text{ MPa})\left(\frac{139.4}{300}\right)^{1.4/0.4} = 0.06840 \text{ MPa}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-33. For $\text{Ma}_1 = 2.4$ we read

$$\text{Ma}_2 = 0.5231 \cong \mathbf{0.523}, \quad \frac{P_{02}}{P_{01}} = 0.5401, \quad \frac{P_2}{P_1} = 6.5533, \quad \text{and} \quad \frac{T_2}{T_1} = 2.0403$$

Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

$P_{02} = 0.5401P_{01} = (0.5401)(1.0 \text{ MPa}) =$ **0.540 MPa = 540 kPa**

$P_2 = 6.5533P_1 = (6.5533)(0.06840 \text{ MPa}) =$ **0.448 MPa = 448 kPa**

$T_2 = 2.0403T_1 = (2.0403)(139.4 \text{ K}) =$ **284 K**

The air velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.5231)\sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(284 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{177 \text{ m/s}}$$

Discussion We can also solve this problem using the relations for normal shock functions. The results would be identical.

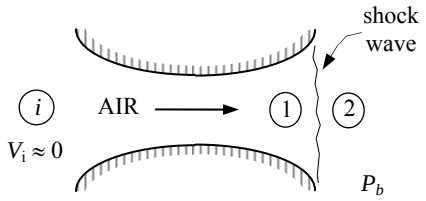
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17-84 Air enters a converging-diverging nozzle at a specified state. The required back pressure that produces a normal shock at the exit plane is to be determined for the specified nozzle geometry.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs. **3** The shock wave occurs at the exit plane.

Analysis The inlet stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. Since the flow before the shock to be isentropic,

$$P_{01} = P_i = 2 \text{ MPa}$$



It is specified that $A/A^* = 3.5$. From Table A-32, Mach number and the pressure ratio which corresponds to this area ratio are the $Ma_1 = 2.80$ and $P_1/P_{01} = 0.0368$. The pressure ratio across the shock for this Ma_1 value is, from Table A-33, $P_2/P_1 = 8.98$. Thus the back pressure, which is equal to the static pressure at the nozzle exit, must be

$$P_2 = 8.98P_1 = 8.98 \times 0.0368 P_{01} = 8.98 \times 0.0368 \times (2 \text{ MPa}) = \mathbf{0.661 \text{ MPa}}$$

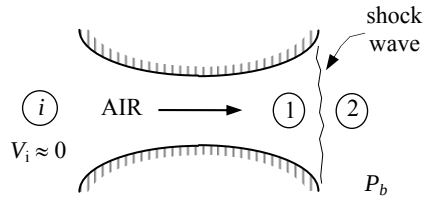
Discussion We can also solve this problem using the relations for compressible flow and normal shock functions. The results would be identical.

17-85 Air enters a converging-diverging nozzle at a specified state. The required back pressure that produces a normal shock at the exit plane is to be determined for the specified nozzle geometry.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Analysis The inlet stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. Since the flow before the shock to be isentropic,

$$P_{0x} = P_i = 2 \text{ MPa}$$



It is specified that $A/A^* = 2$. From Table A-32, the Mach number and the pressure ratio which corresponds to this area ratio are the $Ma_1 = 2.20$ and $P_1/P_{01} = 0.0935$. The pressure ratio across the shock for this M_1 value is, from Table A-33, $P_2/P_1 = 5.48$. Thus the back pressure, which is equal to the static pressure at the nozzle exit, must be

$$P_2 = 5.48P_1 = 5.48 \times 0.0935 P_{01} = 5.48 \times 0.0935 \times (2 \text{ MPa}) = \mathbf{1.02 \text{ MPa}}$$

Discussion We can also solve this problem using the relations for compressible flow and normal shock functions. The results would be identical.

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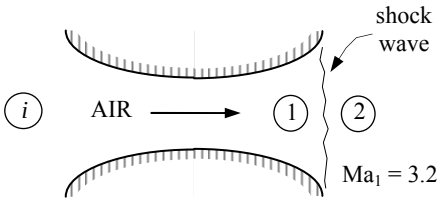
17-86 Air flowing through a nozzle experiences a normal shock. The effect of the shock wave on various properties is to be determined. Analysis is to be repeated for helium under the same conditions.

Assumptions **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are $k = 1.4$ and $R = 0.287$ kJ/kg·K, and the properties of helium are $k = 1.667$ and R = 2.0769 kJ/kg·K.

Analysis The air properties upstream the shock are

$\text{Ma}_1 = 3.2$, $P_1 = 58$ kPa, and $T_1 = 270$ K



Fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions in Table A-33. For $\text{Ma}_1 = 3.2$,

$$\text{Ma}_2 = \mathbf{0.4643},\ \frac{P_{02}}{P_1} = 13.656,\ \frac{P_2}{P_1} = 11.780,\ \text{and}\ \frac{T_2}{T_1} = 2.9220$$

We obtained these values using analytical relations in Table A-33. Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

$P_{02} = 13.656P_1 = (13.656)(58\text{ kPa}) = \mathbf{792.0\ kPa}$

$P_2 = 11.780P_1 = (11.780)(58\text{ kPa}) = \mathbf{683.2\ kPa}$

$T_2 = 2.9220T_1 = (2.9220)(270\text{ K}) = \mathbf{788.9\ K}$

The air velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.4643)\sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(788.9\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{261.4\ m/s}$$

We now repeat the analysis for helium. This time we cannot use the tabulated values in Table A-33 since k is not 1.4. Therefore, we have to calculate the desired quantities using the analytical relations,

$$\text{Ma}_2 = \left(\frac{\text{Ma}_1^2 + 2/(k-1)}{2\text{Ma}_1^2 k/(k-1) - 1}\right)^{1/2} = \left(\frac{3.2^2 + 2/(1.667-1)}{2\times 3.2^2 \times 1.667/(1.667-1) - 1}\right)^{1/2} = \mathbf{0.5136}$$

$$\frac{P_2}{P_1} = \frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2} = \frac{1 + 1.667\times 3.2^2}{1 + 1.667\times 0.5136^2} = 12.551$$

$$\frac{T_2}{T_1} = \frac{1 + \text{Ma}_1^2(k-1)/2}{1 + \text{Ma}_2^2(k-1)/2} = \frac{1 + 3.2^2(1.667-1)/2}{1 + 0.5136^2(1.667-1)/2} = 4.0580$$

$$\frac{P_{02}}{P_1} = \left(\frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2}\right)\left(1 + (k-1)\text{Ma}_2^2/2\right)^{k/(k-1)}$$

$$= \left(\frac{1 + 1.667\times 3.2^2}{1 + 1.667\times 0.5136^2}\right)\left(1 + (1.667-1)\times 0.5136^2/2\right)^{1.667/0.667} = 15.495$$

Thus, $P_{02} = 15.495P_1 = (15.495)(58\text{ kPa}) = \mathbf{898.7\ kPa}$

$P_2 = 12.551P_1 = (12.551)(58\text{ kPa}) = \mathbf{728.0\ kPa}$

$T_2 = 4.0580T_1 = (4.0580)(270\text{ K}) = \mathbf{1096\ K}$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_y} = (0.5136)\sqrt{(1.667)(2.0769\text{ kJ/kg}\cdot\text{K})(1096\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{1000\ m/s}$$

Discussion The velocity and Mach number are higher for helium than for air due to the different values of k and R.

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17-87 Air flowing through a nozzle experiences a normal shock. The entropy change of air across the normal shock wave is to be determined.

Assumptions **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are $R = 0.287$ kJ/kg·K and $c_p = 1.005$ kJ/kg·K, and the properties of helium are $R = 2.0769$ kJ/kg·K and $c_p = 5.1926$ kJ/kg·K.

Analysis The entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (1.005\,\text{kJ/kg}\cdot\text{K})\ln(2.9220) - (0.287\,\text{kJ/kg}\cdot\text{K})\ln(11.780) = \mathbf{0.370\ kJ/kg\cdot K}$$

For helium, the entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (5.1926\,\text{kJ/kg}\cdot\text{K})\ln(4.0580) - (2.0769\,\text{kJ/kg}\cdot\text{K})\ln(12.551) = \mathbf{2.02\,kJ/kg\cdot K}$$

Discussion Note that shock wave is a highly dissipative process, and the entropy generation is large during shock waves.

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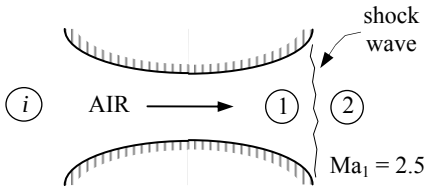
17-88E Air flowing through a nozzle experiences a normal shock. Effect of the shock wave on various properties is to be determined. Analysis is to be repeated for helium

Assumptions **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are $k = 1.4$ and $R = 0.06855$ Btu/lbm·R, and the properties of helium are $k = 1.667$ and R = 0.4961 Btu/lbm·R.

Analysis The air properties upstream the shock are

$\text{Ma}_1 = 2.5$, $P_1 = 10$ psia, and $T_1 = 440.5$ R



Fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-33. For $\text{Ma}_1 = 2.5$,

$$\text{Ma}_2 = \mathbf{0.513},\ \frac{P_{02}}{P_1} = 8.5262,\ \frac{P_2}{P_1} = 7.125,\ \text{and}\ \frac{T_2}{T_1} = 2.1375$$

Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

$P_{02} = 8.5262P_1 = (8.5262)(10\text{ psia}) = \mathbf{85.3\ psia}$

$P_2 = 7.125P_1 = (7.125)(10\text{ psia}) = \mathbf{71.3\ psia}$

$T_2 = 2.1375T_1 = (2.1375)(440.5\text{ R}) = \mathbf{942\ R}$

The air velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.513)\sqrt{(1.4)(0.06855\,\text{Btu/lbm}\cdot\text{R})(941.6\,\text{R})\left(\frac{25{,}037\,\text{ft}^2/\text{s}^2}{1\,\text{Btu/lbm}}\right)} = \mathbf{772\ ft/s}$$

We now repeat the analysis for helium. This time we cannot use the tabulated values in Table A-33 since k is not 1.4. Therefore, we have to calculate the desired quantities using the analytical relations,

$$\text{Ma}_2 = \left(\frac{\text{Ma}_1^2 + 2/(k-1)}{2\text{Ma}_1^2 k/(k-1) - 1}\right)^{1/2} = \left(\frac{2.5^2 + 2/(1.667-1)}{2\times 2.5^2 \times 1.667/(1.667-1) - 1}\right)^{1/2} = \mathbf{0.553}$$

$$\frac{P_2}{P_1} = \frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2} = \frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2} = 7.5632$$

$$\frac{T_2}{T_1} = \frac{1 + \text{Ma}_1^2(k-1)/2}{1 + \text{Ma}_2^2(k-1)/2} = \frac{1 + 2.5^2(1.667-1)/2}{1 + 0.553^2(1.667-1)/2} = 2.7989$$

$$\frac{P_{02}}{P_1} = \left(\frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2}\right)\left(1 + (k-1)\text{Ma}_2^2/2\right)^{k/(k-1)}$$

$$= \left(\frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2}\right)\left(1 + (1.667-1)\times 0.553^2/2\right)^{1.667/0.667} = 9.641$$

Thus, $P_{02} = 11.546P_1 = (11.546)(10\text{ psia}) = \mathbf{115\ psia}$

$P_2 = 7.5632P_1 = (7.5632)(10\text{ psia}) = \mathbf{75.6\ psia}$

$T_2 = 2.7989T_1 = (2.7989)(440.5\text{ R}) = \mathbf{1233\ R}$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.553)\sqrt{(1.667)(0.4961\,\text{Btu/lbm.R})(1232.9\,\text{R})\left(\frac{25{,}037\,\text{ft}^2/\text{s}^2}{1\,\text{Btu/lbm}}\right)} = \mathbf{2794\ ft/s}$$

Discussion This problem could also be solved using the relations for compressible flow and normal shock functions. The results would be identical.

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17-89E We are to reconsider Prob. 17-88E. Using EES (or other) software, we are to study the effects of both air and helium flowing steadily in a nozzle when there is a normal shock at a Mach number in the range 2 < Mx < 3.5. In addition to the required information, we are to calculate the entropy change of the air and helium across the normal shock, and tabulate the results in a parametric table.

Analysis We use EES to calculate the entropy change of the air and helium across the normal shock. The results are given in the Parametric Table for 2 < M_x < 3.5.

```
Procedure NormalShock(M_x,k:M_y,PyOPx, TyOTx,RhoyORhox, PoyOPox, PoyOPx)
    If M_x < 1 Then
            M_y = -1000;PyOPx=-1000;TyOTx=-1000;RhoyORhox=-1000
            PoyOPox=-1000;PoyOPx=-1000
    else
            M_y=sqrt( (M_x^2+2/(k-1)) / (2*M_x^2*k/(k-1)-1) )
            PyOPx=(1+k*M_x^2)/(1+k*M_y^2)
            TyOTx=( 1+M_x^2*(k-1)/2  )/(1+M_y^2*(k-1)/2  )
            RhoyORhox=PyOPx/TyOTx
            PoyOPox=M_x/M_y*(  (1+M_y^2*(k-1)/2)/ (1+M_x^2*(k-1)/2)   )^((k+1)/(2*(k-1)))
            PoyOPx=(1+k*M_x^2)*(1+M_y^2*(k-1)/2)^(k/(k-1))/(1+k*M_y^2)
    Endif
End

Function ExitPress(P_back,P_crit)
If P_back>=P_crit then ExitPress:=P_back     "Unchoked Flow Condition"
If P_back<P_crit then ExitPress:=P_crit      "Choked Flow Condition"
End

Procedure GetProp(Gas$:Cp,k,R) "Cp and k data are from Text Table A.2E"
    M=MOLARMASS(Gas$) "Molar mass of  Gas$"
    R= 1545/M         "Particular gas constant for Gas$, ft-lbf/lbm-R"
            "k = Ratio of Cp to Cv"
            "Cp = Specific heat at constant pressure"
    if Gas$='Air' then
            Cp=0.24"Btu/lbm-R"; k=1.4
    endif
    if Gas$='CO2' then
            Cp=0.203"Btu/lbm_R"; k=1.289
    endif
    if Gas$='Helium' then
            Cp=1.25"Btu/lbm-R"; k=1.667
    endif
End

"Variable Definitions:"
"M = flow Mach Number"
"P_ratio = P/P_o  for compressible, isentropic flow"
"T_ratio = T/T_o for compressible, isentropic flow"
"Rho_ratio= Rho/Rho_o for compressible, isentropic flow"
"A_ratio=A/A* for compressible, isentropic flow"
"Fluid properties before the shock are denoted with a subscript x"
"Fluid properties after the shock are denoted with a subscript y"
"M_y = Mach Number down stream of normal shock"
"PyOverPx= P_y/P_x  Pressue ratio across normal shock"
"TyOverTx =T_y/T_x  Temperature ratio across normal shock"
"RhoyOverRhox=Rho_y/Rho_x Density ratio across normal shock"
"PoyOverPox = P_oy/P_ox  Stagantion pressure ratio across normal shock"
"PoyOverPx = P_oy/P_x  Stagnation pressure after normal shock ratioed to pressure before shock"
```

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```
"Input Data"
{P_x = 10 "psia"}      "Values of P_x, T_x, and M_x are set in the Parametric Table"
{T_x = 440.5 "R"}
{M_x = 2.5}
Gas$='Air'  "This program has been written for the gases Air, CO2, and Helium"
Call GetProp(Gas$:Cp,k,R)
Call NormalShock(M_x,k:M_y,PyOverPx, TyOverTx,RhoyOverRhox, PoyOverPox, PoyOverPx)
P_oy_air=P_x*PoyOverPx      "Stagnation pressure after the shock"
P_y_air=P_x*PyOverPx        "Pressure after the shock"
T_y_air=T_x*TyOverTx        "Temperature after the shock"
M_y_air=M_y         "Mach number after the shock"

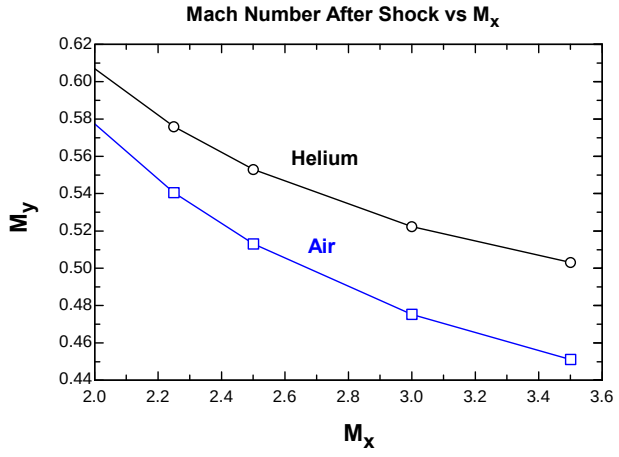
"The velocity after the shock can be found from the product of the Mach number and
speed of sound after the shock."
C_y_air = sqrt(k*R"ft-lbf/lbm_R"*T_y_air"R"*32.2 "lbm-ft/lbf-s^2")
V_y_air=M_y_air*C_y_air
DELTAs_air=entropy(air,T=T_y_air, P=P_y_air) -entropy(air,T=T_x,P=P_x)

Gas2$='Helium'  "Gas2$ can be either Helium or CO2"
Call GetProp(Gas2$:Cp_2,k_2,R_2)
Call NormalShock(M_x,k_2:M_y2,PyOverPx2, TyOverTx2,RhoyOverRhox2, PoyOverPox2, PoyOverPx2)
P_oy_he=P_x*PoyOverPx2      "Stagnation pressure after the shock"
P_y_he=P_x*PyOverPx2        "Pressure after the shock"
T_y_he=T_x*TyOverTx2        "Temperature after the shock"
M_y_he=M_y2         "Mach number after the shock"

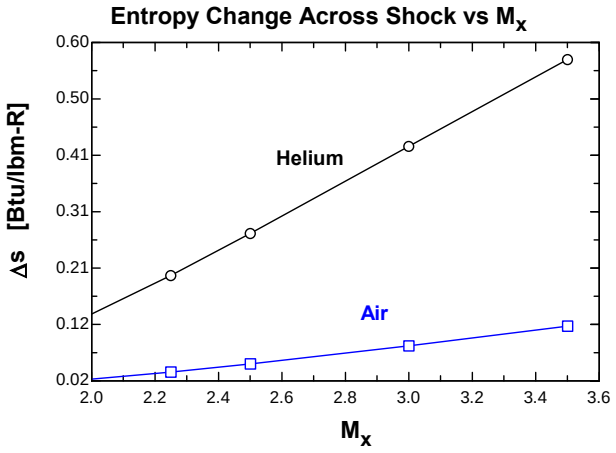
"The velocity after the shock can be found from the product of the Mach number and
speed of sound after the shock."
C_y_he = sqrt(k_2*R_2"ft-lbf/lbm_R"*T_y_he"R"*32.2 "lbm-ft/lbf-s^2")
V_y_he=M_y_he*C_y_he
DELTAs_he=entropy(helium,T=T_y_he, P=P_y_he) -entropy(helium,T=T_x,P=P_x)
```

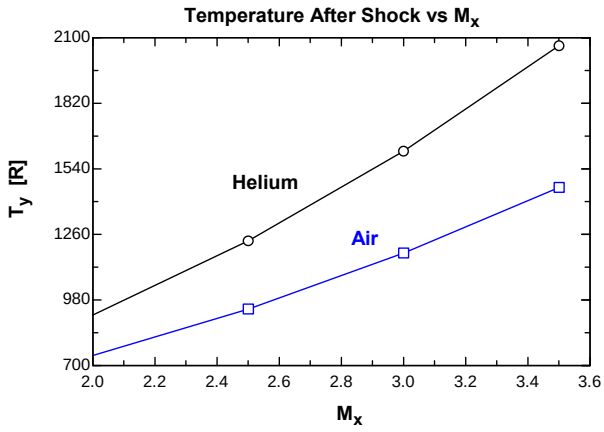
The parametric table and the corresponding plots are shown below.

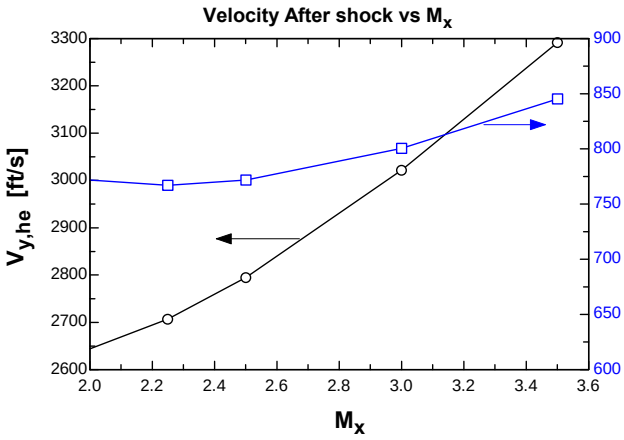
$V_{y,he}$ [ft/s]	$V_{y,air}$ [ft/s]	$T_{y,he}$ [R]	$T_{y,air}$ [R]	T_x [R]	$P_{y,he}$ [psia]	$P_{y,air}$ [psia]	P_x [psia]	$P_{oy,he}$ [psia]	$P_{oy,air}$ [psia]	$M_{y,he}$	$M_{y,air}$	M_x	Δs_{he} [Btu/lbm-R]	Δs_{air} [Btu/lbm-R]
2644	771.9	915.6	743.3	440.5	47.5	45	10	63.46	56.4	0.607	0.5774	2	0.1345	0.0228
2707	767.1	1066	837.6	440.5	60.79	57.4	10	79.01	70.02	0.5759	0.5406	2.25	0.2011	0.0351
2795	771.9	1233	941.6	440.5	75.63	71.25	10	96.41	85.26	0.553	0.513	2.5	0.2728	0.04899
3022	800.4	1616	1180	440.5	110	103.3	10	136.7	120.6	0.5223	0.4752	3	0.4223	0.08
3292	845.4	2066	1460	440.5	150.6	141.3	10	184.5	162.4	0.5032	0.4512	3.5	0.5711	0.1136

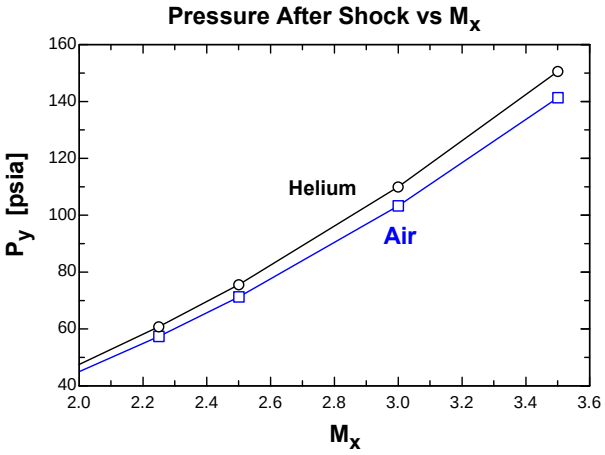


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Discussion In all cases, regardless of the fluid or the Mach number, entropy increases across a shock wave. This is because a shock wave involves irreversibilities.

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17-90 For an ideal gas flowing through a normal shock, a relation for V_2/V_1 in terms of k, Ma_1, and Ma_2 is to be developed.

Analysis The conservation of mass relation across the shock is $\rho_1 V_1 = \rho_2 V_2$ and it can be expressed as

$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{P_1 / RT_1}{P_2 / RT_2} = \left(\frac{P_1}{P_2}\right)\left(\frac{T_2}{T_1}\right)$$

From Eqs. 17-35 and 17-38,

$$\frac{V_2}{V_1} = \left(\frac{1+k\mathrm{Ma}_2^2}{1+k\mathrm{Ma}_1^2}\right)\left(\frac{1+\mathrm{Ma}_1^2(k-1)/2}{1+\mathrm{Ma}_2^2(k-1)/2}\right)$$

Discussion This is an important relation as it enables us to determine the velocity ratio across a normal shock when the Mach numbers before and after the shock are known.



17-91 The entropy change of air across the shock for upstream Mach numbers between 0.5 and 1.5 is to be determined and plotted.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are $k = 1.4$, $R = 0.287$ kJ/kg·K, and $c_p = 1.005$ kJ/kg·K.

Analysis The entropy change across the shock is determined to be

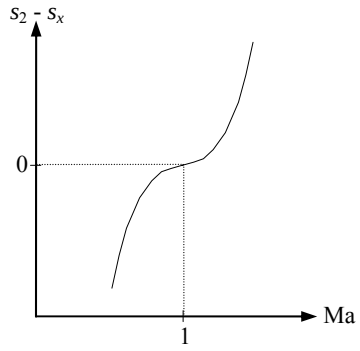
$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

where

$$\mathrm{Ma}_2 = \left(\frac{\mathrm{Ma}_1^2 + 2/(k-1)}{2\mathrm{Ma}_1^2 k/(k-1) - 1}\right)^{1/2}, \quad \frac{P_2}{P_1} = \frac{1+k\mathrm{Ma}_1^2}{1+k\mathrm{Ma}_2^2}, \text{ and } \frac{T_2}{T_1} = \frac{1+\mathrm{Ma}_1^2(k-1)/2}{1+\mathrm{Ma}_2^2(k-1)/2}$$

The results of the calculations can be tabulated as

Ma_1	Ma_2	T_2/T_1	P_2/P_1	$s_2 - s_1$
0.5	2.6458	0.1250	0.4375	-1.853
0.6	1.8778	0.2533	0.6287	-1.247
0.7	1.5031	0.4050	0.7563	-0.828
0.8	1.2731	0.5800	0.8519	-0.501
0.9	1.1154	0.7783	0.9305	-0.231
1.0	1.0000	1.0000	1.0000	0.0
1.1	0.9118	1.0649	1.2450	0.0003
1.2	0.8422	1.1280	1.5133	0.0021
1.3	0.7860	1.1909	1.8050	0.0061
1.4	0.7397	1.2547	2.1200	0.0124
1.5	0.7011	1.3202	2.4583	0.0210

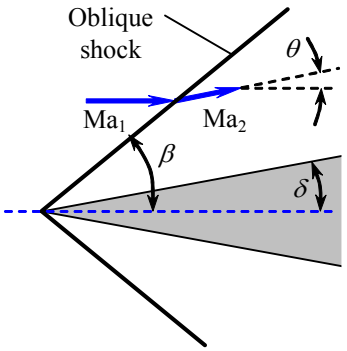


Discussion The total entropy change is negative for upstream Mach numbers Ma_1 less than unity. Therefore, normal shocks cannot occur when $Ma_1 < 1$.

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17-92 Supersonic airflow approaches the nose of a two-dimensional wedge and undergoes a straight oblique shock. For a specified Mach number, the minimum shock angle and the maximum deflection angle are to be determined.

Assumptions Air is an ideal gas with a constant specific heat ratio of $k = 1.4$ (so that Fig. 17-41 is applicable).



Analysis For Ma = 5, we read from Fig. 17-41

Minimum shock (or wave) angle: $\beta_{\min} = \mathbf{12}°$

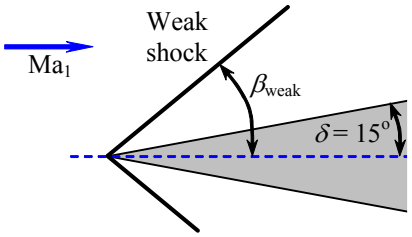
Maximum deflection (or turning) angle: $\theta_{\max} = \mathbf{41.5}°$

Discussion Note that the minimum shock angle decreases and the maximum deflection angle increases with increasing Mach number Ma_1.

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17-93E Air flowing at a specified supersonic Mach number is forced to undergo a compression turn (an oblique shock)., The Mach number, pressure, and temperature downstream of the oblique shock are to be determined.

Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.



Properties The specific heat ratio of air is $k = 1.4$.

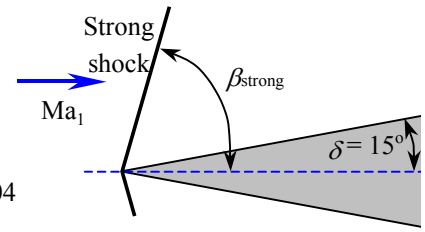
Analysis On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 15^o$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k + \cos 2\beta) + 2} \quad \rightarrow \quad \tan 15^\circ = \frac{2(2^2 \sin^2\beta - 1)/\tan\beta}{2^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\beta_{weak} = 45.34^o$ and $\beta_{strong} = 79.83^o$. Then the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

Weak shock: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 45.34^\circ = 1.423$

Strong shock: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 79.83^\circ = 1.969$



Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

Weak shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.423)^2 + 2}{2(1.4)(1.423)^2 - 1.4 + 1}} = 0.7304$$

Strong shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.969)^2 + 2}{2(1.4)(1.969)^2 - 1.4 + 1}} = 0.5828$$

The downstream pressure and temperature for each case are determined to be

Weak shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (6\ \mathrm{psia})\frac{2(1.4)(1.423)^2 - 1.4 + 1}{1.4+1} = \mathbf{13.2\ psia}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2 + (k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (480\ \mathrm{R})\frac{13.2\ \mathrm{psia}}{6\ \mathrm{psia}}\frac{2 + (1.4-1)(1.423)^2}{(1.4+1)(1.423)^2} = \mathbf{609\ R}$$

Strong shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (6\ \mathrm{psia})\frac{2(1.4)(1.969)^2 - 1.4 + 1}{1.4+1} = \mathbf{26.1\ psia}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2 + (k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (480\ \mathrm{R})\frac{26.1\ \mathrm{psia}}{6\ \mathrm{psia}}\frac{2 + (1.4-1)(1.969)^2}{(1.4+1)(1.969)^2} = \mathbf{798\ R}$$

The downstream Mach number is determined to be

Weak shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.7304}{\sin(45.34^\circ - 15^\circ)} = \mathbf{1.45}$$

Strong shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.5828}{\sin(79.83^\circ - 15^\circ)} = \mathbf{0.644}$$

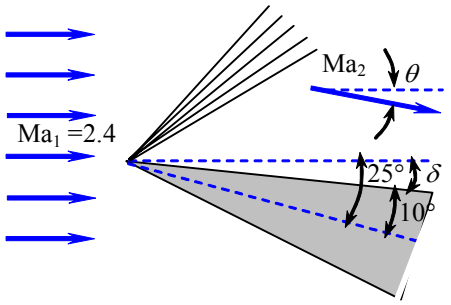
Discussion Note that the change in Mach number, pressure, temperature across the *strong shock* are much greater than the changes across the *weak shock*, as expected. For both the weak and strong oblique shock cases, $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ is subsonic. However, Ma_2 is *supersonic* across the weak oblique shock, but *subsonic* across the strong oblique shock.

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17-94 Air flowing at a specified supersonic Mach number undergoes an expansion turn over a tilted wedge. The Mach number, pressure, and temperature downstream of the sudden expansion above the wedge are to be determined.

Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is $k = 1.4$.



Analysis On the basis of Assumption #2, the deflection angle is determined to be $\theta \approx \delta = 25° - 10° = 15°$. Then the upstream and downstream Prandtl-Meyer functions are determined to be

$$\nu(\mathrm{Ma}) = \sqrt{\frac{k+1}{k-1}}\tan^{-1}\left(\sqrt{\frac{k-1}{k+1}(\mathrm{Ma}^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}^2-1}\right)$$

Upstream:

$$\nu(\mathrm{Ma}_1) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(2.4^2-1)}\right) - \tan^{-1}\left(\sqrt{2.4^2-1}\right) = 36.75°$$

Then the downstream Prandtl-Meyer function becomes

$$\nu(\mathrm{Ma}_2) = \theta + \nu(\mathrm{Ma}_1) = 15° + 36.75° = 51.75°$$

Now Ma_2 is found from the Prandtl-Meyer relation, which is now implicit:

$$\textit{Downstream: } \nu(\mathrm{Ma}_2) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(\mathrm{Ma}_2^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}_2^2-1}\right) = 51.75°$$

It gives $\mathrm{Ma}_2 = \mathbf{3.105}$. Then the downstream pressure and temperature are determined from the isentropic flow relations

$$P_2 = \frac{P_2/P_0}{P_1/P_0}P_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-k/(k-1)}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-k/(k-1)}}P_1 = \frac{[1+3.105^2(1.4-1)/2]^{-1.4/0.4}}{[1+2.4^2(1.4-1)/2]^{-1.4/0.4}}(70\ \mathrm{kPa}) = \mathbf{23.8\ kPa}$$

$$T_2 = \frac{T_2/T_0}{T_1/T_0}T_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-1}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-1}}T_1 = \frac{[1+3.105^2(1.4-1)/2]^{-1}}{[1+2.4^2(1.4-1)/2]^{-1}}(260\ \mathrm{K}) = \mathbf{191\ K}$$

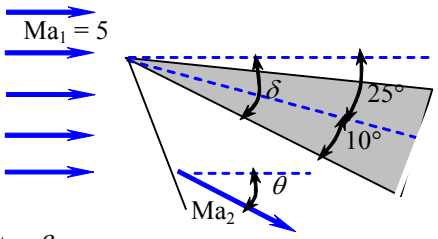
Note that this is an expansion, and Mach number increases while pressure and temperature decrease, as expected.

Discussion There are compressible flow calculators on the Internet that solve these implicit equations that arise in the analysis of compressible flow, along with both normal and oblique shock equations; e.g., see www.aoe.vt.edu/~devenpor/aoe3114/calc.html .

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17-95 Air flowing at a specified supersonic Mach number undergoes a compression turn (an oblique shock) over a tilted wedge. The Mach number, pressure, and temperature downstream of the shock below the wedge are to be determined.

Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.



Properties The specific heat ratio of air is $k = 1.4$.

Analysis On the basis of Assumption #2, the deflection angle is determined to be $\theta \approx \delta = 25° + 10° = 35°$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k + \cos 2\beta) + 2} \quad \rightarrow \quad \tan 12° = \frac{2(3.4^2 \sin^2\beta - 1)/\tan\beta}{3.4^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\beta_{\text{weak}} = 49.86°$ and $\beta_{\text{strong}} = 77.66°$. Then for the case of strong oblique shock, the upstream "normal" Mach number $\mathrm{Ma}_{1,\text{n}}$ becomes

$$\mathrm{Ma}_{1,\text{n}} = \mathrm{Ma}_1 \sin\beta = 5 \sin 77.66° = 4.884$$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,\text{n}}$ become

$$\mathrm{Ma}_{2,\text{n}} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,\text{n}}^2 + 2}{2k\mathrm{Ma}_{1,\text{n}}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(4.884)^2 + 2}{2(1.4)(4.884)^2 - 1.4 + 1}} = 0.4169$$

The downstream pressure and temperature are determined to be

$$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,\text{n}}^2 - k + 1}{k+1} = (70\ \text{kPa}) \frac{2(1.4)(4.884)^2 - 1.4 + 1}{1.4+1} = \mathbf{1940\ kPa}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1} \frac{2 + (k-1)\mathrm{Ma}_{1,\text{n}}^2}{(k+1)\mathrm{Ma}_{1,\text{n}}^2} = (260\ \text{K}) \frac{1940\ \text{kPia}}{70\ \text{kPa}} \frac{2 + (1.4-1)(4.884)^2}{(1.4+1)(4.884)^2} = \mathbf{1450\ K}$$

The downstream Mach number is determined to be

$$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,\text{n}}}{\sin(\beta - \theta)} = \frac{0.4169}{\sin(77.66° - 35°)} = \mathbf{0.615}$$

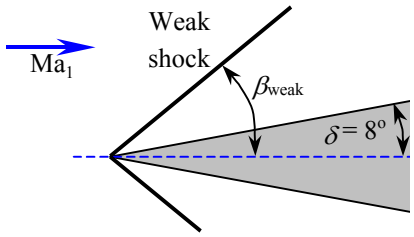
Discussion Note that $\mathrm{Ma}_{1,\text{n}}$ is supersonic and $\mathrm{Ma}_{2,\text{n}}$ and Ma_2 are subsonic. Also note the huge rise in temperature and pressure across the strong oblique shock, and the challenges they present for spacecraft during reentering the earth's atmosphere.

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17-96E Air flowing at a specified supersonic Mach number is forced to turn upward by a ramp, and weak oblique shock forms. The wave angle, Mach number, pressure, and temperature after the shock are to be determined.

Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is $k = 1.4$.



Analysis On the basis of Assumption #2, we take the deflection angle as equal to the ramp, i.e., $\theta \approx \delta = 8^o$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k + \cos 2\beta) + 2} \quad \rightarrow \quad \tan 8° = \frac{2(2^2 \sin^2\beta - 1)/\tan\beta}{2^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\boldsymbol{\beta_{weak} = 37.21^o}$ and $\beta_{strong} = 85.05^o$. Then for the case of weak oblique shock, the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

$$\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 37.21° = 1.209$$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

$$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.209)^2 + 2}{2(1.4)(1.209)^2 - 1.4 + 1}} = 0.8363$$

The downstream pressure and temperature are determined to be

$$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (12\ \mathrm{psia})\frac{2(1.4)(1.209)^2 - 1.4 + 1}{1.4+1} = \boldsymbol{18.5\ \mathrm{psia}}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2 + (k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (490\ \mathrm{R})\frac{18.5\ \mathrm{psia}}{12\ \mathrm{psia}}\frac{2 + (1.4-1)(1.209)^2}{(1.4+1)(1.209)^2} = \boldsymbol{556\ \mathrm{R}}$$

The downstream Mach number is determined to be

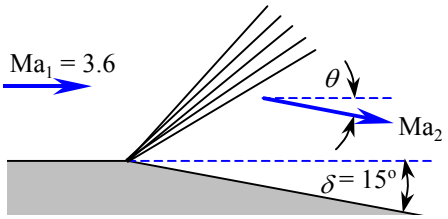
$$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.8363}{\sin(37.21° - 8°)} = \boldsymbol{1.71}$$

Discussion Note that $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ is subsonic. However, Ma_2 is *supersonic* across the weak oblique shock (it is *subsonic* across the strong oblique shock).

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17-97 Air flowing at a specified supersonic Mach number undergoes an expansion turn. The Mach number, pressure, and temperature downstream of the sudden expansion along a wall are to be determined.

Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.



Properties The specific heat ratio of air is $k = 1.4$.

Analysis On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 15^\circ$. Then the upstream and downstream Prandtl-Meyer functions are determined to be

$$\nu(\mathrm{Ma}) = \sqrt{\frac{k+1}{k-1}}\tan^{-1}\left(\sqrt{\frac{k-1}{k+1}(\mathrm{Ma}^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}^2-1}\right)$$

Upstream:

$$\nu(\mathrm{Ma}_1) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(3.6^2-1)}\right) - \tan^{-1}\left(\sqrt{3.6^2-1}\right) = 60.09^\circ$$

Then the downstream Prandtl-Meyer function becomes

$$\nu(\mathrm{Ma}_2) = \theta + \nu(\mathrm{Ma}_1) = 15^\circ + 60.09^\circ = 75.09^\circ$$

Ma_2 is found from the Prandtl-Meyer relation, which is now implicit:

Downstream: $$\nu(\mathrm{Ma}_2) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}\mathrm{Ma}_2^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}_2^2-1}\right) = 75.09^\circ$$

Solution of this implicit equation gives $\mathrm{Ma}_2 =$ **4.81**. Then the downstream pressure and temperature are determined from the isentropic flow relations:

$$P_2 = \frac{P_2/P_0}{P_1/P_0}P_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-k/(k-1)}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-k/(k-1)}}P_1 = \frac{[1+4.81^2(1.4-1)/2]^{-1.4/0.4}}{[1+3.6^2(1.4-1)/2]^{-1.4/0.4}}(40\ \mathrm{kPa}) = \mathbf{8.31\ kPa}$$

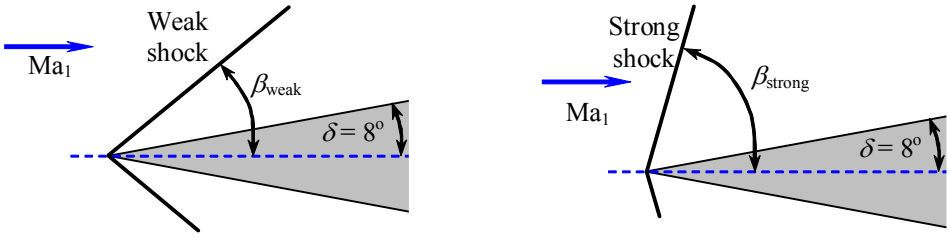
$$T_2 = \frac{T_2/T_0}{T_1/T_0}T_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-1}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-1}}T_1 = \frac{[1+4.81^2(1.4-1)/2]^{-1}}{[1+3.6^2(1.4-1)/2]^{-1}}(280\ \mathrm{K}) = \mathbf{179\ K}$$

Note that this is an expansion, and Mach number increases while pressure and temperature decrease, as expected.

Discussion There are compressible flow calculators on the Internet that solve these implicit equations that arise in the analysis of compressible flow, along with both normal and oblique shock equations; e.g., see www.aoe.vt.edu/~devenpor/aoe3114/calc.html .

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17-98 Air flowing at a specified supersonic Mach number impinges on a two-dimensional wedge, The shock angle, Mach number, and pressure downstream of the weak and strong oblique shock formed by a wedge are to be determined.



Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is $k = 1.4$.

Analysis On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 8^\circ$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k + \cos 2\beta) + 2} \quad \rightarrow \quad \tan 8^\circ = \frac{2(3.4^2 \sin^2\beta - 1)/\tan\beta}{3.4^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\boldsymbol{\beta_{weak} = 23.15^\circ}$ and $\boldsymbol{\beta_{strong} = 87.45^\circ}$. Then the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

Weak shock: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 3.4 \sin 23.15^\circ = 1.336$

Strong shock: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 3.4 \sin 87.45^\circ = 3.397$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

Weak shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.336)^2 + 2}{2(1.4)(1.336)^2 - 1.4 + 1}} = 0.7681$$

Strong shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(3.397)^2 + 2}{2(1.4)(3.397)^2 - 1.4 + 1}} = 0.4553$$

The downstream pressure for each case is determined to be

Weak shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (60\ \mathrm{kPa}) \frac{2(1.4)(1.336)^2 - 1.4 + 1}{1.4+1} = \boldsymbol{115.0\ \mathrm{kPa}}$$

Strong shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (60\ \mathrm{kPa}) \frac{2(1.4)(3.397)^2 - 1.4 + 1}{1.4+1} = \boldsymbol{797.6\ \mathrm{kPa}}$$

The downstream Mach number is determined to be

Weak shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.7681}{\sin(23.15^\circ - 8^\circ)} = \boldsymbol{2.94}$$

Strong shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.4553}{\sin(87.45^\circ - 8^\circ)} = \boldsymbol{0.463}$$

Discussion Note that the change in Mach number and pressure across the *strong shock* are much greater than the changes across the *weak shock*, as expected. For both the weak and strong oblique shock cases, $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ is subsonic. However, Ma_2 is *supersonic* across the weak oblique shock, but *subsonic* across the strong oblique shock.

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Duct Flow with Heat Transfer and Negligible Friction (Rayleigh Flow)

17-99C The characteristic aspect of Rayleigh flow is **its involvement of heat transfer**. The main assumptions associated with Rayleigh flow are: the flow is **steady**, **one-dimensional**, and **frictionless** through a constant-area duct, and the fluid is an **ideal gas with constant specific heats**.

Discussion Of course, there is no such thing as frictionless flow. It is better to say that frictional effects are negligible compared to the heating effects.

17-100C The points on the Rayleigh line represent the **states that satisfy the conservation of mass, momentum, and energy equations as well as the property relations for a given state**. Therefore, for a given inlet state, the fluid cannot exist at any downstream state outside the Rayleigh line on a *T*-*s* diagram.

Discussion The *T*-*s* diagram is quite useful, since any downstream state must lie on the Rayleigh line.

17-101C In Rayleigh flow, **the effect of heat gain is to increase the entropy** of the fluid, and **the effect of heat loss is to decrease the entropy**.

Discussion You should recall from thermodynamics that the entropy of a system can be lowered by removing heat.

17-102C In Rayleigh flow, the **stagnation temperature T_0 always increases with heat transfer to the fluid**, but the temperature *T* decreases with heat transfer in the Mach number range of $0.845 < \text{Ma} < 1$ for air. Therefore, the **temperature in this case will decrease**.

Discussion This at first seems counterintuitive, but if heat were *not* added, the temperature would drop even *more* if the air were accelerated isentropically from Ma = 0.92 to 0.95.

17-103C Heating the fluid **increases the flow velocity in subsonic Rayleigh flow**, but **decreases the flow velocity in supersonic Rayleigh flow**.

Discussion These results are not necessarily intuitive, but must be true in order to satisfy the conservation laws.

17-104C The flow is choked, and thus the flow at the duct exit **remains sonic**.

Discussion There is no mechanism for the flow to become supersonic in this case.

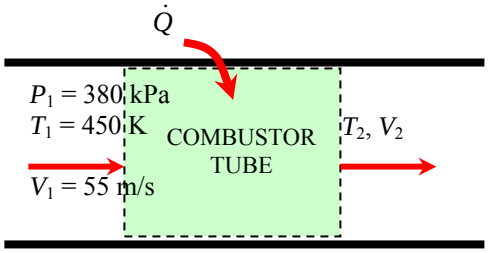
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17-105 Fuel is burned in a tubular combustion chamber with compressed air. For a specified exit Mach number, the exit temperature and the rate of fuel consumption are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Combustion is complete, and it is treated as a heat addition process, with no change in the chemical composition of flow. **3** The increase in mass flow rate due to fuel injection is disregarded.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis The inlet density and mass flow rate of air are



$$\rho_1 = \frac{P_1}{RT_1} = \frac{380\ \text{kPa}}{(0.287\ \text{kJ/kgK})(450\ \text{K})} = 2.942\ \text{kg/m}^3$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (2.942\ \text{kg/m}^3)[\pi(0.16\ \text{m})^2/4](55\ \text{m/s}) = 3.254\ \text{kg/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 450\ \text{K} + \frac{(55\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 451.5\ \text{K}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(450\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 425.2\ \text{m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{55\ \text{m/s}}{425.2\ \text{m/s}} = 0.1293$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34) (We used analytical functions):

$\text{Ma}_1 = 0.1293$: $T_1/T^* = 0.09201$, $T_{01}/T^* = 0.07693$, $V_1/V^* = 0.03923$

$\text{Ma}_2 = 0.8$: $T_2/T^* = 1.0255$, $T_{02}/T^* = 0.9639$, $V_2/V^* = 0.8101$

The exit temperature, stagnation temperature, and velocity are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1.0255}{0.09201} = 11.146 \quad \rightarrow \quad T_2 = 11.146T_1 = 11.146(450\ \text{K}) = \mathbf{5016\ K}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{0.9639}{0.07693} = 12.530 \rightarrow \quad T_{02} = 12.530T_{01} = 12.530(451.5\ \text{K}) = 5658\ \text{K}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{0.8101}{0.03923} = 20.650 \quad \rightarrow \quad V_2 = 20.650V_1 = 20.650(55\ \text{m/s}) = 1136\ \text{m/s}$$

Then the mass flow rate of the fuel is determined to be

$$q = c_p(T_{02} - T_{01}) = (1.005\ \text{kJ/kg}\cdot\text{K})(5658 - 451.5)\ \text{K} = 5232\ \text{kJ/kg}$$

$$\dot{Q} = \dot{m}_{\text{air}} q = (3.254\ \text{kg/s})(5232\ \text{kJ/kg}) = 17{,}024\ \text{kW}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}}{HV} = \frac{17{,}024\ \text{kJ/s}}{39{,}000\ \text{kJ/kg}} = \mathbf{0.4365\ kg/s}$$

Discussion Note that both the temperature and velocity increase during this subsonic Rayleigh flow with heating, as expected. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

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17-106 Air is heated in a duct during subsonic flow until it is choked. For specified pressure and velocity at the exit, the temperature, pressure, and velocity at the inlet are to be determined.

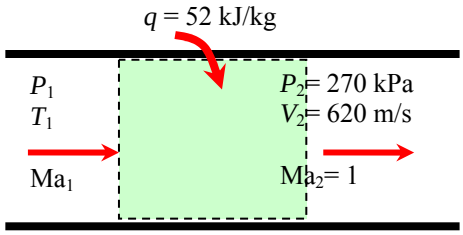
Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis Noting that sonic conditions exist at the exit, the exit temperature is

$$c_2 = V_2/\text{Ma}_2 = (620\text{ m/s})/1 = 620\text{ m/s}$$

$$c_2 = \sqrt{kRT_2} \rightarrow \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})T_2\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 620\text{ m/s}$$



It gives $T_2 = 956.7$ K. Then the exit stagnation temperature becomes

$$T_{02} = T_2 + \frac{V_2^2}{2c_p} = 956.7\text{ K} + \frac{(620\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 1148\text{ K}$$

The inlet stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{01} = T_{02} - \frac{q}{c_p} = 1148\text{ K} - \frac{52\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 1096\text{ K}$$

The maximum value of stagnation temperature T_0^* occurs at Ma = 1, and its value in this case is T_{02} since the flow is choked. Therefore, $T_0^* = T_{02} = 1148$ K. Then the stagnation temperature ratio at the inlet, and the Mach number corresponding to it are, from Table A-34,

$$\frac{T_{01}}{T_0^*} = \frac{1096\text{ K}}{1148\text{ K}} = 0.9547 \quad \rightarrow \quad \text{Ma}_1 = 0.7792 \cong \mathbf{0.779}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 0.7792$: $T_1/T^* = 1.022$, $P_1/P^* = 1.297$, $V_1/V^* = 0.7877$

$\text{Ma}_2 = 1$: $T_2/T^* = 1$, $P_2/P^* = 1$, $V_2/V^* = 1$

Then the inlet temperature, pressure, and velocity are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{1.017} \quad \rightarrow \quad T_1 = 1.022T_2 = 1.022(956.7\text{ K}) = \mathbf{977.5\text{ K}}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{1.319} \quad \rightarrow \quad P_1 = 1.319P_2 = 1.297(270\text{ kPa}) = \mathbf{350.3\text{ kPa}}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{1}{0.7719} \quad \rightarrow \quad V_1 = 0.7877V_2 = 0.7877(620\text{ m/s}) = \mathbf{488.4\text{ m/s}}$$

Discussion Note that the temperature and pressure decreases with heating during this subsonic Rayleigh flow while velocity increases. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

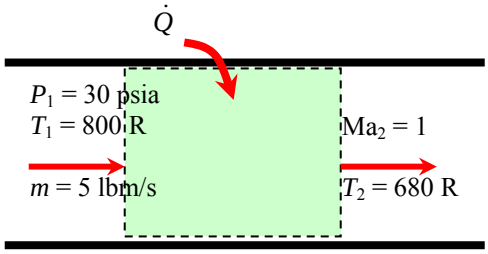
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17-107E Air flowing with a subsonic velocity in a round duct is accelerated by heating until the flow is choked at the exit. The rate of heat transfer and the pressure drop are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The flow is choked at the duct exit. **3** Mass flow rate remains constant.

Properties We take the properties of air to be $k = 1.4$, $c_p = 0.2400$ Btu/lbm·R, and $R = 0.06855$ Btu/lbm·R = 0.3704 psia·ft^3/lbm·R.

Analysis The inlet density and velocity of air are



$$\rho_1 = \frac{P_1}{RT_1} = \frac{30 \text{ psia}}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(800 \text{ R})} = 0.1012 \text{ lbm/ft}^3$$

$$V_1 = \frac{\dot{m}_{air}}{\rho_1 A_{c1}} = \frac{5 \text{ lbm/s}}{(0.1012 \text{ lbm/ft}^3)[\pi(4/12 \text{ ft})^2/4]} = 565.9 \text{ ft/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 800 \text{ R} + \frac{(565.9 \text{ ft/s})^2}{2\times 0.2400 \text{ Btu/lbm}\cdot\text{R}}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 826.7 \text{ R}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.06855 \text{ Btu/lbm}\cdot\text{R})(800 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = 1386 \text{ ft/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{565.9 \text{ ft/s}}{1386 \text{ ft/s}} = 0.4082$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 0.4082$: $T_1/T^* = 0.6310$, $P_1/P^* = 1.946$, $T_{01}/T_0^* = 0.5434$

$\text{Ma}_2 = 1$: $T_2/T^* = 1$, $P_2/P^* = 1$, $T_{02}/T_0^* = 1$

Then the exit temperature, pressure, and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.6310} \quad \rightarrow \quad T_2 = T_1/0.6310 = (800 \text{ R})/0.6310 = 1268 \text{ R}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{1.946} \quad \rightarrow \quad P_2 = P_1/2.272 = (30 \text{ psia})/1.946 = 15.4 \text{ psia}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.5434} \quad \rightarrow \quad T_{02} = T_{01}/0.1743 = (826.7 \text{ R})/0.5434 = 1521 \text{ R}$$

Then the rate of heat transfer and the pressure drop become

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (5 \text{ lbm/s})(0.2400 \text{ Btu/lbm}\cdot\text{R})(1521 - 826.7) \text{ R} = \mathbf{834 \text{ Btu/s}}$$

$$\Delta P = P_1 - P_2 = 30 - 15.4 = \mathbf{14.6 \text{ psia}}$$

Discussion Note that the entropy of air increases during this heating process, as expected.

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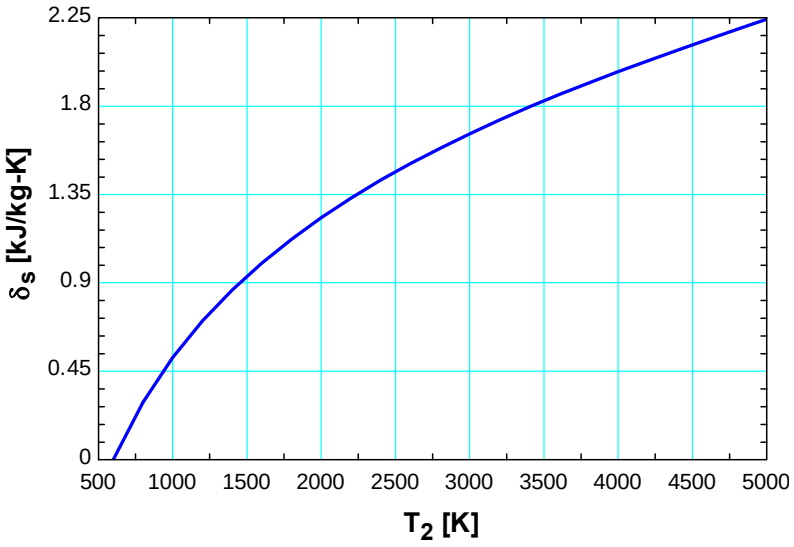
17-108 Air flowing with a subsonic velocity in a duct. The variation of entropy with temperature is to be investigated as the exit temperature varies from 600 K to 5000 K in increments of 200 K. The results are to be tabulated and plotted.

Analysis We solve this problem using EES making use of Rayleigh functions. The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
k=1.4
cp=1.005
R=0.287
P1=350
T1=600
V1=70
C1=sqrt(k*R*T1*1000)
Ma1=V1/C1
T01=T1*(1+0.5*(k-1)*Ma1^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))
F1=1+0.5*(k-1)*Ma1^2
T01Ts=2*(k+1)*Ma1^2*F1/(1+k*Ma1^2)^2
P01Ps=((1+k)/(1+k*Ma1^2))*(2*F1/(k+1))^(k/(k-1))
T1Ts=(Ma1*((1+k)/(1+k*Ma1^2)))^2
P1Ps=(1+k)/(1+k*Ma1^2)
V1Vs=Ma1^2*(1+k)/(1+k*Ma1^2)
F2=1+0.5*(k-1)*Ma2^2
T02Ts=2*(k+1)*Ma2^2*F2/(1+k*Ma2^2)^2
P02Ps=((1+k)/(1+k*Ma2^2))*(2*F2/(k+1))^(k/(k-1))
T2Ts=(Ma2*((1+k)/(1+k*Ma2^2)))^2
P2Ps=(1+k)/(1+k*Ma2^2)
V2Vs=Ma2^2*(1+k)/(1+k*Ma2^2)
T02=T02Ts/T01Ts*T01
P02=P02Ps/P01Ps*P01
T2=T2Ts/T1Ts*T1
P2=P2Ps/P1Ps*P1
V2=V2Vs/V1Vs*V1
Delta_s=cp*ln(T2/T1)-R*ln(P2/P1)
```

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Exit temperature T_2, K	Exit Mach number, Ma_2	Exit entropy relative to inlet, s_2, kJ/kg·K
600.1	0.143	0.000
800	0.166	0.292
1000	0.188	0.519
1200	0.208	0.705
1400	0.227	0.863
1600	0.245	1.001
1800	0.263	1.123
2000	0.281	1.232
2200	0.299	1.331
2400	0.316	1.423
2600	0.333	1.507
2800	0.351	1.586
3000	0.369	1.660
3200	0.387	1.729
3400	0.406	1.795
3600	0.426	1.858
3800	0.446	1.918
4000	0.467	1.975
4200	0.490	2.031
4400	0.515	2.085
4600	0.541	2.138
4800	0.571	2.190
5000	0.606	2.242



Discussion Note that the entropy of air increases during this heating process, as expected.

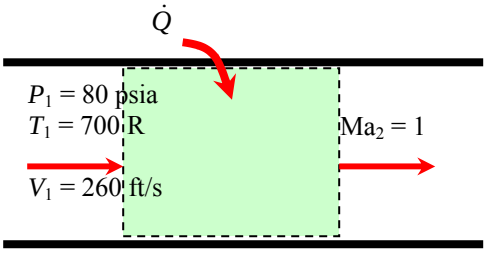
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17-109E Air flowing with a subsonic velocity in a square duct is accelerated by heating until the flow is choked at the exit. The rate of heat transfer and the entropy change are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The flow is choked at the duct exit. **3** Mass flow rate remains constant.

Properties We take the properties of air to be $k = 1.4$, $c_p = 0.2400$ Btu/lbm·R, and $R = 0.06855$ Btu/lbm·R = 0.3704 psia·ft^3/lbm·R.

Analysis The inlet density and mass flow rate of air are



$$\rho_1 = \frac{P_1}{RT_1} = \frac{80 \text{ psia}}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(700 \text{ R})} = 0.3085 \text{ lbm/ft}^3$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (0.3085 \text{ lbm/ft}^3)(6 \times 6/144 \text{ ft}^2)(260 \text{ ft/s}) = 20.06 \text{ lbm/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 700 \text{ R} + \frac{(260 \text{ ft/s})^2}{2 \times 0.2400 \text{ Btu/lbm} \cdot \text{R}} \left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2} \right) = 705.6 \text{ R}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.06855 \text{ Btu/lbm} \cdot \text{R})(700 \text{ R}) \left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right)} = 1297 \text{ ft/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{260 \text{ ft/s}}{1297 \text{ ft/s}} = 0.2005$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 0.2005$: $T_1/T^* = 0.2075$, $P_1/P^* = 2.272$, $T_{01}/T_0^* = 0.1743$

$\text{Ma}_2 = 1$: $T_2/T^* = 1$, $P_2/P^* = 1$, $T_{02}/T_0^* = 1$

Then the exit temperature, pressure, and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.2075} \quad \rightarrow \quad T_2 = T_1/0.2075 = (700 \text{ R})/0.2075 = 3374 \text{ R}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{2.272} \quad \rightarrow \quad P_2 = P_1/2.272 = (80 \text{ psia})/2.272 = 35.2 \text{ psia}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.1743} \quad \rightarrow \quad T_{02} = T_{01}/0.1743 = (705.6 \text{ R})/0.1743 = 4048 \text{ R}$$

Then the rate of heat transfer and entropy change become

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (20.06 \text{ lbm/s})(0.2400 \text{ Btu/lbm} \cdot \text{R})(4048 - 705.6) \text{ R} = \mathbf{16{,}090 \ Btu/s}$$

$$\Delta s = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}$$

$$= (0.2400 \text{ Btu/lbm} \cdot \text{R}) \ln \frac{3374 \text{ R}}{700 \text{ R}} - (0.06855 \text{ Btu/lbm} \cdot \text{R}) \ln \frac{35.2 \text{ psia}}{80 \text{ psia}} = \mathbf{0.434 \ Btu/lbm \cdot R}$$

Discussion Note that the entropy of air increases during this heating process, as expected.

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17-110 Air enters the combustion chamber of a gas turbine at a subsonic velocity. For a specified rate of heat transfer, the Mach number at the exit and the loss in stagnation pressure to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The cross-sectional area of the combustion chamber is constant. **3** The increase in mass flow rate due to fuel injection is disregarded.

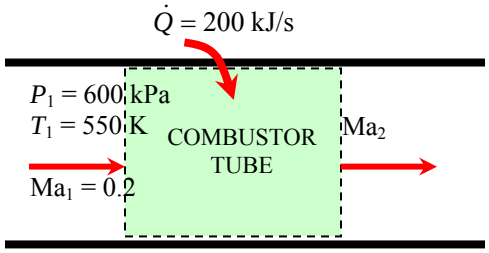
Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis The inlet stagnation temperature and pressure are

$$T_{01} = T_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right) = (550\text{ K})\left(1+\frac{1.4-1}{2}0.2^2\right) = 554.4\text{ K}$$

$$P_{01} = P_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)^{k/(k-1)} = (600\text{ kPa})\left(1+\frac{1.4-1}{2}0.2^2\right)^{1.4/0.4}$$

$$= 617.0\text{ kPa}$$



The exit stagnation temperature is determined from

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) \quad \rightarrow \quad 200\text{ kJ/s} = (0.3\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(T_{02} - 554.4)\text{ K}$$

It gives

$$T_{02} = 1218\text{ K}.$$

At $\text{Ma}_1 = 0.2$ we read from $T_{01}/T_0^* = 0.1736$ (Table A-34). Therefore,

$$T_0^* = \frac{T_{01}}{0.1736} = \frac{554.4\text{ K}}{0.1736} = 3193.5\text{ K}$$

Then the stagnation temperature ratio at the exit and the Mach number corresponding to it are (Table A-34)

$$\frac{T_{02}}{T_0^*} = \frac{1218\text{ K}}{3193.5\text{ K}} = 0.3814 \quad \rightarrow \quad \text{Ma}_2 = 0.3187 \cong \mathbf{0.319}$$

Also,

$$\text{Ma}_1 = 0.2 \quad \rightarrow P_{01}/P_0^* = 1.2346$$

$$\text{Ma}_2 = 0.3187 \quad \rightarrow P_{02}/P_0^* = 1.191$$

Then the stagnation pressure at the exit and the pressure drop become

$$\frac{P_{02}}{P_{01}} = \frac{P_{02}/P_0^*}{P_{01}/P_0^*} = \frac{1.191}{1.2346} = 0.9647 \quad \rightarrow \quad P_{02} = 0.9647P_{01} = 0.9647(617\text{ kPa}) = 595.2\text{ kPa}$$

and

$$\Delta P_0 = P_{01} - P_{02} = 617.0 - 595.2 = \mathbf{21.8\text{ kPa}}$$

Discussion This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

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17-111 Air enters the combustion chamber of a gas turbine at a subsonic velocity. For a specified rate of heat transfer, the Mach number at the exit and the loss in stagnation pressure to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The cross-sectional area of the combustion chamber is constant. **3** The increase in mass flow rate due to fuel injection is disregarded.

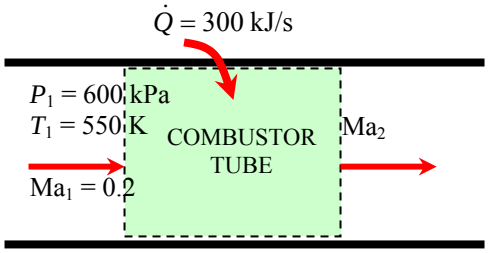
Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis The inlet stagnation temperature and pressure are

$$T_{01} = T_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right) = (550\text{ K})\left(1+\frac{1.4-1}{2}0.2^2\right) = 554.4\text{ K}$$

$$P_{01} = P_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)^{k/(k-1)} = (600\text{ kPa})\left(1+\frac{1.4-1}{2}0.2^2\right)^{1.4/0.4}$$

$$= 617.0\text{ kPa}$$



The exit stagnation temperature is determined from

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) \quad \rightarrow \quad 300\text{ kJ/s} = (0.3\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(T_{02} - 554.4)\text{ K}$$

It gives

$$T_{02} = 1549\text{ K}.$$

At $\text{Ma}_1 = 0.2$ we read from $T_{01}/T_0^* = 0.1736$ (Table A-34). Therefore,

$$T_0^* = \frac{T_{01}}{0.1736} = \frac{554.4\text{ K}}{0.1736} = 3193.5\text{ K}$$

Then the stagnation temperature ratio at the exit and the Mach number corresponding to it are (Table A-34)

$$\frac{T_{02}}{T_0^*} = \frac{1549\text{ K}}{3193.5\text{ K}} = 0.4850 \quad \rightarrow \quad \text{Ma}_2 = 0.3753 \cong \mathbf{0.375}$$

Also,

$$\text{Ma}_1 = 0.2 \quad \rightarrow P_{01}/P_0^* = 1.2346$$

$$\text{Ma}_2 = 0.3753 \quad \rightarrow P_{02}/P_0^* = 1.167$$

Then the stagnation pressure at the exit and the pressure drop become

$$\frac{P_{02}}{P_{01}} = \frac{P_{02}/P_0^*}{P_{01}/P_0^*} = \frac{1.167}{1.2346} = 0.9452 \quad \rightarrow \quad P_{02} = 0.9452P_{01} = 0.9452(617\text{ kPa}) = 583.3\text{ kPa}$$

and

$$\Delta P_0 = P_{01} - P_{02} = 617.0 - 583.3 = \mathbf{33.7\text{ kPa}}$$

Discussion This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

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17-112 Fuel is burned in a rectangular duct with compressed air. For specified heat transfer, the exit temperature and Mach number are to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

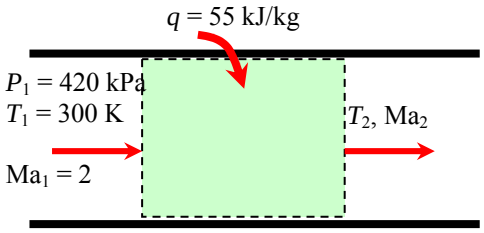
Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis The stagnation temperature and Mach number at the inlet are

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(300\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 347.2\ \text{m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 2(347.2\ \text{m/s}) = 694.4\ \text{m/s}$$

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 300\ \text{K} + \frac{(694.4\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 539.9\ \text{K}$$



The exit stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{02} = T_{01} + \frac{q}{c_p} = 539.9\ \text{K} + \frac{55\ \text{kJ/kg}}{1.005\ \text{kJ/kg}\cdot\text{K}} = 594.6\ \text{K}$$

The maximum value of stagnation temperature T_0^* occurs at Ma = 1, and its value can be determined from Table A-34 or from the appropriate relation. At $\text{Ma}_1 = 2$ we read $T_{01}/T_0^* = 0.7934$. Therefore,

$$T_0^* = \frac{T_{01}}{0.7934} = \frac{539.9\ \text{K}}{0.7934} = 680.5\ \text{K}$$

The stagnation temperature ratio at the exit and the Mach number corresponding to it are, from Table A-34,

$$\frac{T_{02}}{T_0^*} = \frac{594.6\ \text{K}}{680.5\ \text{K}} = 0.8738 \quad \rightarrow \quad \text{Ma}_2 = 1.642 \cong \mathbf{1.64}$$

Also,

$$\text{Ma}_1 = 2 \quad \rightarrow \quad T_1/T^* = 0.5289$$

$$\text{Ma}_2 = 1.642 \quad \rightarrow \quad T_2/T^* = 0.6812$$

Then the exit temperature becomes

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.6812}{0.5289} = 1.288 \quad \rightarrow \quad T_2 = 1.288T_1 = 1.288(300\ \text{K}) = \mathbf{386\ K}$$

Discussion Note that the temperature increases during this supersonic Rayleigh flow with heating. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

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17-113 Compressed air is cooled as it flows in a rectangular duct. For specified heat rejection, the exit temperature and Mach number are to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

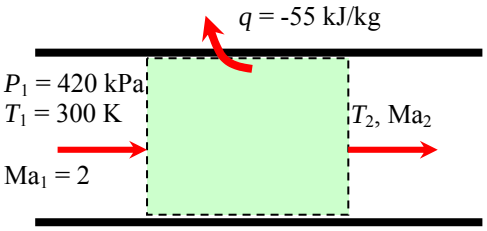
Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis The stagnation temperature and Mach number at the inlet are

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(300\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 347.2\ \text{m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 2(347.2\ \text{m/s}) = 694.4\ \text{m/s}$$

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 300\ \text{K} + \frac{(694.4\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 539.9\ \text{K}$$



The exit stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{02} = T_{01} + \frac{q}{c_p} = 539.9\ \text{K} + \frac{-55\ \text{kJ/kg}}{1.005\ \text{kJ/kg}\cdot\text{K}} = 485.2\ \text{K}$$

The maximum value of stagnation temperature T_0^* occurs at Ma = 1, and its value can be determined from Table A-34 or from the appropriate relation. At $\text{Ma}_1 = 2$ we read $T_{01}/T_0^* = 0.7934$. Therefore,

$$T_0^* = \frac{T_{01}}{0.7934} = \frac{539.9\ \text{K}}{0.7934} = 680.5\ \text{K}$$

The stagnation temperature ratio at the exit and the Mach number corresponding to it are, from Table A-34,

$$\frac{T_{02}}{T_0^*} = \frac{485.2\ \text{K}}{680.5\ \text{K}} = 0.7130 \quad \rightarrow \quad \text{Ma}_2 = 2.479 \cong \mathbf{2.48}$$

Also,

$$\text{Ma}_1 = 2 \quad \rightarrow \quad T_1/T^* = 0.5289$$

$$\text{Ma}_2 = 2.479 \quad \rightarrow \quad T_2/T^* = 0.3838$$

Then the exit temperature becomes

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.3838}{0.5289} = 0.7257 \quad \rightarrow \quad T_2 = 0.7257T_1 = 0.7257(300\ \text{K}) = \mathbf{218\ K}$$

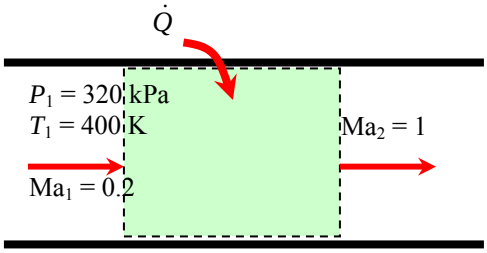
Discussion Note that the temperature decreases and Mach number increases during this supersonic Rayleigh flow with cooling. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

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17-114 Argon flowing at subsonic velocity in a constant-diameter duct is accelerated by heating. The highest rate of heat transfer without reducing the mass flow rate is to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Mass flow rate remains constant.

Properties We take the properties of argon to be $k = 1.667$, $c_p = 0.5203$ kJ/kg·K, and $R = 0.2081$ kJ/kg·K.



Analysis Heat transfer stops when the flow is choked, and thus $Ma_2 = V_2/c_2 = 1$. The inlet stagnation temperature is

$$T_{01} = T_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right) = (400\text{ K})\left(1+\frac{1.667\text{-}1}{2}0.2^2\right) = 405.3\text{ K}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are

$$T_{02}/T_0^* = 1 \text{ (since Ma}_2 = 1)$$

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\text{Ma}_1^2[2+(k-1)\text{Ma}_1^2]}{(1+k\text{Ma}_1^2)^2} = \frac{(1.667+1)0.2^2[2+(1.667-1)0.2^2]}{(1+1.667\times 0.2^2)^2} = 0.1900$$

Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.1900} \quad \rightarrow \quad T_{02} = T_{01}/0.1900 = (405.3\text{ K})/0.1900 = 2133\text{ K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\text{air}}c_p(T_{02}-T_{01}) = (1.2\text{ kg/s})(0.5203\text{ kJ/kg}\cdot\text{K})(2133-400)\text{ K} = \mathbf{1080\ kW}$$

Discussion It can also be shown that $T_2 = 1600$ K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. Also, in the solution of this problem, we cannot use the values of Table A-34 since they are based on $k = 1.4$.

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17-115 Air flowing at a supersonic velocity in a duct is decelerated by heating. The highest temperature air can be heated by heat addition and the rate of heat transfer are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Mass flow rate remains constant.

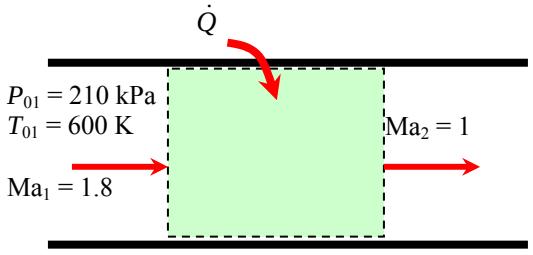
Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis Heat transfer will stop when the flow is choked, and thus $\text{Ma}_2 = V_2/c_2 = 1$. Knowing stagnation properties, the static properties are determined to be

$$T_1 = T_{01}\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)^{-1} = (600\text{ K})\left(1+\frac{1.4-1}{2}1.8^2\right)^{-1} = 364.1\text{ K}$$

$$P_1 = P_{01}\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)^{-k/(k-1)} = (210\text{ kPa})\left(1+\frac{1.4-1}{2}1.8^2\right)^{-1.4/0.4} = 36.55\text{ kPa}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{36.55\text{ kPa}}{(0.287\text{ kJ/kgK})(364.1\text{ K})} = 0.3498\text{ kg/m}^3$$



Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(364.1\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 382.5\text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 1.8(382.5\text{ m/s}) = 688.5\text{ m/s}$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (0.3498\text{ kg/m}^3)[\pi(0.10\text{ m})^2/4](688.5\text{ m/s}) = 1.891\text{ kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 1.8$: $\quad T_1/T^* = 0.6089, \quad T_{01}/T_0^* = 0.8363$

$\text{Ma}_2 = 1$: $T_2/T^* = 1, \quad T_{02}/T_0^* = 1$

Then the exit temperature and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.6089} \quad \rightarrow \quad T_2 = T_1/0.6089 = (364.1\text{ K})/0.6089 = \mathbf{598\text{ K}}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.8363} \quad \rightarrow \quad T_{02} = T_{01}/0.8363 = (600\text{ K})/0.8363 = 717.4\text{ K} \cong \mathbf{717\text{ K}}$$

Finally, the rate of heat transfer is

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (1.891\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(717.4 - 600)\text{ K} = \mathbf{223\text{ kW}}$$

Discussion Note that this is the highest temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature will cause the mass flow rate to decrease. Also, once the sonic conditions are reached, the thermodynamic temperature can be increased further by cooling the fluid and reducing the velocity (see the *T*-*s* diagram for Rayleigh flow).

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Steam Nozzles

17-116C The delay in the condensation of the steam is called supersaturation. It occurs in high-speed flows where there isn't sufficient time for the necessary heat transfer and the formation of liquid droplets.

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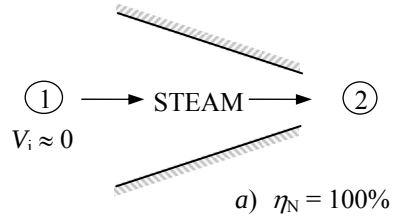
17-117 Steam enters a converging nozzle with a low velocity. The exit velocity, mass flow rate, and exit Mach number are to be determined for isentropic and 90 percent efficient nozzle cases.

Assumptions **1** Flow through the nozzle is steady and one-dimensional. **2** The nozzle is adiabatic.

Analysis (*a*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,

$$\left.\begin{aligned} P_1 &= P_{01} = 4\ \text{MPa} \\ T_1 &= T_{01} = 400°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= h_{01} = 3214.5\ \text{kJ/kg} \\ s_1 &= s_{2s} = 6.7714\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$



At the exit,

$$\left.\begin{aligned} P_2 &= 2.5\ \text{MPa} \\ s_2 &= 6.7714\ \text{kJ/kg}\cdot\text{K} \end{aligned}\right\} \begin{aligned} h_2 &= 3083.4\ \text{kJ/kg} \\ \nu_2 &= 0.1058\ \text{m}^3\text{/kg} \end{aligned}$$

a) $\eta_N = 100\%$
b) $\eta_N = 94\%$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3214.5 - 3083.4)\ \text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{512.0\ m/s}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 = \frac{1}{0.1058\ \text{m}^3\text{/kg}}(32\times10^{-4}\ \text{m}^2)(512.0\ \text{m/s}) = \mathbf{15.49\ kg/s}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\nu)}\right)_s^{1/2}$$

The specific volume of steam at $s_2 = 6.7714$ kJ/kg·K and at pressures just below and just above the specified pressure (2.0 and 3.0 MPa) are determined to be 0.1257 and 0.09183 m^3/kg. Substituting,

$$c_2 = \sqrt{\frac{(3000 - 2000)\ \text{kPa}}{\left(\frac{1}{0.09183} - \frac{1}{0.1257}\right)\text{kg/m}^3}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kPa.m}^3}\right)} = 583.7\ \text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{512.0\ \text{m/s}}{583.7\ \text{m/s}} = \mathbf{0.877}$$

(*b*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,

$$\left.\begin{aligned} P_1 &= P_{01} = 4\ \text{MPa} \\ T_1 &= T_{01} = 400°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= h_{01} = 3214.5\ \text{kJ/kg} \\ s_1 &= s_{2s} = 6.7714\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

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At state 2s,

$$\left.\begin{array}{l} P_{2s} = 2.5\ \text{MPa} \\ s_{2s} = 6.7714\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} h_{2s} = 3083.4\ \text{kJ/kg}$$

The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.94 = \frac{3214.5 - h_2}{3214.5 - 3288.7} \longrightarrow h_2 = 3091.3\ \text{kJ/kg}$$

Therefore,

$$\left.\begin{array}{l} P_2 = 2.5\ \text{MPa} \\ h_2 = 3091.3\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} \nu_2 = 0.1065\ \text{m}^3\text{/kg} \\ s_2 = 6.7844\ \text{kJ/kg}\cdot\text{K} \end{array}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3214.5 - 3091.3)\ \text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{496.4\ m/s}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 = \frac{1}{0.1065\ \text{m}^3/\text{kg}}(32\times 10^{-4}\ \text{m}^2)(496.4\ \text{m/s}) = \mathbf{14.92\ kg/s}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\nu)}\right)_s^{1/2}$$

The specific volume of steam at s_2 = 6.7844 kJ/kg·K and at pressures just below and just above the specified pressure (2.0 and 3.0 MPa) are determined to be 0.1266 and 0.09246 m^3/kg. Substituting,

$$c_2 = \sqrt{\frac{(3000 - 2000)\ \text{kPa}}{\left(\frac{1}{0.09246} - \frac{1}{0.1266}\right)\text{kg/m}^3}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kPa.m}^3}\right)} = 585.7\ \text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{496.4\ \text{m/s}}{585.7\ \text{m/s}} = \mathbf{0.848}$$

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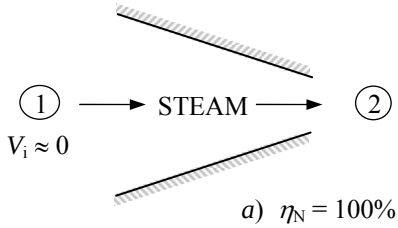
17-118E Steam enters a converging nozzle with a low velocity. The exit velocity, mass flow rate, and exit Mach number are to be determined for isentropic and 90 percent efficient nozzle cases.

Assumptions **1** Flow through the nozzle is steady and one-dimensional. **2** The nozzle is adiabatic.

Analysis (*a*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,

$$\left.\begin{aligned} P_1 = P_{01} &= 450 \text{ psia} \\ T_1 = T_{01} &= 900°\text{F} \end{aligned}\right\} \begin{aligned} h_1 = h_{01} &= 1468.6 \text{ Btu/lbm} \\ s_1 = s_{2s} &= 1.7117 \text{ Btu/lbm} \cdot \text{R} \end{aligned}$$



a) $\eta_N = 100\%$
b) $\eta_N = 90\%$

At the exit,

$$\left.\begin{aligned} P_2 &= 275 \text{ psia} \\ s_{2s} &= 1.7117 \text{ Btu/lbm} \cdot \text{R} \end{aligned}\right\} \begin{aligned} h_2 &= 1400.5 \text{ Btu/lbm} \\ \nu_2 &= 2.5732 \text{ ft}^3/\text{lbm} \end{aligned}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{in} = \dot{E}_{out}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(1468.6 - 1400.5) \text{ Btu/lbm} \left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right)} = \mathbf{1847 \text{ ft/s}}$$

Then,

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 = \frac{1}{2.5732 \text{ ft}^3/\text{lbm}} (3.75/144 \text{ ft}^2)(1847 \text{ ft/s}) = \mathbf{18.7 \text{ lbm/s}}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho} \right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\nu)} \right)_s^{1/2}$$

The specific volume of steam at $s_2 = 1.7117$ Btu/lbm·R and at pressures just below and just above the specified pressure (250 and 300 psia) are determined to be 2.7709 and 2.4048 ft^3/lbm. Substituting,

$$c_2 = \sqrt{\frac{(300 - 250) \text{ psia}}{\left(\frac{1}{2.4048} - \frac{1}{2.7709} \right) 1 \text{bm/ft}^3} \left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right) \left(\frac{1 \text{ Btu}}{5.4039 \text{ ft}^3 \cdot \text{psia}} \right)} = 2053 \text{ ft/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{1847 \text{ ft/s}}{2053 \text{ ft/s}} = \mathbf{0.900}$$

(*b*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,

$$\left.\begin{aligned} P_1 = P_{01} &= 450 \text{ psia} \\ T_1 = T_{01} &= 900°\text{F} \end{aligned}\right\} \begin{aligned} h_1 = h_{01} &= 1468.6 \text{ Btu/lbm} \\ s_1 = s_{2s} &= 1.7117 \text{ Btu/lbm} \cdot \text{R} \end{aligned}$$

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At state 2s,

$$\left.\begin{array}{l} P_{2s} = 275 \text{ psia} \\ s_{2s} = 1.7117 \text{ Btu/lbm} \cdot \text{R} \end{array}\right\} h_{2s} = 1400.5 \text{ Btu/lbm}$$

The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.90 = \frac{1468.6 - h_2}{1468.6 - 1400.5} \longrightarrow h_2 = 1407.3 \text{ Btu/1bm}$$

Therefore,

$$\left.\begin{array}{l} P_2 = 275 \text{ psia} \\ h_2 = 1407.3 \text{ Btu/lbm} \end{array}\right\} \begin{array}{l} \boldsymbol{\nu}_2 = 2.6034 \text{ ft}^3\text{/1bm} \\ \text{s}_2 = 1.7173 \text{ Btu/1bm} \cdot \text{R} \end{array}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(1468.6 - 1407.3) \text{ Btu/1bm} \left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/1bm}}\right)} = \mathbf{1752 \text{ ft/s}}$$

Then,

$$\dot{m} = \frac{1}{\boldsymbol{\nu}_2} A_2 V_2 = \frac{1}{2.6034 \text{ ft}^3/\text{1bm}} (3.75/144 \text{ ft}^2)(1752 \text{ ft/s}) = \mathbf{17.53 \text{ 1bm/s}}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{\nu})}\right)_s^{1/2}$$

The specific volume of steam at s_2 = 1.7173 Btu/lbm·R and at pressures just below and just above the specified pressure (250 and 300 psia) are determined to be 2.8036 and 2.4329 ft^3/lbm. Substituting,

$$c_2 = \sqrt{\frac{(300 - 250) \text{ psia}}{\left(\frac{1}{2.4329} - \frac{1}{2.8036}\right) \text{lbm/ft}^3} \left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right) \left(\frac{1 \text{ Btu}}{5.4039 \text{ ft}^3 \cdot \text{psia}}\right)} = 2065 \text{ ft/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{1752 \text{ ft/s}}{2065 \text{ ft/s}} = \mathbf{0.849}$$

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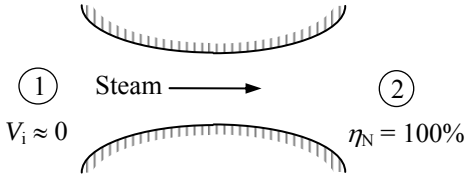
17-119 Steam enters a converging-diverging nozzle with a low velocity. The exit area and the exit Mach number are to be determined.

Assumptions Flow through the nozzle is steady, one-dimensional, and isentropic.

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,

$$\left.\begin{aligned} P_1 = P_{01} = 1\,\text{MPa} \\ T_1 = T_{01} = 500°\text{C} \end{aligned}\right\} \begin{aligned} h_1 = h_{01} = 3479.1\,\text{kJ/kg} \\ s_1 = s_{2s} = 7.7642\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$



At the exit,

$$\left.\begin{aligned} P_2 = 0.2\,\text{MPa} \\ s_2 = 7.7642\,\text{kJ/kg}\cdot\text{K} \end{aligned}\right\} \begin{aligned} h_2 = 3000.0\,\text{kJ/kg} \\ \nu_2 = 1.2325\,\text{m}^3\text{/kg} \end{aligned}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3479.1 - 3000.0)\,\text{kJ/kg}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 978.9\,\text{m/s}$$

The exit area is determined from

$$A_2 = \frac{\dot{m}\nu_2}{V_2} = \frac{(2.5\,\text{kg/s})(1.2325\,\text{m}^3/\text{kg})}{(978.9\,\text{m/s})} = 31.5\times10^{-4}\,\text{m}^2 = \mathbf{31.5\,cm^2}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\nu)}\right)_s^{1/2}$$

The specific volume of steam at $s_2 = 7.7642$ kJ/kg·K and at pressures just below and just above the specified pressure (0.1 and 0.3 MPa) are determined to be 2.0935 and 0.9024 m^3/kg. Substituting,

$$c_2 = \sqrt{\frac{(300-100)\,\text{kPa}}{\left(\frac{1}{0.9024} - \frac{1}{2.0935}\right)\text{kg/m}^3}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kPa}\cdot\text{m}^3}\right)} = 563.2\,\text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{978.9\,\text{m/s}}{563.2\,\text{m/s}} = \mathbf{1.738}$$

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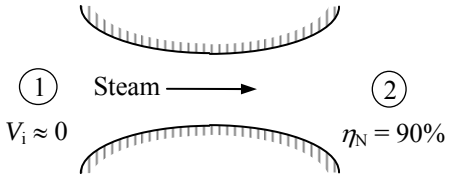
17-120 Steam enters a converging-diverging nozzle with a low velocity. The exit area and the exit Mach number are to be determined.

Assumptions Flow through the nozzle is steady and one-dimensional.

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet, $\left.\begin{matrix} P_1 = P_{01} = 1\,\text{MPa} \\ T_1 = T_{01} = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_1 = h_{01} = 3479.1\,\text{kJ/kg} \\ s_1 = s_{2s} = 7.7642\,\text{kJ/kg}\cdot\text{K} \end{matrix}$

At state 2s, $\left.\begin{matrix} P_2 = 0.2\,\text{MPa} \\ s_2 = 7.7642\,\text{kJ/kg}\cdot\text{K} \end{matrix}\right\} h_{2s} = 3000.0\,\text{kJ/kg}$



The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.90 = \frac{3479.1 - h_2}{3479.1 - 3000.0} \longrightarrow h_2 = 3047.9\,\text{kJ/kg}$$

Therefore,

$$\left.\begin{matrix} P_2 = 0.2\,\text{MPa} \\ h_2 = 3047.9\,\text{kJ/kg} \end{matrix}\right\} \begin{matrix} \nu_2 = 1.2882\,\text{m}^3\text{/kg} \\ s_2 = 7.7642\,\text{kJ/kg}\cdot\text{K} \end{matrix}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3479.1 - 3047.9)\,\text{kJ/kg}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 928.7\,\text{m/s}$$

The exit area is determined from

$$A_2 = \frac{\dot{m}\nu_2}{V_2} = \frac{(2.5\,\text{kg/s})(1.2882\,\text{m}^3\text{/kg})}{928.7\,\text{m/s}} = 34.7\times10^{-4}\ \text{m}^2 = \mathbf{34.7\,cm^2}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\nu)}\right)_s^{1/2}$$

The specific volume of steam at $s_2 = 7.7642$ kJ/kg·K and at pressures just below and just above the specified pressure (0.1 and 0.3 MPa) are determined to be 2.1903 and 0.9425 m^3/kg. Substituting,

$$c_2 = \sqrt{\frac{(300 - 100)\,\text{kPa}}{\left(\frac{1}{0.9425} - \frac{1}{2.1903}\right)\text{kg/m}^3}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kPa}\cdot\text{m}^3}\right)} = 575.2\,\text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{928.7\,\text{m/s}}{575.2\,\text{m/s}} = \mathbf{1.61}$$

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Review Problems

17-121 A leak develops in an automobile tire as a result of an accident. The initial mass flow rate of air through the leak is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow of air through the hole is isentropic.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K. The specific heat ratio of air at room temperature is $k = 1.4$ (Table A-2a).

Analysis The absolute pressure in the tire is

$$P = P_{\text{gage}} + P_{\text{atm}} = 220 + 94 = 314 \text{ kPa}$$

The critical pressure is, from Table 17-2,

$$P^* = 0.5283 P_0 = (0.5283)(314 \text{ kPa}) = 166 \text{ kPa} > 94 \text{ kPa}$$

Therefore, the flow is choked, and the velocity at the exit of the hole is the sonic speed. Then the flow properties at the exit becomes

$$\rho_0 = \frac{P_0}{RT_0} = \frac{314 \text{ kPa}}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = 3.671 \text{ kg/m}^3$$

$$\rho^* = \rho_0 \left(\frac{2}{k+1}\right)^{1/(k-1)} = (3.671 \text{ kg/m}^3)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)} = 2.327 \text{ kg/m}^3$$

$$T^* = \frac{2}{k+1} T_0 = \frac{2}{1.4+1}(298 \text{ K}) = 248.3 \text{ K}$$

$$V = c = \sqrt{kRT^*} = \sqrt{(1.4)(0.287 \text{ kJ/kg} \cdot \text{K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)(248.3 \text{ K})} = 315.9 \text{ m/s}$$

Then the initial mass flow rate through the hole becomes

$$\dot{m} = \rho A V = (2.327 \text{ kg/m}^3)[\pi (0.004 \text{ m})^2/4](315.9 \text{ m/s}) = 0.00924 \text{ kg/s} = \mathbf{0.554 \text{ kg/min}}$$

Discussion The mass flow rate will decrease with time as the pressure inside the tire drops.

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17-122 The thrust developed by the engine of a Boeing 777 is about 380 kN. The mass flow rate of air through the nozzle is to be determined.

Assumptions **1** Air is an ideal gas with constant specific properties. **2** Flow of combustion gases through the nozzle is isentropic. **3** Choked flow conditions exist at the nozzle exit. **4** The velocity of gases at the nozzle inlet is negligible.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1), and it can also be used for combustion gases. The specific heat ratio of combustion gases is $k = 1.33$ (Table 17-2).

Analysis The velocity at the nozzle exit is the sonic velocity, which is determined to be

$$V = c = \sqrt{kRT} = \sqrt{(1.33)(0.287\,\text{kJ/kg}\cdot\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)(265\,\text{K})} = 318.0\,\text{m/s}$$

Noting that thrust F is related to velocity by $F = \dot{m}V$, the mass flow rate of combustion gases is determined to be

$$\dot{m} = \frac{F}{V} = \frac{380{,}000\,\text{N}}{318.0\,\text{m/s}}\left(\frac{1\,\text{kg.m/s}^2}{1\,\text{N}}\right) = \mathbf{1194.8\,kg/s}$$

Discussion The combustion gases are mostly nitrogen (due to the 78% of N_2 in air), and thus they can be treated as air with a good degree of approximation.

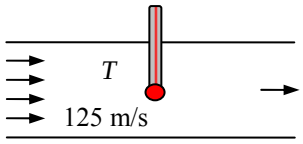
17-123 A stationary temperature probe is inserted into an air duct reads 50°C. The actual temperature of air is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The stagnation process is isentropic.

Properties The specific heat of air at room temperature is $c_p = 1.005$ kJ/kg·K (Table A-2a).

Analysis The air that strikes the probe will be brought to a complete stop, and thus it will undergo a stagnation process. The thermometer will sense the temperature of this stagnated air, which is the stagnation temperature. The actual air temperature is determined from

$$T = T_0 - \frac{V^2}{2c_p} = 50°\text{C} - \frac{(125\,\text{m/s})^2}{2\times 1.005\,\text{kJ/kg}\cdot\text{K}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{42.2°C}$$

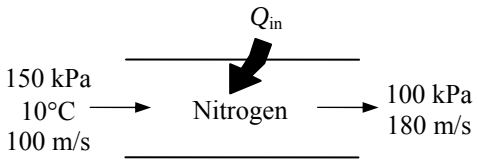


Discussion Temperature rise due to stagnation is very significant in high-speed flows, and should always be considered when compressibility effects are not negligible.

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17-124 Nitrogen flows through a heat exchanger. The stagnation pressure and temperature of the nitrogen at the inlet and the exit states are to be determined.

Assumptions **1** Nitrogen is an ideal gas with constant specific properties. **2** Flow of nitrogen through the heat exchanger is isentropic.



Properties The properties of nitrogen are $c_p = 1.039$ kJ/kg.K and $k = 1.4$ (Table A-2a).

Analysis The stagnation temperature and pressure of nitrogen at the inlet and the exit states are determined from

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 10°\text{C} + \frac{(100\ \text{m/s})^2}{2\times 1.039\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{14.8°C}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (150\ \text{kPa})\left(\frac{288.0\ \text{K}}{283.2\ \text{K}}\right)^{1.4/(1.4-1)} = \mathbf{159.1\ kPa}$$

From the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ with $w = 0$

$$q_{in} = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2} + \Delta\text{pe}^{\nearrow 0}$$

$$125\ \text{kJ/kg} = (1.039\ \text{kJ/kg}\cdot°\text{C})(T_2 - 10°\text{C}) + \frac{(180\ \text{m/s})^2 - (100\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$T_2 = 119.5°\text{C}$$

and

$$T_{02} = T_2 + \frac{V_2^2}{2c_p} = 119.5°\text{C} + \frac{(180\ \text{m/s})^2}{2\times 1.039\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{135.1°C}$$

$$P_{02} = P_2\left(\frac{T_{02}}{T_2}\right)^{k/(k-1)} = (100\ \text{kPa})\left(\frac{408.3\ \text{K}}{392.7\ \text{K}}\right)^{1.4/(1.4-1)} = \mathbf{114.6\ kPa}$$

Discussion Note that the stagnation temperature and pressure can be very different than their thermodynamic counterparts when dealing with compressible flow.

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17-125 An expression for the speed of sound based on van der Waals equation of state is to be derived. Using this relation, the speed of sound in carbon dioxide is to be determined and compared to that obtained by ideal gas behavior.

Properties The properties of CO_2 are $R = 0.1889$ kJ/kg·K and $k = 1.279$ at $T = 50°C = 323.2$ K (Table A-2b).

Analysis Van der Waals equation of state can be expressed as

$$P = \frac{RT}{v - b} - \frac{a}{v^2}$$

Differentiating,

$$\left(\frac{\partial P}{\partial v}\right)_T = -\frac{RT}{(v-b)^2} + \frac{2a}{v^3}$$

Noting that $\rho = 1/v \longrightarrow d\rho = -dv/v^2$, the speed of sound relation becomes

$$c^2 = k\left(\frac{\partial P}{\partial r}\right)_T = -v^2 k\left(\frac{\partial P}{\partial v}\right)_T$$

Substituting,

$$c^2 = \frac{v^2 kRT}{(v-b)^2} - \frac{2ak}{v}$$

Using the molar mass of CO_2 ($M = 44$ kg/kmol), the constant a and b can be expressed per unit mass as

$$a = 0.1882\ \text{kPa}\cdot\text{m}^6/\text{kg}^2 \quad \text{and} \qquad b = 9.705\times10^{-4}\ \text{m}^3/\text{kg}$$

The specific volume of CO_2 is determined to be

$$200\ \text{kPa} = \frac{(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323.2\ \text{K})}{v - 0.0009705\ \text{m}^3/\text{kg}} - \frac{2\times0.1882\ \text{kPa}\cdot\text{m}^6/\text{kg}^2}{v^2} \rightarrow v = 0.3031\ \text{m}^3/\text{kg}$$

Substituting,

$$c = \left(\frac{(0.3031\ \text{m}^3/\text{kg})^2 (1.279)(0.1889\ \text{kJ/kg}\cdot\text{K})(323.2\ \text{K})}{(0.3031 - 0.0009705\ \text{m}^3/\text{kg})^2} \frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}} - \frac{2(0.1882\ \text{kPa.m}^6/\text{kg}^3)(1.279)}{(0.3031\ \text{m}^3/\text{kg})^2} \frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kPa}\cdot\text{m}^3/\text{kg}} \right)^{1/2} = \mathbf{277.5\ m/s}$$

If we treat CO_2 as an ideal gas, the speed of sound becomes

$$c = \sqrt{kRT} = \sqrt{(1.279)(0.1889\ \text{kJ/kg}\cdot\text{K})(323.2\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{279.4\ m/s}$$

Discussion Note that the ideal gas relation is the simplest equation of state, and it is very accurate for most gases encountered in practice. At high pressures and/or low temperatures, however, the gases deviate from ideal gas behavior, and it becomes necessary to use more complicated equations of state.

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17-126 The equivalent relation for the speed of sound is to be verified using thermodynamic relations.

Analysis The two relations are $c^2 = \left(\frac{\partial P}{\partial \rho}\right)_s$ and $c^2 = k\left(\frac{\partial P}{\partial \rho}\right)_T$

From $r = 1/\nu \longrightarrow dr = -d\nu/\nu^2$. Thus,

$$c^2 = \left(\frac{\partial P}{\partial r}\right)_s = -\nu^2\left(\frac{\partial P}{\partial \nu}\right)_s = -\nu^2\left(\frac{\partial P}{\partial T}\frac{\partial T}{\partial \nu}\right)_s = -\nu^2\left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial \nu}\right)_s$$

From the cyclic rule,

$$(P,T,s): \left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial s}\right)_P\left(\frac{\partial s}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_s = -\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial P}{\partial s}\right)_T$$

$$(T,\nu,s): \left(\frac{\partial T}{\partial \nu}\right)_s\left(\frac{\partial \nu}{\partial s}\right)_T\left(\frac{\partial s}{\partial T}\right)_\nu = -1 \longrightarrow \left(\frac{\partial T}{\partial \nu}\right)_s = -\left(\frac{\partial s}{\partial \nu}\right)_T\left(\frac{\partial T}{\partial s}\right)_\nu$$

Substituting,

$$c^2 = -\nu^2\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial P}{\partial s}\right)_T\left(\frac{\partial s}{\partial \nu}\right)_T\left(\frac{\partial T}{\partial s}\right)_\nu = -\nu^2\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial T}{\partial s}\right)_\nu\left(\frac{\partial P}{\partial s}\right)_T$$

Recall that

$$\frac{c_p}{T} = \left(\frac{\partial s}{\partial T}\right)_P \qquad \text{and} \qquad \frac{c_\nu}{T} = \left(\frac{\partial s}{\partial T}\right)_\nu$$

Substituting,

$$c^2 = -\nu^2\left(\frac{c_p}{T}\right)\left(\frac{T}{c_\nu}\right)\left(\frac{\partial P}{\partial \nu}\right)_T = -\nu^2 k\left(\frac{\partial P}{\partial \nu}\right)_T$$

Replacing $-d\nu/\nu^2$ by $d\rho$,

$$c^2 = k\left(\frac{\partial P}{\partial \rho}\right)_T$$

Discussion Note that the differential thermodynamic property relations are very useful in the derivation of other property relations in differential form.

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17-127 For ideal gases undergoing isentropic flows, expressions for P/P^*, T/T^*, and ρ/ρ^* as functions of k and Ma are to be obtained.

Analysis Equations 17-18 and 17-21 are given to be

$$\frac{T_0}{T} = \frac{2+(k-1)\text{Ma}^2}{2}$$

and

$$\frac{T^*}{T_0} = \frac{2}{k+1}$$

Multiplying the two,

$$\left(\frac{T_0}{T}\frac{T^*}{T_0}\right) = \left(\frac{2+(k-1)\text{Ma}^2}{2}\right)\left(\frac{2}{k+1}\right)$$

Simplifying and inverting,

$$\frac{T}{T^*} = \frac{k+1}{2+(k-1)\text{Ma}^2} \tag{1}$$

From

$$\frac{P}{P^*} = \left(\frac{T}{T^*}\right)^{k/(k-1)} \longrightarrow \frac{P}{P^*} = \left(\frac{k+1}{2+(k-1)\text{Ma}^2}\right)^{k/(k-1)} \tag{2}$$

From

$$\frac{\rho}{\rho^*} = \left(\frac{\rho}{\rho^*}\right)^{k/(k-1)} \longrightarrow \frac{\rho}{\rho^*} = \left(\frac{k+1}{2+(k-1)\text{Ma}^2}\right)^{k/(k-1)} \tag{3}$$

Discussion Note that some very useful relations can be obtained by very simple manipulations.

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17-128 It is to be verified that for the steady flow of ideal gases $dT_0/T = dA/A + (1\text{-Ma}^2)\, dV/V$. The effect of heating and area changes on the velocity of an ideal gas in steady flow for subsonic flow and supersonic flow are to be explained.

Analysis We start with the relation $\dfrac{V^2}{2} = c_p(T_0 - T)$, (1)

Differentiating, $V\,dV = c_p(dT_0 - dT)$ (2)

We also have $\dfrac{d\rho}{\rho} + \dfrac{dA}{A} + \dfrac{dV}{V} = 0$ (3)

and $\dfrac{dP}{\rho} + V\,dV = 0$ (4)

Differentiating the ideal gas relation $P = \rho RT$, $\dfrac{dP}{P} = \dfrac{d\rho}{\rho} + \dfrac{dT}{T} = 0$ (5)

From the speed of sound relation, $c^2 = kRT = (k-1)c_pT = kP/\rho$ (6)

Combining Eqs. (3) and (5), $\dfrac{dP}{P} - \dfrac{dT}{T} + \dfrac{dA}{A} + \dfrac{dV}{V} = 0$ (7)

Combining Eqs. (4) and (6), $\dfrac{dP}{\rho} = \dfrac{dP}{kP/c^2} = -V\,dV$

or, $\dfrac{dP}{P} = -\dfrac{k}{c^2}V\,dV = -k\dfrac{V^2}{c^2}\dfrac{dV}{V} = -k\text{Ma}^2\dfrac{dV}{V}$ (8)

Combining Eqs. (2) and (6),

$$dT = dT_0 - V\frac{dV}{c_p}$$

or, $\dfrac{dT}{T} = \dfrac{dT_0}{T} - \dfrac{V^2}{c_pT}\dfrac{dV}{V} = \dfrac{dT}{T} = \dfrac{dT_0}{T} - \dfrac{V^2}{c^2/(k-1)}\dfrac{dV}{V} = \dfrac{dT_0}{T} - (k-1)\text{Ma}^2\dfrac{dV}{V}$ (9)

Combining Eqs. (7), (8), and (9),

$$-(k-1)\text{Ma}^2\frac{dV}{V} - \frac{dT_0}{T} + (k-1)\text{Ma}^2\frac{dV}{V} + \frac{dA}{A} + \frac{dV}{V} = 0$$

or, $\dfrac{dT_0}{T} = \dfrac{dA}{A} + \left[-k\text{Ma}^2 + (k-1)\text{Ma}^2 + 1\right]\dfrac{dV}{V}$

Thus, $\dfrac{dT_0}{T} = \dfrac{dA}{A} + (1 - \text{Ma}^2)\dfrac{dV}{V}$ (10)

Differentiating the steady-flow energy equation $q = h_{02} - h_{01} = c_p(T_{02} - T_{01})$

$$\delta q = c_p\,dT_0 \tag{11}$$

Eq. (11) relates the stagnation temperature change dT_0 to the net heat transferred to the fluid. Eq. (10) relates the velocity changes to area changes dA, and the stagnation temperature change dT_0 or the heat transferred.

(*a*) When Ma < 1 (subsonic flow), the fluid will accelerate if the duck converges ($dA < 0$) or the fluid is heated ($dT_0 > 0$ or $\delta q > 0$). The fluid will decelerate if the duck converges ($dA < 0$) or the fluid is cooled ($dT_0 < 0$ or $\delta q < 0$).

(*b*) When Ma > 1 (supersonic flow), the fluid will accelerate if the duck diverges ($dA > 0$) or the fluid is cooled ($dT_0 < 0$ or $\delta q < 0$). The fluid will decelerate if the duck converges ($dA < 0$) or the fluid is heated ($dT_0 > 0$ or $\delta q > 0$).

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17-129 A pitot tube measures the difference between the static and stagnation pressures for a subsonic airplane. The speed of the airplane and the flight Mach number are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heat ratio. **2** The stagnation process is isentropic.

Properties The properties of air are $R = 0.287$ kJ/kg.K and $k = 1.4$ (Table A-2a).

Analysis The stagnation pressure of air at the specified conditions is

$$P_0 = P + \Delta P = 30.8 + 20 = 50.8 \text{ kPa}$$

Then,

$$\frac{P_0}{P} = \left(1 + \frac{(k-1)\text{Ma}^2}{2}\right)^{k/k-1} \longrightarrow \frac{50.8}{30.8} = \left(1 + \frac{(1.4-1)\text{Ma}^2}{2}\right)^{1.4/0.4}$$

It yields

Ma = **0.877**

The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287 \text{ kJ/kg} \cdot \text{K})(240 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 310.5 \text{ m/s}$$

Thus, $V = \text{Ma} \times c = (0.877)(310.5 \text{ m/s}) =$ **272 m/s**

Discussion Note that the flow velocity can be measured in a simple and accurate way by simply measuring pressure.

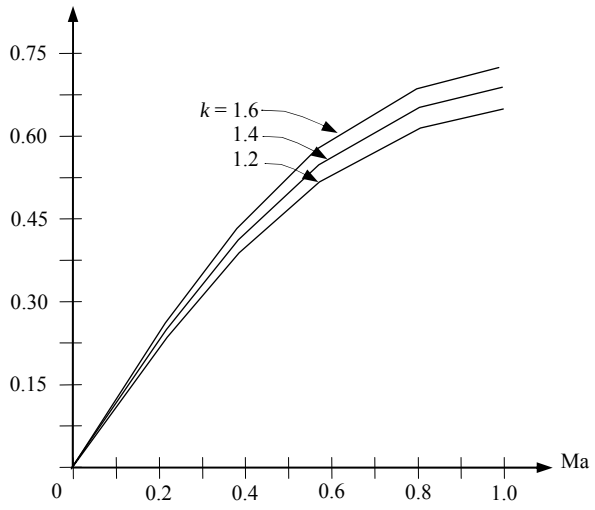
17-130 The mass flow parameter $\dot{m}\sqrt{RT_0} / (AP_0)$ versus the Mach number for $k = 1.2$, 1.4, and 1.6 in the range of $0 \le \text{Ma} \le 1$ is to be plotted.

Analysis The mass flow rate parameter $(\dot{m}\sqrt{RT_0}) / P_0 A$ can be expressed as

$$\frac{\dot{m}\sqrt{RT_0}}{P_0 A} = \text{Ma}\sqrt{k}\left(\frac{2}{2 + (k-1)M^2}\right)^{(k+1)/2(k-1)}$$

Thus,

Ma	***k* = 1.2**	***k* = 1.4**	***k* = 1.6**
0.0	0	0	0
0.1	0.1089	0.1176	0.1257
0.2	0.2143	0.2311	0.2465
0.3	0.3128	0.3365	0.3582
0.4	0.4015	0.4306	0.4571
0.5	0.4782	0.5111	0.5407
0.6	0.5411	0.5763	0.6077
0.7	0.5894	0.6257	0.6578
0.8	0.6230	0.6595	0.6916
0.9	0.6424	0.6787	0.7106
1.0	0.6485	0.6847	0.7164



Discussion Note that the mass flow rate increases with increasing Mach number and specific heat ratio. It levels off at Ma = 1, and remains constant (choked flow).

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17-131 Helium gas is accelerated in a nozzle. The pressure and temperature of helium at the location where Ma = 1 and the ratio of the flow area at this location to the inlet flow area are to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

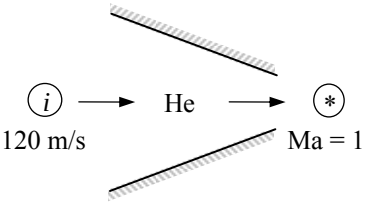
Properties The properties of helium are $R = 2.0769$ kJ/kg·K, $c_p = 5.1926$ kJ/kg·K, and $k = 1.667$ (Table A-2a).

Analysis The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T_i + \frac{V_i^2}{2c_p} = 500\text{ K} + \frac{(120\text{ m/s})^2}{2\times 5.1926\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 501.4\text{ K}$$

and

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (0.8\text{ MPa})\left(\frac{501.4\text{ K}}{500\text{ K}}\right)^{1.667/(1.667-1)} = 0.806\text{ MPa}$$



The Mach number at the nozzle exit is given to be Ma = 1. Therefore, the properties at the nozzle exit are the *critical properties* determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (501.4\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{376\text{ K}}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.806\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.393\text{ MPa}}$$

The speed of sound and the Mach number at the nozzle inlet are

$$c_i = \sqrt{kRT}_i = \sqrt{(1.667)(2.0769\text{ kJ/kg}\cdot\text{K})(500\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 1316\text{ m/s}$$

$$\text{Ma}_i = \frac{V_i}{c_i} = \frac{120\text{ m/s}}{1316\text{ m/s}} = 0.0912$$

The ratio of the entrance-to-throat area is

$$\frac{A_i}{A^*} = \frac{1}{\text{Ma}_i}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}_i^2\right)\right]^{(k+1)/[2(k-1)]}$$

$$= \frac{1}{0.0912}\left[\left(\frac{2}{1.667+1}\right)\left(1+\frac{1.667-1}{2}(0.0912)^2\right)\right]^{2.667/(2\times 0.667)}$$

$$= 6.20$$

Then the ratio of the throat area to the entrance area becomes

$$\frac{A^*}{A_i} = \frac{1}{6.20} = \mathbf{0.161}$$

Discussion The compressible flow functions are essential tools when determining the proper shape of the compressible flow duct.

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17-132 Helium gas enters a nozzle with negligible velocity, and is accelerated in a nozzle. The pressure and temperature of helium at the location where Ma = 1 and the ratio of the flow area at this location to the inlet flow area are to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The entrance velocity is negligible.

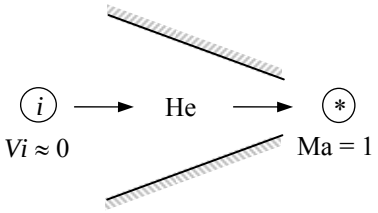
Properties The properties of helium are $R = 2.0769$ kJ/kg·K, $c_p = 5.1926$ kJ/kg·K, and $k = 1.667$ (Table A-2a).

Analysis We treat helium as an ideal gas with $k = 1.667$. The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *.

The stagnation temperature and pressure in this case are identical to the inlet temperature and pressure since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

$T_0 = T_i = 500$ K

$P_0 = P_i = 0.8$ MPa



The Mach number at the nozzle exit is given to be Ma = 1. Therefore, the properties at the nozzle exit are the *critical properties* determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (500\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{375\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.8\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.390\ MPa}$$

The ratio of the nozzle inlet area to the throat area is determined from

$$\frac{A_i}{A^*} = \frac{1}{\text{Ma}_i}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}_i^2\right)\right]^{(k+1)/[2(k-1)]}$$

But the Mach number at the nozzle inlet is Ma = 0 since $V_i \cong 0$. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A_i} = \frac{1}{\infty} = \mathbf{0}$$

Discussion The compressible flow functions are essential tools when determining the proper shape of the compressible flow duct.

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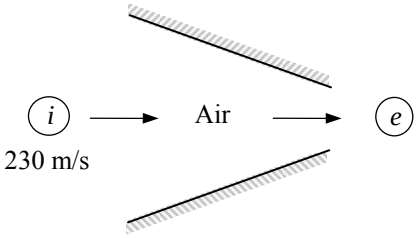
17-133 Air enters a converging nozzle. The mass flow rate, the exit velocity, the exit Mach number, and the exit pressure-stagnation pressure ratio versus the back pressure-stagnation pressure ratio for a specified back pressure range are to be calculated and plotted.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air at room temperature are $R = 0.287$ kJ/kg·K, $c_p = 1.005$ kJ/kg·K, and $k = 1.4$ (Table A-2a).

Analysis The stagnation properties remain constant throughout the nozzle since the flow is isentropic. They are determined from

$$T_0 = T_i + \frac{V_i^2}{2c_p} = 500\ \text{K} + \frac{(230\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 526.3\ \text{K}$$



and

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (900\ \text{kPa})\left(\frac{526.3\ \text{K}}{500\ \text{K}}\right)^{1.4/(1.4-1)} = 1077\ \text{kPa}$$

The critical pressure is determined to be

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1077\ \text{kPa})\left(\frac{2}{1.4+1}\right)^{1.4/0.4} = 569.0\ \text{kPa}$$

Then the pressure at the exit plane (throat) will be

$P_e = P_b$ for $P_b \ge 569.0$ kPa

$P_e = P^* = 569.0$ kPa for $P_b < 569.0$ kPa (choked flow)

Thus the back pressure will not affect the flow when $100 < P_b < 569.0$ kPa. For a specified exit pressure P_e, the temperature, the velocity and the mass flow rate can be determined from

Temperature $$T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (526.3\ \text{K})\left(\frac{P_e}{1077}\right)^{0.4/1.4}$$

Velocity $$V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.005\ \text{kJ/kg}\cdot\text{K})(526.3 - T_e)\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)}$$

Speed of sound $$c_e = \sqrt{kRT}_e = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)}$$

Mach number $$\text{Ma}_e = V_e / c_e$$

Density $$\rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_e}$$

Mass flow rate $$\dot{m} = \rho_e V_e A_e = \rho_e V_e (0.001\ \text{m}^2)$$

The EES solution and the results are given below

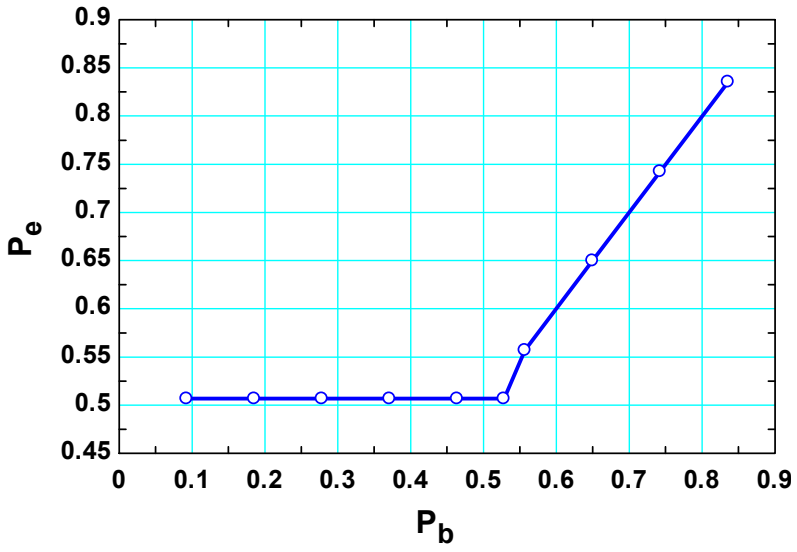
```
"Given"
P_i=900 "[kPa]"
T_i=500 "[K]"
Vel_i=230 "[m/s]"
A_e=10E-4 "[m^2]"
```

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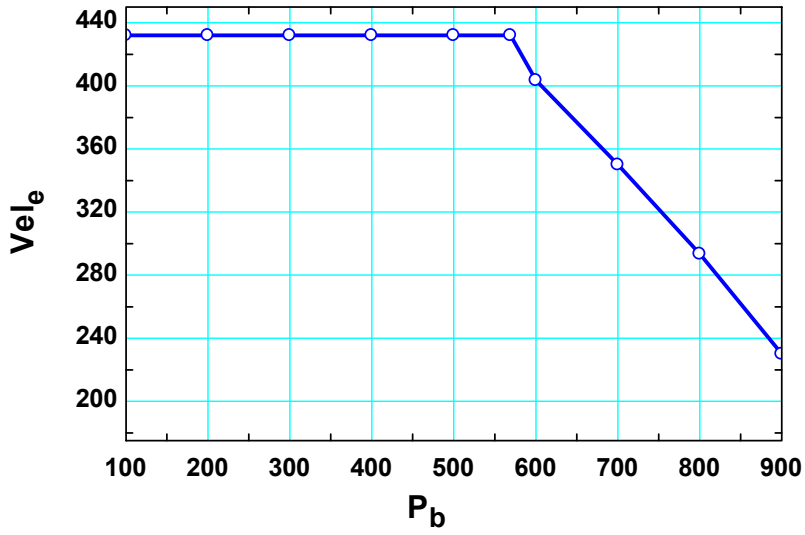
"Properties"
C_p=1.005
k=1.4
R=0.287

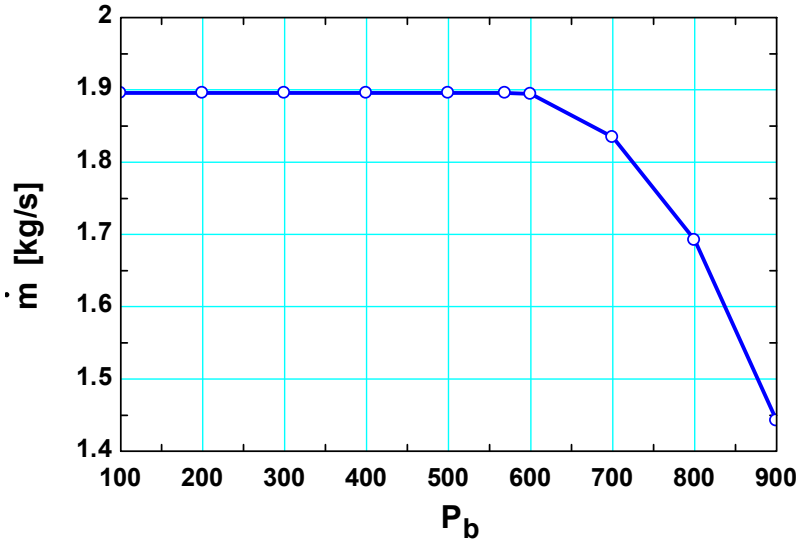
"Analysis"
T0=T_i+Vel_i^2/(2*C_p)*Convert(m^2/s^2, kJ/kg)
P0=P_i*(T0/T_i)^(k/(k-1))
P_star=P0*(2/(k+1))^(k/(k-1))
T_e=T0*(P_e/P0)^((k-1)/k)
Vel_e=sqrt(2*C_p*(T0-T_e)*Convert(kJ/kg, m^2/s^2))
C_e=sqrt(k*R*T_e*Convert(kJ/kg, m^2/s^2))
M_e=Vel_e/C_e
rho_e=P_e/(R*T_e)
m_dot=rho_e*Vel_e*A_e
RatioP_e=P_e/P0
RatioP_b=P_b/P0
"P_e=P_b for P_b >= P_star and P_e=P_star for P_b < P_star

P_b, kPa	RatioP$_b$	P_e, kPa	RatioP$_e$	T_e, K	V_e, m/s	M$_e$	ρ_e, kg/m^3	$\dot{m}$, kg / s
900	0.836	900	0.836	500	230	0.51	6.272	1.443
800	0.743	800	0.743	483.5	293.5	0.67	5.766	1.692
700	0.650	700	0.650	465.4	350	0.81	5.241	1.835
600	0.557	600	0.557	445.3	403.5	0.95	4.695	1.894
569	0.528	545.9	0.507	433.4	432.1	1.04	4.388	1.896
500	0.464	545.9	0.507	433.4	432.1	1.04	4.388	1.896
400	0.371	545.9	0.507	433.4	432.1	1.04	4.388	1.896
300	0.279	545.9	0.507	433.4	432.1	1.04	4.388	1.896
200	0.186	545.9	0.507	433.4	432.1	1.04	4.388	1.896
100	0.093	545.9	0.507	433.4	432.1	1.04	4.388	1.896



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17-134 Steam enters a converging nozzle. The exit pressure, the exit velocity, and the mass flow rate versus the back pressure for a specified back pressure range are to be plotted.

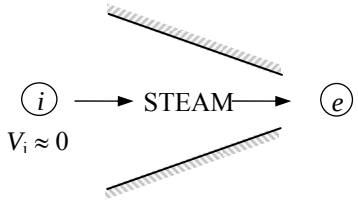
Assumptions **1** Steam is to be treated as an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The ideal gas properties of steam are given to be $R = 0.462$ kJ/kg.K, $c_p = 1.872$ kJ/kg.K, and $k = 1.3$.

Analysis The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Since the flow is isentropic, they remain constant throughout the nozzle,

$$P_0 = P_i = 6 \text{ MPa}$$

$$T_0 = T_i = 700 \text{ K}$$



The critical pressure is determined from to be

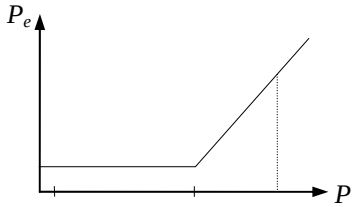
$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (6 \text{ MPa})\left(\frac{2}{1.3+1}\right)^{1.3/0.3} = 3.274 \text{ MPa}$$

Then the pressure at the exit plane (throat) will be

$P_e = P_b$ for $P_b \geq 3.274$ MPa

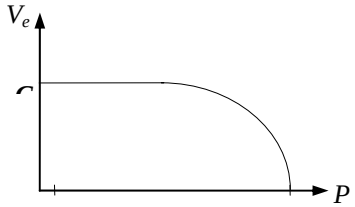
$P_e = P^* = 3.274$ MPa for $P_b < 3.274$ MPa (choked flow)

Thus the back pressure will not affect the flow when $3 < P_b < 3.274$ MPa. For a specified exit pressure P_e, the temperature, the velocity and the mass flow rate can be determined from



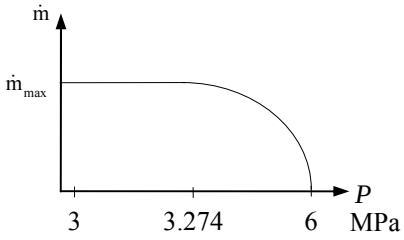
Temperature

$$T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (700 \text{ K})\left(\frac{P_e}{6}\right)^{0.3/1.3}$$



Velocity

$$V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.872 \text{ kJ/kg}\cdot\text{K})(700 - T_e)\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)}$$



Density

$$\rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.462 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_e}$$

Mass flow rate

$$\dot{m} = \rho_e V_e A_e = \rho_e V_e (0.0008 \text{ m}^2)$$

The results of the calculations can be tabulated as follows:

P_b, MPa	P_e, MPa	T_e, K	V_e, m/s	ρ_e, kg/m^3	$\dot{m}$, kg/s
6.0	6.0	700	0	18.55	0
5.5	5.5	686.1	228.1	17.35	3.166
5.0	5.0	671.2	328.4	16.12	4.235
4.5	4.5	655.0	410.5	14.87	4.883
4.0	4.0	637.5	483.7	13.58	5.255
3.5	3.5	618.1	553.7	12.26	5.431
3.274	3.274	608.7	584.7	11.64	5.445
3.0	3.274	608.7	584.7	11.64	5.445

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17-135 An expression for the ratio of the stagnation pressure after a shock wave to the static pressure before the shock wave as a function of k and the Mach number upstream of the shock wave is to be found.

Analysis The relation between P_1 and P_2 is

$$\frac{P_2}{P_1}=\frac{1+k\text{Ma}_2^2}{1+k\text{Ma}_1^2}\longrightarrow P_2=P_1\left(\frac{1+k\text{Ma}_1^2}{1+k\text{Ma}_2^2}\right)$$

Substituting this into the isentropic relation

$$\frac{P_{02}}{P_1}=\left(1+(k-1)\text{Ma}_2^2/2\right)^{k/(k-1)}$$

Then,

$$\frac{P_{02}}{P_1}=\left(\frac{1+k\text{Ma}_1^2}{1+k\text{Ma}_2^2}\right)\left(1+(k-1)\text{Ma}_2^2/2\right)^{k/(k-1)}$$

where

$$\text{Ma}_2^2=\frac{\text{Ma}_1^2+2/(k-1)}{2k\text{Ma}_2^2/(k-1)-1}$$

Substituting,

$$\frac{P_{02}}{P_1}=\left(\frac{(1+k\text{Ma}_1^2)(2k\text{Ma}_1^2-k+1)}{k\text{Ma}_1^2(k+1)-k+3}\right)\left(1+\frac{(k-1)\text{Ma}_1^2/2+1}{2k\text{Ma}_1^2/(k-1)-1}\right)^{k/(k-1)}$$

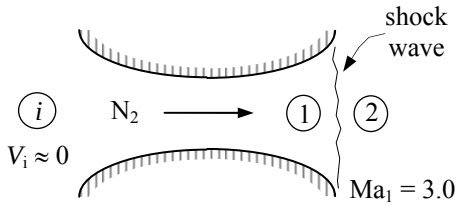
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17-136 Nitrogen entering a converging-diverging nozzle experiences a normal shock. The pressure, temperature, velocity, Mach number, and stagnation pressure downstream of the shock are to be determined. The results are to be compared to those of air under the same conditions.

Assumptions **1** Nitrogen is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The properties of nitrogen are $R = 0.2968$ kJ/kg·K and $k = 1.4$ (Table A-2a).

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Assuming the flow before the shock to be isentropic,



$$P_{01} = P_i = 700\ \text{kPa}$$
$$T_{01} = T_i = 300\ \text{K}$$

Then,

$$T_1 = T_{01}\left(\frac{2}{2+(k-1)\text{Ma}_1^2}\right) = (300\ \text{K})\left(\frac{2}{2+(1.4-1)3^2}\right) = 107.1\ \text{K}$$

and

$$P_1 = P_{01}\left(\frac{T_1}{T_{01}}\right)^{k/(k-1)} = (700\ \text{kPa})\left(\frac{107.1}{300}\right)^{1.4/0.4} = 19.06\ \text{kPa}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-33. For $\text{Ma}_1 = 3.0$ we read

$$\text{Ma}_2 = \mathbf{0.4752},\ \frac{P_{02}}{P_{01}} = 0.32834,\ \frac{P_2}{P_1} = 10.333,\ \text{and}\ \frac{T_2}{T_1} = 2.679$$

Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

$$P_{02} = 0.32834P_{01} = (0.32834)(700\ \text{kPa}) = \mathbf{230\ kPa}$$
$$P_2 = 10.333P_1 = (10.333)(19.06\ \text{kPa}) = \mathbf{197\ kPa}$$
$$T_2 = 2.679T_1 = (2.679)(107.1\ \text{K}) = \mathbf{287\ K}$$

The velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.4752)\sqrt{(1.4)(0.2968\ \text{kJ/kg}\cdot\text{K})(287\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{164\ m/s}$$

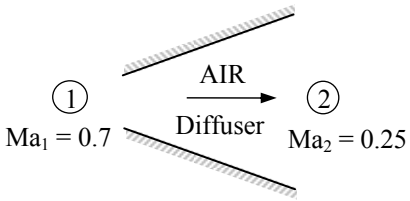
Discussion For **air** at specified conditions $k = 1.4$ (same as nitrogen) and $R = 0.287$ kJ/kg·K. Thus the only quantity which will be different in the case of air is the velocity after the normal shock, which happens to be 161.3 m/s.

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17-137 The diffuser of an aircraft is considered. The static pressure rise across the diffuser and the exit area are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the diffuser is steady, one-dimensional, and isentropic. **3** The diffuser is adiabatic.

Properties Air properties at room temperature are $R = 0.287$ kJ/kg·K, $c_p = 1.005$ kJ/kg·K, and $k = 1.4$ (Table A-2a).



Analysis The inlet velocity is

$$V_1 = \text{Ma}_1 c_1 = M_1\sqrt{kRT_1} = (0.7)\sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(242.7\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 218.6\text{ m/s}$$

Then the stagnation temperature and pressure at the diffuser inlet become

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 242.7 + \frac{(218.6\text{ m/s})^2}{2(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 266.5\text{ K}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (41.1\text{ kPa})\left(\frac{266.5\text{ K}}{242.7\text{ K}}\right)^{1.4/(1.4-1)} = 57.0\text{ kPa}$$

For an adiabatic diffuser, the energy equation reduces to $h_{01} = h_{02}$. Noting that $h = c_p T$ and the specific heats are assumed to be constant, we have

$$T_{01} = T_{02} = T_0 = 266.5\text{ K}$$

The isentropic relation between states 1 and 02 gives

$$P_{02} = P_1\left(\frac{T_{02}}{T_1}\right)^{k/(k-1)} = (41.1\text{ kPa})\left(\frac{266.5\text{ K}}{242.7\text{ K}}\right)^{1.4/(1.4-1)} = 55.59\text{ kPa}$$

The exit velocity can be expressed as

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.25)\sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})\,T_2\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 5.01\sqrt{T_2}$$

Thus, $$T_2 = T_{02} - \frac{V_2^2}{2c_p} = (266.5) - \frac{5.01^2 T_2\text{ m}^2/\text{s}^2}{2(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \longrightarrow T_2 = 263.2\text{ K}$$

Then the static exit pressure becomes

$$P_2 = P_{02}\left(\frac{T_2}{T_{02}}\right)^{k/(k-1)} = (55.59\text{ kPa})\left(\frac{263.2\text{ K}}{266.5\text{ K}}\right)^{1.4/(1.4-1)} = 53.23\text{ kPa}$$

Thus the static pressure rise across the diffuser is

$$\Delta P = P_2 - P_1 = 53.23 - 41.1 = \mathbf{12.13\ kPa}$$

Also, $$\rho_2 = \frac{P_2}{RT_2} = \frac{53.23\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(263.2\text{ K})} = 0.7047\text{ kg/m}^3$$

$$V_2 = 6.01\sqrt{T_2} = 5.01\sqrt{263.2} = 81.3\text{ m/s}$$

Thus, $$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{30\text{ kg/s}}{(0.7047\text{ kg/m}^3)(81.3\text{ m/s})} = \mathbf{0.524\ m^2}$$

Discussion The pressure rise in actual diffusers will be lower because of the irreversibilities. However, flow through well-designed diffusers is very nearly isentropic.

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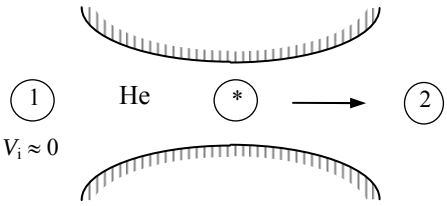
17-138 Helium gas is accelerated in a nozzle isentropically. For a specified mass flow rate, the throat and exit areas of the nozzle are to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The properties of helium are $R = 2.0769$ kJ/kg.K, $c_p = 5.1926$ kJ/kg.K, $k = 1.667$ (Table A-2a).

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible,

$$T_{01} = T_1 = 500 \text{ K}$$
$$P_{01} = P_1 = 1.0 \text{ MPa}$$



The flow is assumed to be isentropic, thus the stagnation temperature and pressure remain constant throughout the nozzle,

$$T_{02} = T_{01} = 500 \text{ K}$$
$$P_{02} = P_{01} = 1.0 \text{ MPa}$$

The critical pressure and temperature are determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (500 \text{ K})\left(\frac{2}{1.667+1}\right) = 375.0 \text{ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1.0 \text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = 0.487 \text{ MPa}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{487 \text{ kPa}}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(375 \text{ K})} = 0.625 \text{ kg/m}^3$$

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.667)(2.0769 \text{ kJ/kg}\cdot\text{K})(375 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 1139.4 \text{ m/s}$$

Thus the throat area is

$$A^* = \frac{\dot{m}}{\rho^* V^*} = \frac{0.25 \text{ kg/s}}{(0.625 \text{ kg/m}^3)(1139.4 \text{ m/s})} = 3.51\times10^{-4} \text{ m}^2 = \mathbf{3.51 \text{ cm}^2}$$

At the nozzle exit the pressure is $P_2 = 0.1$ MPa. Then the other properties at the nozzle exit are determined to be

$$\frac{P_0}{P_2} = \left(1+\frac{k-1}{2}\text{Ma}_2^2\right)^{k/(k-1)} \longrightarrow \frac{1.0 \text{ MPa}}{0.1 \text{ MPa}} = \left(1+\frac{1.667-1}{2}\text{Ma}_2^2\right)^{1.667/0.667}$$

It yields $\text{Ma}_2 = 2.130$, which is greater than 1. Therefore, the nozzle must be converging-diverging.

$$T_2 = T_0\left(\frac{2}{2+(k-1)\text{Ma}_2^2}\right) = (500 \text{ K})\left(\frac{2}{2+(1.667-1)\times 2.13^2}\right) = 199.0 \text{ K}$$

$$\rho_2 = \frac{P_2}{RT_2} = \frac{100 \text{ kPa}}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(199 \text{ K})} = 0.242 \text{ kg/m}^3$$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (2.13)\sqrt{(1.667)(2.0769 \text{ kJ/kg}\cdot\text{K})(199 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 1768.0 \text{ m/s}$$

Thus the exit area is

$$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.25 \text{ kg/s}}{(0.242 \text{ kg/m}^3)(1768 \text{ m/s})} = 5.84\times10^{-4} \text{ m}^2 = \mathbf{5.84 \text{ cm}^2}$$

Discussion Flow areas in actual nozzles would be somewhat larger to accommodate the irreversibilities.

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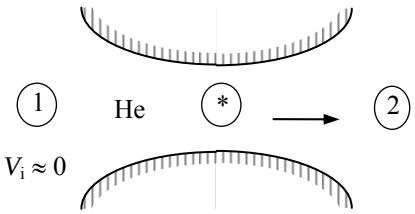
17-139E Helium gas is accelerated in a nozzle. For a specified mass flow rate, the throat and exit areas of the nozzle are to be determined for the cases of isentropic and 97% efficient nozzles.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The properties of helium are $R = 0.4961$ Btu/lbm·R = 2.6809 psia·ft^3/lbm·R, $c_p = 1.25$ Btu/lbm·R, and $k = 1.667$ (Table A-2Ea).

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible,

$$T_{01} = T_1 = 900 \text{ R}$$
$$P_{01} = P_1 = 150 \text{ psia}$$



The flow is assumed to be isentropic, thus the stagnation temperature and pressure remain constant throughout the nozzle,

$$T_{02} = T_{01} = 900 \text{ R}$$
$$P_{02} = P_{01} = 150 \text{ psia}$$

The critical pressure and temperature are determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (900 \text{ R})\left(\frac{2}{1.667+1}\right) = 674.9 \text{ R}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (150 \text{ psia})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = 73.1 \text{ psia}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{73.1 \text{ psia}}{(2.6809 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(674.9 \text{ R})} = 0.04041 \text{ lbm/ft}^3$$

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(674.9 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = 3738 \text{ ft/s}$$

and $$A^* = \frac{\dot{m}}{\rho^* V^*} = \frac{0.2 \text{ lbm/s}}{(0.04041 \text{ lbm/ft}^3)(3738 \text{ ft/s})} = \mathbf{0.00132 \text{ ft}^2}$$

At the nozzle exit the pressure is $P_2 = 15$ psia. Then the other properties at the nozzle exit are determined to be

$$\frac{p_0}{p_2} = \left(1 + \frac{k-1}{2}\text{Ma}_2^2\right)^{k/(k-1)} \longrightarrow \frac{150 \text{ psia}}{15 \text{ psia}} = \left(1 + \frac{1.667-1}{2}\text{Ma}_2^2\right)^{1.667/0.667}$$

It yields $\text{Ma}_2 = 2.130$, which is greater than 1. Therefore, the nozzle must be converging-diverging.

$$T_2 = T_0\left(\frac{2}{2+(k-1)\text{Ma}_2^2}\right) = (900 \text{ R})\left(\frac{2}{2+(1.667-1)\times 2.13^2}\right) = 358.1 \text{ R}$$

$$\rho_2 = \frac{P_2}{RT_2} = \frac{15 \text{ psia}}{(2.6809 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(358.1 \text{ R})} = 0.01561 \text{ lbm/ft}^3$$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (2.13)\sqrt{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(358.1 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = 5800 \text{ ft/s}$$

Thus the exit area is

$$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.2 \text{ lbm/s}}{(0.0156 \text{ lbm/ft}^3)(5800 \text{ ft/s})} = \mathbf{0.00221 \text{ ft}^2}$$

Discussion Flow areas in actual nozzles would be somewhat larger to accommodate the irreversibilities.

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17-140 Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-32 for an ideal gas with $k = 1.667$.

Properties The specific heat ratio of the ideal gas is given to be $k = 1.667$.

Analysis The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}}$$

$$\frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)}$$

$$\frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

k=1.667
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M

Ma	Ma^*	A/A^*	P/P_0	ρ/ρ_0	T/T_0
0.0	0	∞	1.0000	1.0000	1.0000
0.1	0.1153	5.6624	0.9917	0.9950	0.9967
0.2	0.2294	2.8879	0.9674	0.9803	0.9868
0.3	0.3413	1.9891	0.9288	0.9566	0.9709
0.4	0.4501	1.5602	0.8782	0.9250	0.9493
0.5	0.5547	1.3203	0.8186	0.8869	0.9230
0.6	0.6547	1.1760	0.7532	0.8437	0.8928
0.7	0.7494	1.0875	0.6850	0.7970	0.8595
0.8	0.8386	1.0351	0.6166	0.7482	0.8241
0.9	0.9222	1.0081	0.5501	0.6987	0.7873
1.0	1.0000	1.0000	0.4871	0.6495	0.7499
1.2	1.1390	1.0267	0.3752	0.5554	0.6756
1.4	1.2572	1.0983	0.2845	0.4704	0.6047
1.6	1.3570	1.2075	0.2138	0.3964	0.5394
1.8	1.4411	1.3519	0.1603	0.3334	0.4806
2.0	1.5117	1.5311	0.1202	0.2806	0.4284
2.2	1.5713	1.7459	0.0906	0.2368	0.3825
2.4	1.6216	1.9980	0.0686	0.2005	0.3424
2.6	1.6643	2.2893	0.0524	0.1705	0.3073
2.8	1.7007	2.6222	0.0403	0.1457	0.2767
3.0	1.7318	2.9990	0.0313	0.1251	0.2499
5.0	1.8895	9.7920	0.0038	0.0351	0.1071
∞	1.9996	∞	0	0	0

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17-141 Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-33 for an ideal gas with $k = 1.667$.

Properties The specific heat ratio of the ideal gas is given to be $k = 1.667$.

Analysis The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2 / P_1}{T_2 / T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

k=1.667
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)

Ma_1	Ma_2	P_2/P_1	ρ_2/ρ_1	T_2/T_1	P_{02}/P_{01}	P_{02}/P_1
1.0	1.0000	1.0000	1.0000	1.0000	1	2.0530
1.1	0.9131	1.2625	1.1496	1.0982	0.999	2.3308
1.2	0.8462	1.5500	1.2972	1.1949	0.9933	2.6473
1.3	0.7934	1.8626	1.4413	1.2923	0.9813	2.9990
1.4	0.7508	2.2001	1.5805	1.3920	0.9626	3.3838
1.5	0.7157	2.5626	1.7141	1.4950	0.938	3.8007
1.6	0.6864	2.9501	1.8415	1.6020	0.9085	4.2488
1.7	0.6618	3.3627	1.9624	1.7135	0.8752	4.7278
1.8	0.6407	3.8002	2.0766	1.8300	0.8392	5.2371
1.9	0.6227	4.2627	2.1842	1.9516	0.8016	5.7767
2.0	0.6070	4.7503	2.2853	2.0786	0.763	6.3462
2.1	0.5933	5.2628	2.3802	2.2111	0.7243	6.9457
2.2	0.5814	5.8004	2.4689	2.3493	0.6861	7.5749
2.3	0.5708	6.3629	2.5520	2.4933	0.6486	8.2339
2.4	0.5614	6.9504	2.6296	2.6432	0.6124	8.9225
2.5	0.5530	7.5630	2.7021	2.7989	0.5775	9.6407
2.6	0.5455	8.2005	2.7699	2.9606	0.5442	10.3885
2.7	0.5388	8.8631	2.8332	3.1283	0.5125	11.1659
2.8	0.5327	9.5506	2.8923	3.3021	0.4824	11.9728
2.9	0.5273	10.2632	2.9476	3.4819	0.4541	12.8091
3.0	0.5223	11.0007	2.9993	3.6678	0.4274	13.6750
4.0	0.4905	19.7514	3.3674	5.8654	0.2374	23.9530
5.0	0.4753	31.0022	3.5703	8.6834	0.1398	37.1723
∞	0.4473	∞	3.9985	∞	0	∞

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17-142 The critical temperature, pressure, and density of an equimolar mixture of oxygen and nitrogen for specified stagnation properties are to be determined.

Assumptions Both oxygen and nitrogen are ideal gases with constant specific heats at room temperature.

Properties The specific heat ratio and molar mass are $k = 1.395$ and $M = 32$ kg/kmol for oxygen, and $k = 1.4$ and $M = 28$ kg/kmol for nitrogen (Tables A-1 and A-2).

Analysis The gas constant of the mixture is

$$M_m = y_{O_2} M_{O_2} + y_{N_2} M_{N_2} = 0.5 \times 32 + 0.5 \times 28 = 30 \text{ kg/kmol}$$

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{30 \text{ kg/kmol}} = 0.2771 \text{ kJ/kg} \cdot \text{K}$$

The specific heat ratio is 1.4 for nitrogen, and nearly 1.4 for oxygen. Therefore, the specific heat ratio of the mixture is also 1.4. Then the critical temperature, pressure, and density of the mixture become

$$T^* = T_0\left(\frac{2}{k+1}\right) = (600 \text{ K})\left(\frac{2}{1.4+1}\right) = \mathbf{500.0\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (300 \text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/(1.4-1)} = \mathbf{158.5\ kPa}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{158.5 \text{ kPa}}{(0.2771 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(500 \text{ K})} = \mathbf{1.144\ kg/m^3}$$

Discussion If the specific heat ratios k of the two gases were different, then we would need to determine the k of the mixture from $k = c_{p,m}/c_{v,m}$ where the specific heats of the mixture are determined from

$$c_{p,m} = \text{mf}_{O_2} c_{p,O_2} + \text{mf}_{N_2} c_{p,N_2} = (y_{O_2} M_{O_2} / M_m) c_{p,O_2} + (y_{N_2} M_{N_2} / M_m) c_{p,N_2}$$

$$c_{v,m} = \text{mf}_{O_2} c_{v,O_2} + \text{mf}_{N_2} c_{v,N_2} = (y_{O_2} M_{O_2} / M_m) c_{v,O_2} + (y_{N_2} M_{N_2} / M_m) c_{v,N_2}$$

where mf is the mass fraction and y is the mole fraction. In this case it would give

$$c_{p,m} = (0.5 \times 32/30) \times 0.918 + (0.5 \times 28/30) \times 1.039 = 0.9745 \text{ kJ/kg.K}$$

$$c_{v,m} = (0.5 \times 32/30) \times 0.658 + (0.5 \times 28/30) \times 0.743 = 0.6977 \text{ kJ/kg.K}$$

and

$$k = 0.9745/0.6977 = 1.397$$

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17-143 Using EES (or other) software, the shape of a converging-diverging nozzle is to be determined for specified flow rate and stagnation conditions. The nozzle and the Mach number are to be plotted.

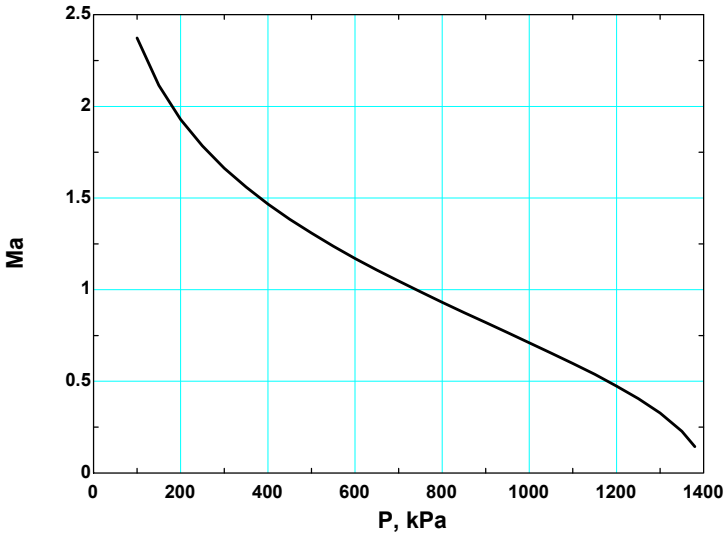
Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

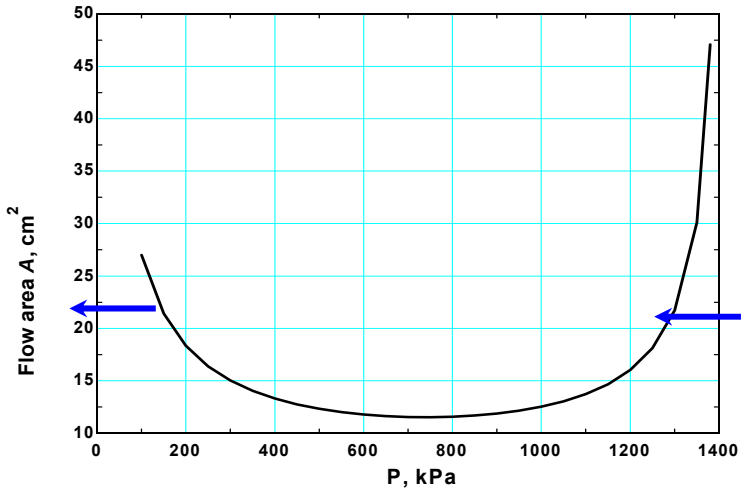
Properties The specific heat ratio of air at room temperature is 1.4 (Table A-2a).

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
P0=1400 "kPa"
T0=200+273 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
T=T0*(P/P0)^((k-1)/k)
V=SQRT(2*Cp*(T0-T)*1000)
A=m/(rho*V)*10000 "cm2"
C=SQRT(k*R*T*1000)
Ma=V/C
```

Pressure P, kPa	Flow area A, cm^2	Mach number Ma
1400	∞	0
1350	30.1	0.229
1300	21.7	0.327
1250	18.1	0.406
1200	16.0	0.475
1150	14.7	0.538
1100	13.7	0.597
1050	13.0	0.655
1000	12.5	0.710
950	12.2	0.766
900	11.9	0.820
850	11.7	0.876
800	11.6	0.931
750	11.5	0.988
700	11.5	1.047
650	11.6	1.107
600	11.8	1.171
550	12.0	1.237
500	12.3	1.308
450	12.8	1.384
400	13.3	1.467
350	14.0	1.559
300	15.0	1.663
250	16.4	1.784
200	18.3	1.929
150	21.4	2.114
100	27.0	2.373





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17-144 Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-32 for air.

Properties The specific heat ratio is given to be $k = 1.4$ for air

Analysis The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}}$$

$$\frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)}$$

$$\frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

Air:

```
k=1.4
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

Ma	Ma^*	A/A^*	P/P_0	ρ/ρ_0	T/T_0
1.0	1.0000	1.0000	0.5283	0.6339	0.8333
1.5	1.3646	1.1762	0.2724	0.3950	0.6897
2.0	1.6330	1.6875	0.1278	0.2300	0.5556
2.5	1.8257	2.6367	0.0585	0.1317	0.4444
3.0	1.9640	4.2346	0.0272	0.0762	0.3571
3.5	2.0642	6.7896	0.0131	0.0452	0.2899
4.0	2.1381	10.7188	0.0066	0.0277	0.2381
4.5	2.1936	16.5622	0.0035	0.0174	0.1980
5.0	2.2361	25.0000	0.0019	0.0113	0.1667
5.5	2.2691	36.8690	0.0011	0.0076	0.1418
6.0	2.2953	53.1798	0.0006	0.0052	0.1220
6.5	2.3163	75.1343	0.0004	0.0036	0.1058
7.0	2.3333	104.1429	0.0002	0.0026	0.0926
7.5	2.3474	141.8415	0.0002	0.0019	0.0816
8.0	2.3591	190.1094	0.0001	0.0014	0.0725
8.5	2.3689	251.0862	0.0001	0.0011	0.0647
9.0	2.3772	327.1893	0.0000	0.0008	0.0581
9.5	2.3843	421.1314	0.0000	0.0006	0.0525
10.0	2.3905	535.9375	0.0000	0.0005	0.0476

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17-145 Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-32 for methane.

Properties The specific heat ratio is given to be $k = 1.3$ for methane.

Analysis The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}}$$

$$\frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)}$$

$$\frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

Methane:

```
k=1.3
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

Ma	Ma^*	A/A^*	P/P_0	ρ/ρ_0	T/T_0
1.0	1.0000	1.0000	0.5457	0.6276	0.8696
1.5	1.3909	1.1895	0.2836	0.3793	0.7477
2.0	1.6956	1.7732	0.1305	0.2087	0.6250
2.5	1.9261	2.9545	0.0569	0.1103	0.5161
3.0	2.0986	5.1598	0.0247	0.0580	0.4255
3.5	2.2282	9.1098	0.0109	0.0309	0.3524
4.0	2.3263	15.9441	0.0050	0.0169	0.2941
4.5	2.4016	27.3870	0.0024	0.0095	0.2477
5.0	2.4602	45.9565	0.0012	0.0056	0.2105
5.5	2.5064	75.2197	0.0006	0.0033	0.1806
6.0	2.5434	120.0965	0.0003	0.0021	0.1563
6.5	2.5733	187.2173	0.0002	0.0013	0.1363
7.0	2.5978	285.3372	0.0001	0.0008	0.1198
7.5	2.6181	425.8095	0.0001	0.0006	0.1060
8.0	2.6350	623.1235	0.0000	0.0004	0.0943
8.5	2.6493	895.5077	0.0000	0.0003	0.0845
9.0	2.6615	1265.6040	0.0000	0.0002	0.0760
9.5	2.6719	1761.2133	0.0000	0.0001	0.0688
10.0	2.6810	2416.1184	0.0000	0.0001	0.0625

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17-146 Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-33 for air.

Properties The specific heat ratio is given to be $k = 1.4$ for air.

Analysis The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2 / P_1}{T_2 / T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

Air:

```
k=1.4
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)
```

Ma_1	Ma_2	P_2/P_1	ρ_2/ρ_1	T_2/T_1	P_{02}/P_{01}	P_{02}/P_1
1.0	1.0000	1.0000	1.0000	1.0000	1	1.8929
1.5	0.7011	2.4583	1.8621	1.3202	0.9298	3.4133
2.0	0.5774	4.5000	2.6667	1.6875	0.7209	5.6404
2.5	0.5130	7.1250	3.3333	2.1375	0.499	8.5261
3.0	0.4752	10.3333	3.8571	2.6790	0.3283	12.0610
3.5	0.4512	14.1250	4.2609	3.3151	0.2129	16.2420
4.0	0.4350	18.5000	4.5714	4.0469	0.1388	21.0681
4.5	0.4236	23.4583	4.8119	4.8751	0.0917	26.5387
5.0	0.4152	29.0000	5.0000	5.8000	0.06172	32.6535
5.5	0.4090	35.1250	5.1489	6.8218	0.04236	39.4124
6.0	0.4042	41.8333	5.2683	7.9406	0.02965	46.8152
6.5	0.4004	49.1250	5.3651	9.1564	0.02115	54.8620
7.0	0.3974	57.0000	5.4444	10.4694	0.01535	63.5526
7.5	0.3949	65.4583	5.5102	11.8795	0.01133	72.8871
8.0	0.3929	74.5000	5.5652	13.3867	0.008488	82.8655
8.5	0.3912	84.1250	5.6117	14.9911	0.006449	93.4876
9.0	0.3898	94.3333	5.6512	16.6927	0.004964	104.7536
9.5	0.3886	105.1250	5.6850	18.4915	0.003866	116.6634
10.0	0.3876	116.5000	5.7143	20.3875	0.003045	129.2170

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17-147 Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-33 for methane.

Properties The specific heat ratio is given to be $k = 1.3$ for methane.

Analysis The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2 / P_1}{T_2 / T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

Methane:

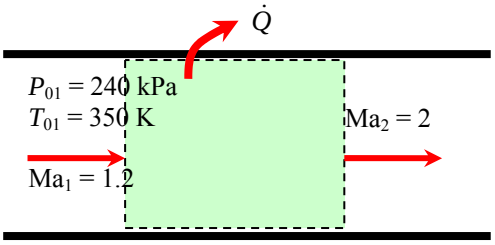
k=1.3
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)

Ma_1	Ma_2	P_2/P_1	ρ_2/ρ_1	T_2/T_1	P_{02}/P_{01}	P_{02}/P_1
1.0	1.0000	1.0000	1.0000	1.0000	1	1.8324
1.5	0.6942	2.4130	1.9346	1.2473	0.9261	3.2654
2.0	0.5629	4.3913	2.8750	1.5274	0.7006	5.3700
2.5	0.4929	6.9348	3.7097	1.8694	0.461	8.0983
3.0	0.4511	10.0435	4.4043	2.2804	0.2822	11.4409
3.5	0.4241	13.7174	4.9648	2.7630	0.1677	15.3948
4.0	0.4058	17.9565	5.4118	3.3181	0.09933	19.9589
4.5	0.3927	22.7609	5.7678	3.9462	0.05939	25.1325
5.0	0.3832	28.1304	6.0526	4.6476	0.03613	30.9155
5.5	0.3760	34.0652	6.2822	5.4225	0.02243	37.3076
6.0	0.3704	40.5652	6.4688	6.2710	0.01422	44.3087
6.5	0.3660	47.6304	6.6218	7.1930	0.009218	51.9188
7.0	0.3625	55.2609	6.7485	8.1886	0.006098	60.1379
7.5	0.3596	63.4565	6.8543	9.2579	0.004114	68.9658
8.0	0.3573	72.2174	6.9434	10.4009	0.002827	78.4027
8.5	0.3553	81.5435	7.0190	11.6175	0.001977	88.4485
9.0	0.3536	91.4348	7.0837	12.9079	0.001404	99.1032
9.5	0.3522	101.8913	7.1393	14.2719	0.001012	110.367
10.0	0.3510	112.9130	7.1875	15.7096	0.000740	122.239

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17-148 Air flowing at a supersonic velocity in a duct is accelerated by cooling. For a specified exit Mach number, the rate of heat transfer is to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.



Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K (Table A-2a).

Analysis Knowing stagnation properties, the static properties are determined to be

$$T_1 = T_{01}\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)^{-1} = (350\text{ K})\left(1+\frac{1.4-1}{2}1.2^2\right)^{-1} = 271.7\text{ K}$$

$$P_1 = P_{01}\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)^{-k/(k-1)} = (240\text{ kPa})\left(1+\frac{1.4-1}{2}1.2^2\right)^{-1.4/0.4} = 98.97\text{ kPa}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{98.97\text{ kPa}}{(0.287\text{ kJ/kgK})(271.7\text{ K})} = 1.269\text{ kg/m}^3$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(271.7\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 330.4\text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 1.2(330.4\text{ m/s}) = 396.5\text{ m/s}$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1 = (1.269\text{ kg/m}^3)[\pi(0.20\text{ m})^2/4](330.4\text{ m/s}) = 15.81\text{ kg/s}$$

The Rayleigh flow functions T_0/T_0^* corresponding to the inlet and exit Mach numbers are (Table A-34):

Ma$_1$ = 1.8: $T_{01}/T_0^* = 0.9787$

Ma$_2$ = 2: $T_{02}/T_0^* = 0.7934$

Then the exit stagnation temperature is determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{0.7934}{0.9787} = 0.8107 \quad \rightarrow \quad T_{02} = 0.8107T_{01} = 0.8107(350\text{ K}) = 283.7\text{ K}$$

Finally, the rate of heat transfer is

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (15.81\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(283.7-350)\text{ K} = \mathbf{-1053\text{ kW}}$$

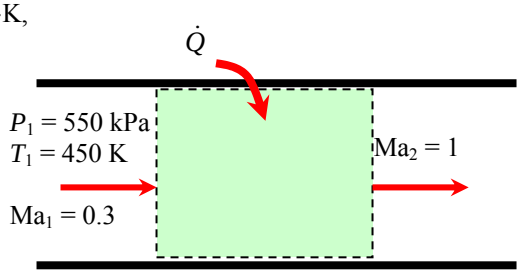
Discussion The negative sign confirms that the gas needs to be cooled in order to be accelerated. Also, it can be shown that the thermodynamic temperature drops to 158 K at the exit, which is extremely low. Therefore, the duct may need to be heavily insulated to maintain indicated flow conditions.

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17-149 Air flowing at a subsonic velocity in a duct is accelerated by heating. The highest rate of heat transfer without affecting the inlet conditions is to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K (Table A-2a).



Analysis Heat transfer will stop when the flow is choked, and thus $\mathrm{Ma}_2 = V_2/c_2 = 1$. The inlet density and stagnation temperature are

$$\rho_1 = \frac{P_1}{RT_1} = \frac{550\ \text{kPa}}{(0.287\ \text{kJ/kgK})(450\ \text{K})} = 4.259\ \text{kg/m}^3$$

$$T_{01} = T_1\left(1 + \frac{k-1}{2}\mathrm{Ma}_1^2\right) = (450\ \text{K})\left(1 + \frac{1.4\text{-}1}{2}0.3^2\right) = 458.1\ \text{K}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(450\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 425.2\ \text{m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 0.3(425.2\ \text{m/s}) = 127.6\ \text{m/s}$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1 = (4.259\ \text{kg/m}^3)(0.08 \times 0.08\ \text{m}^2)(127.6\ \text{m/s}) = 3.477\ \text{kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are

$T_{02}/T_0^* = 1$ (since $\mathrm{Ma}_2 = 1$)

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\mathrm{Ma}_1^2[2+(k-1)\mathrm{Ma}_1^2]}{(1+k\mathrm{Ma}_1^2)^2} = \frac{(1.4+1)0.3^2[2+(1.4-1)0.3^2]}{(1+1.4\times 0.3^2)^2} = 0.3469$$ Ther

efore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.3469} \quad \rightarrow \quad T_{02} = T_{01}/0.3469 = (458.1\ \text{K})/0.3469 = 1320.7\ \text{K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (3.477\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})(1320.7 - 458.1)\ \text{K} = \mathbf{3014\ kW}$$

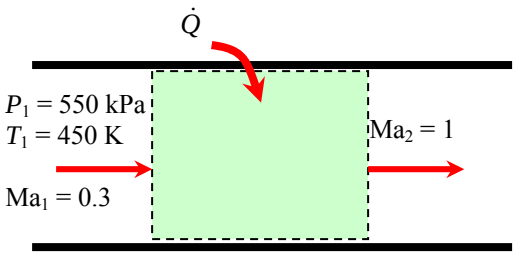
Discussion It can also be shown that $T_2 = 1101$ K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. We can also solve this problem using the Rayleigh function values listed in Table A-34.

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17-150 Helium flowing at a subsonic velocity in a duct is accelerated by heating. The highest rate of heat transfer without affecting the inlet conditions is to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

Properties We take the properties of helium to be $k = 1.667$, $c_p = 5.193$ kJ/kg·K, and $R = 2.077$ kJ/kg·K (Table A-2a).



Analysis Heat transfer will stop when the flow is choked, and thus $\mathrm{Ma}_2 = V_2/c_2 = 1$. The inlet density and stagnation temperature are

$$\rho_1 = \frac{P_1}{RT_1} = \frac{550\ \text{kPa}}{(2.077\ \text{kJ/kgK})(450\ \text{K})} = 0.5885\ \text{kg/m}^3$$

$$T_{01} = T_1\left(1 + \frac{k-1}{2}\mathrm{Ma}_1^2\right) = (450\ \text{K})\left(1 + \frac{1.667\text{-}1}{2}0.3^2\right) = 463.5\ \text{K}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.667)(2.077\ \text{kJ/kg}\cdot\text{K})(450\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 1248\ \text{m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 0.3(1248\ \text{m/s}) = 374.5\ \text{m/s}$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1 = (0.5885\ \text{kg/m}^3)(0.08\times 0.08\ \text{m}^2)(374.5\ \text{m/s}) = 1.410\ \text{kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are

$T_{02}/T_0^* = 1$ (since $\mathrm{Ma}_2 = 1$)

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\mathrm{Ma}_1^2[2+(k-1)\mathrm{Ma}_1^2]}{(1+k\mathrm{Ma}_1^2)^2} = \frac{(1.667+1)0.33^2[2+(1.667-1)0.3^2]}{(1+1.667\times 0.3^2)^2} = 0.3739$$

Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.3739} \quad \rightarrow \quad T_{02} = T_{01}/0.3739 = (463.5\ \text{K})/0.3739 = 1239.8\ \text{K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (1.410\ \text{kg/s})(5.193\ \text{kJ/kg}\cdot\text{K})(1239.8 - 463.5)\ \text{K} = \mathbf{5685\ kW}$$

Discussion It can also be shown that $T_2 = 930$ K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. Also, in the solution of this problem, we cannot use the values of Table A-34 since they are based on $k = 1.4$.

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17-151 Air flowing at a subsonic velocity in a duct is accelerated by heating. For a specified exit Mach number, the heat transfer for a specified exit Mach number as well as the maximum heat transfer are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

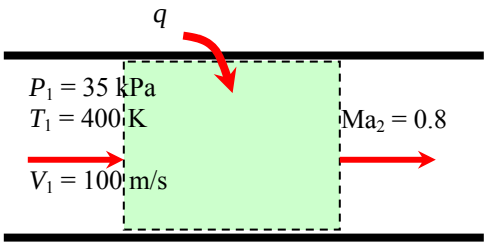
Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K (Table A-2a).

Analysis The inlet Mach number and stagnation temperature are

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(400\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 400.9\ \text{m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{100\ \text{m/s}}{400.9\ \text{m/s}} = 0.2494$$

$$T_{01} = T_1\left(1 + \frac{k-1}{2}\text{Ma}_1^2\right)$$
$$= (400\ \text{K})\left(1 + \frac{1.4-1}{2}0.2494^2\right)$$
$$= 405.0\ \text{K}$$



The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 0.2494$: $T_{01}/T^* = 0.2559$

$\text{Ma}_2 = 0.8$: $T_{02}/T^* = 0.9639$

Then the exit stagnation temperature and the heat transfer are determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{0.9639}{0.2559} = 3.7667 \rightarrow \quad T_{02} = 3.7667T_{01} = 3.7667(405.0\ \text{K}) = 1526\ \text{K}$$

$$q = c_p(T_{02} - T_{01}) = (1.005\ \text{kJ/kg}\cdot\text{K})(1526 - 405)\ \text{K} = \mathbf{1126\ kJ/kg}$$

Maximum heat transfer will occur when the flow is choked, and thus $\text{Ma}_2 = 1$ and thus $T_{02}/T^* = 1$. Then,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.2559} \rightarrow \quad T_{02} = T_{01}/0.2559 = 405.0\ \text{K})/0.2559 = 1583\ \text{K}$$

$$q_{\max} = c_p(T_{02} - T_{01}) = (1.005\ \text{kJ/kg}\cdot\text{K})(1583 - 405)\ \text{K} = \mathbf{1184\ kJ/kg}$$

Discussion This is the maximum heat that can be transferred to the gas without affecting the mass flow rate. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease.

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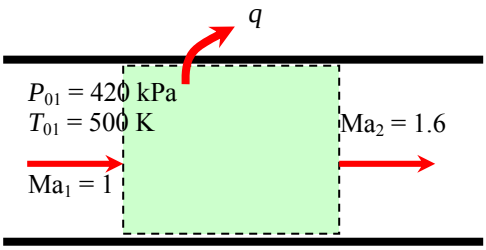
17-152 Air flowing at sonic conditions in a duct is accelerated by cooling. For a specified exit Mach number, the amount of heat transfer per unit mass is to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K (Table A-2a).

Analysis Noting that $\text{Ma}_1 = 1$, the inlet stagnation temperature is

$$T_{01} = T_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)$$
$$= (500\text{ K})\left(1+\frac{1.4-1}{2}1^2\right) = 600\text{ K}$$



The Rayleigh flow functions T_0/T_0^* corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 1$: $\quad T_{01}/T_0^* = 1$

$\text{Ma}_2 = 1.6$: $\quad T_{02}/T_0^* = 0.8842$

Then the exit stagnation temperature and heat transfer are determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{0.8842}{1} = 0.8842 \quad \rightarrow \quad T_{02} = 0.8842T_{01} = 0.8842(600\text{ K}) = 530.5\text{ K}$$

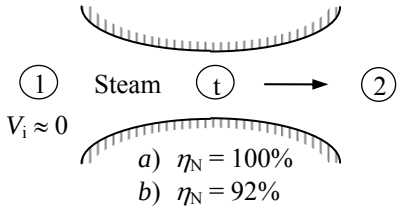
$$q = c_p(T_{02}-T_{01}) = (1.005\text{ kJ/kg}\cdot\text{K})(530.5-600)\text{ K} = \mathbf{-69.8\ kJ/kg}$$

Discussion The negative sign confirms that the gas needs to be cooled in order to be accelerated. Also, it can be shown that the thermodynamic temperature drops to 351 K at the exit

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17-153 Saturated steam enters a converging-diverging nozzle with a low velocity. The throat area, exit velocity, mass flow rate, and exit Mach number are to be determined for isentropic and 90 percent efficient nozzle cases.

Assumptions **1** Flow through the nozzle is steady and one-dimensional.
2 The nozzle is adiabatic.



Analysis (*a*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{10} = h_1$. At the inlet,

$$h_1 = (h_f + x_1 h_{fg})_{@1.75\,\text{MPa}} = 878.16 + 0.90 \times 1917.1 = 2603.5\ \text{kJ/kg}$$
$$s_1 = (s_f + x_1 s_{fg})_{@1.75\,\text{MPa}} = 2.3844 + 0.90 \times 4.0033 = 5.9874\ \text{kJ/kg} \cdot \text{K}$$

At the exit, $P_2 = 1.2$ MPa and $s_2 = s_{2s} = s_1 = 5.9874$ kJ/kg·K. Thus,

$$s_2 = s_f + x_2 s_{fg} \rightarrow 5.9874 = 2.2159 + x_2(4.3058) \rightarrow x_2 = 0.8759$$
$$h_2 = h_f + x_2 h_{fg} = 798.33 + 0.8759 \times 1985.4 = 2537.4\ \text{kJ/kg}$$
$$\nu_2 = \nu_f + x_2 \nu_{fg} = 0.001138 + 0.8759 \times (0.16326 - 0.001138) = 0.14314\ \text{m}^3/\text{kg}$$

Then the exit velocity is determined from the steady-flow energy balance to be

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \ \rightarrow\ 0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(2603.5 - 2537.4)\text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{363.7\ m/s}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 = \frac{1}{0.14314\ \text{m}^3/\text{kg}}(25 \times 10^{-4}\ \text{m}^2)(363.7\ \text{m/s}) = \mathbf{6.35\ kg/s}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial r}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\nu)}\right)_s^{1/2}$$

The specific volume of steam at $s_2 = 5.9874$ kJ/kg·K and at pressures just below and just above the specified pressure (1.1 and 1.3 MPa) are determined to be 0.1547 and 0.1333 m^3/kg. Substituting,

$$c_2 = \sqrt{\frac{(1300 - 1100)\text{kPa}}{\left(\frac{1}{0.1333} - \frac{1}{0.1547}\right)\text{kg/m}^3}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kPa} \cdot \text{m}^3}\right)} = 438.9\ \text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{363.7\ \text{m/s}}{438.9\ \text{m/s}} = \mathbf{0.829}$$

The steam is saturated, and thus the critical pressure which occurs at the throat is taken to be

$$P_t = P^* = 0.576 \times P_{01} = 0.576 \times 1.75 = 1.008\ \text{MPa}$$

Then at the throat,

$$P_t = 1.008\ \text{MPa} \ \text{ and } \ s_t = s_1 = 5.9874\ \text{kJ/kg} \cdot \text{K}$$

Thus,

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$$h_t = 2507.7 \text{ kJ/kg}$$
$$\nu_t = 0.1672 \text{ m}^3/\text{kg}$$

Then the throat velocity is determined from the steady-flow energy balance,

$$h_1 + \frac{V_1^{2\nearrow 0}}{2} = h_t + \frac{V_t^2}{2} \rightarrow 0 = h_t - h_1 + \frac{V_t^2}{2}$$

Solving for V_t,

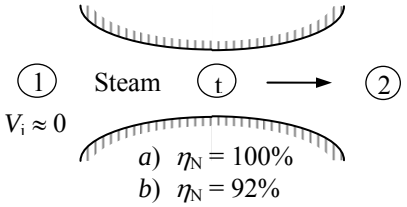
$$V_t = \sqrt{2(h_1 - h_t)} = \sqrt{2(2603.5 - 2507.7)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 437.7 \text{ m/s}$$

Thus the throat area is

$$A_t = \frac{\dot{m}\nu_t}{V_t} = \frac{(6.35 \text{ kg/s})(0.1672 \text{ m}^3/\text{kg})}{437.7 \text{ m/s}} = 24.26 \times 10^{-4} \text{ m}^2 = \mathbf{24.26 \text{ cm}^2}$$

(*b*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{10} = h_1$. At the inlet,

$$h_1 = (h_f + x_1 h_{fg})_{@1.75 \text{ MPa}} = 878.16 + 0.90 \times 1917.1 = 2603.5 \text{ kJ/kg}$$
$$s_1 = (s_f + x_1 s_{fg})_{@1.75 \text{ MPa}} = 2.3844 + 0.90 \times 4.0033 = 5.9874 \text{ kJ/kg} \cdot \text{K}$$



At state 2s, $P_2 = 1.2$ MPa and $s_2 = s_{2s} = s_1 = 5.9874$ kJ/kg·K. Thus,

$$s_{2s} = s_f + x_2 s_{fg} \rightarrow 5.9874 = 2.2159 + x_2(4.3058) \rightarrow x_{2s} = 0.8759$$
$$h_{2s} = h_f + x_2 h_{fg} = 798.33 + 0.8759 \times 1985.4 = 2537.4 \text{ kJ/kg}$$

The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.92 = \frac{2603.5 - h_2}{2603.4 - 2537.4} \longrightarrow h_2 = 2542.7 \text{ kJ/kg}$$

Therefore at the exit, $P_2 = 1.2$ MPa and $h_2 = 2542.7$ kJ/kg·K. Thus,

$$h_2 = h_f + x_2 h_{fg} \longrightarrow 2542.7 = 798.33 + x_2(1985.4) \longrightarrow x_2 = 0.8786$$
$$s_2 = s_f + x_2 s_{fg} = 2.2159 + 0.8786 \times 4.3058 = 5.9989$$
$$\nu_2 = \nu_f + x_2 \nu_{fg} = 0.001138 + 0.8786 \times (0.16326 - 0.001138) = 0.1436 \text{ kJ/kg}$$

Then the exit velocity is determined from the steady-flow energy balance to be

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \rightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Solving for V_2,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(2603.5 - 2542.7)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{348.9 \text{ m/s}}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 = \frac{1}{0.1436 \text{ m}^3/\text{kg}} (25 \times 10^{-4} \text{ m}^2)(348.9 \text{ m/s}) = \mathbf{6.07 \text{ kg/s}}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\nu)}\right)_s^{1/2}$$

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The specific volume of steam at s_2 = 5.9989 kJ/kg·K and at pressures just below and just above the specified pressure (1.1 and 1.3 MPa) are determined to be 0.1551 and 0.1337 m^3/kg. Substituting,

$$c_2 = \sqrt{\frac{(1300-1100)\text{kPa}}{\left(\frac{1}{0.1337}-\frac{1}{0.1551}\right)\text{kg/m}^3}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kPa}\cdot\text{m}^3}\right)} = 439.7\ \text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{348.9\ \text{m/s}}{439.7\ \text{m/s}} = \mathbf{0.793}$$

The steam is saturated, and thus the critical pressure which occurs at the throat is taken to be

$$P_t = P^* = 0.576 \times P_{01} = 0.576 \times 1.75 = 1.008\ \text{MPa}$$

At state 2ts, P_{ts} = 1.008 MPa and $s_{ts} = s_1$ = 5.9874 kJ/kg·K. Thus, h_{ts} = 2507.7 kJ/kg.

The actual enthalpy of steam at the throat is

$$\eta_N = \frac{h_{01}-h_t}{h_{01}-h_{ts}} \longrightarrow 0.92 = \frac{2603.5-h_t}{2603.5-2507.7} \longrightarrow h_t = 2515.4\ \text{kJ/kg}$$

Therefore at the throat, $P_2 = 1.008$ MPa and $h_t = 2515.4$ kJ/kg. Thus, υ_t = 0.1679 m^3/kg.

Then the throat velocity is determined from the steady-flow energy balance,

$$h_1 + \frac{V_1^{2\nearrow 0}}{2} = h_t + \frac{V_t^2}{2} \ \rightarrow\ 0 = h_t - h_1 + \frac{V_t^2}{2}$$

Solving for V_t,

$$V_t = \sqrt{2(h_1 - h_t)} = \sqrt{2(2603.5-2515.4)\text{kJ/kg}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 419.9\ \text{m/s}$$

Thus the throat area is

$$A_t = \frac{\dot{m}\upsilon_t}{V_t} = \frac{(6.07\ \text{kg/s})(0.1679\ \text{m}^3/\text{kg})}{419.9\ \text{m/s}} = 24.30\times10^{-4}\,\text{m}^2 = \mathbf{24.30\ cm^2}$$

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Fundamentals of Engineering (FE) Exam Problems

17-154 An aircraft is cruising in still air at 5°C at a velocity of 400 m/s. The air temperature at the nose of the aircraft where stagnation occurs is

(a) 5°C (b) 25°C (c) 55°C (d) 80°C (e) 85°C

Answer (e) 85°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
T1=5 "C"
Vel1= 400 "m/s"
T1_stag=T1+Vel1^2/(2*Cp*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Tstag=T1 "Assuming temperature rise"
W2_Tstag=Vel1^2/(2*Cp*1000) "Using just the dynamic temperature"
W3_Tstag=T1+Vel1^2/(Cp*1000) "Not using the factor 2"
```

17-155 Air is flowing in a wind tunnel at 25°C, 80 kPa, and 250 m/s. The stagnation pressure at a probe inserted into the flow stream is

(a) 87 kPa (b) 93 kPa (c) 113 kPa (d) 119 kPa (e) 125 kPa

Answer (c) 113 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
T1=25 "K"
P1=80 "kPa"
Vel1= 250 "m/s"
T1_stag=(T1+273)+Vel1^2/(2*Cp*1000) "C"
T1_stag/(T1+273)=(P1_stag/P1)^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
T11_stag/T1=(W1_P1stag/P1)^((k-1)/k);  T11_stag=T1+Vel1^2/(2*Cp*1000)  "Using deg. C for temperatures"
T12_stag/(T1+273)=(W2_P1stag/P1)^((k-1)/k);  T12_stag=(T1+273)+Vel1^2/(Cp*1000)  "Not using the factor 2"
T13_stag/(T1+273)=(W3_P1stag/P1)^(k-1);  T13_stag=(T1+273)+Vel1^2/(2*Cp*1000)  "Using wrong isentropic relation"
```

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17-156 An aircraft is reported to be cruising in still air at -20°C and 40 kPa at a Mach number of 0.86. The velocity of the aircraft is

(a) 91 m/s (b) 220 m/s (c) 186 m/s (d) 280 m/s (e) 378 m/s

Answer (d) 280 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
T1=-20+273 "K"
P1=40 "kPa"
Mach=0.86
VS1=SQRT(k*R*T1*1000)
Mach=Vel1/VS1

"Some Wrong Solutions with Common Mistakes:"
W1_vel=Mach*VS2; VS2=SQRT(k*R*T1) "Not using the factor 1000"
W2_vel=VS1/Mach "Using Mach number relation backwards"
W3_vel=Mach*VS3; VS3=k*R*T1 "Using wrong relation"
```

17-157 Air is flowing in a wind tunnel at 12°C and 66 kPa at a velocity of 230 m/s. The Mach number of the flow is

(a) 0.54 (b) 0.87 (c) 3.3 (d) 0.36 (e) 0.68

Answer (e) 0.68

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
T1=12+273 "K"
P1=66 "kPa"
Vel1=230 "m/s"
VS1=SQRT(k*R*T1*1000)
Mach=Vel1/VS1

"Some Wrong Solutions with Common Mistakes:"
W1_Mach=Vel1/VS2; VS2=SQRT(k*R*(T1-273)*1000) "Using C for temperature"
W2_Mach=VS1/Vel1 "Using Mach number relation backwards"
W3_Mach=Vel1/VS3; VS3=k*R*T1 "Using wrong relation"
```

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17-158 Consider a converging nozzle with a low velocity at the inlet and sonic velocity at the exit plane. Now the nozzle exit diameter is reduced by half while the nozzle inlet temperature and pressure are maintained the same. The nozzle exit velocity will

(a) remain the same. (b) double. (c) quadruple. (d) go down by half. (e) go down to one-fourth.

Answer (a) remain the same.

17-159 Air is approaching a converging-diverging nozzle with a low velocity at 12°C and 200 kPa, and it leaves the nozzle at a supersonic velocity. The velocity of air at the throat of the nozzle is

(a) 338 m/s (b) 309 m/s (c) 280 m/s (d) 256 m/s (e) 95 m/s

Answer (b) 309 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
"Properties at the inlet"
T1=12+273 "K"
P1=200 "kPa"
Vel1=0 "m/s"
To=T1 "since velocity is zero"
Po=P1
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Vthroat=SQRT(k*R*T1*1000) "Using T1 for temperature"
W2_Vthroat=SQRT(k*R*T2_throat*1000); T2_throat=2*(To-273)/(k+1) "Using C for temperature"
W3_Vthroat=k*R*T_throat "Using wrong relation"
```

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17-160 Argon gas is approaching a converging-diverging nozzle with a low velocity at 20°C and 120 kPa, and it leaves the nozzle at a supersonic velocity. If the cross-sectional area of the throat is 0.015 m^2, the mass flow rate of argon through the nozzle is

(a) 0.41 kg/s (b) 3.4 kg/s (c) 5.3 kg/s (d) 17 kg/s (e) 22 kg/s

Answer (c) 5.3 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=0.5203 "kJ/kg.K"
R=0.2081 "kJ/kg.K"
A=0.015 "m^2"
"Properties at the inlet"
T1=20+273 "K"
P1=120 "kPa"
Vel1=0 "m/s"
To=T1 "since velocity is zero"
Po=P1
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
rho_throat=P_throat/(R*T_throat)
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)
m=rho_throat*A*V_throat

"Some Wrong Solutions with Common Mistakes:"
W1_mass=rho_throat*A*V1_throat;  V1_throat=SQRT(k*R*T1_throat*1000); T1_throat=2*(To-273)/(k+1) "Using C for temp"
W2_mass=rho2_throat*A*V_throat; rho2_throat=P1/(R*T1)  "Using density at inlet"
```

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17-161 Carbon dioxide enters a converging-diverging nozzle at 60 m/s, 310°C, and 300 kPa, and it leaves the nozzle at a supersonic velocity. The velocity of carbon dioxide at the throat of the nozzle is

(a) 125 m/s (b) 225 m/s (c) 312 m/s (d) 353 m/s (e) 377 m/s

Answer (d) 353 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
"Properties at the inlet"
T1=310+273 "K"
P1=300 "kPa"
Vel1=60 "m/s"
To=T1+Vel1^2/(2*Cp*1000)
To/T1=(Po/P1)^((k-1)/k)
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Vthroat=SQRT(k*R*T1*1000) "Using T1 for temperature"
W2_Vthroat=SQRT(k*R*T2_throat*1000); T2_throat=2*(T_throat-273)/(k+1) "Using C for temperature"
W3_Vthroat=k*R*T_throat "Using wrong relation"
```

17-162 Consider gas flow through a converging-diverging nozzle. Of the five statements below, select the one that is incorrect:

(a) The fluid velocity at the throat can never exceed the speed of sound.

(b) If the fluid velocity at the throat is below the speed of sound, the diversion section will act like a diffuser.

(c) If the fluid enters the diverging section with a Mach number greater than one, the flow at the nozzle exit will be supersonic.

(d) There will be no flow through the nozzle if the back pressure equals the stagnation pressure.

(e) The fluid velocity decreases, the entropy increases, and stagnation enthalpy remains constant during flow through a normal shock.

Answer (c) If the fluid enters the diverging section with a Mach number greater than one, the flow at the nozzle exit will be supersonic.

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17-163 Combustion gases with $k = 1.33$ enter a converging nozzle at stagnation temperature and pressure of 350°C and 400 kPa, and are discharged into the atmospheric air at 20°C and 100 kPa. The lowest pressure that will occur within the nozzle is

(a) 13 kPa (b) 100 kPa (c) 216 kPa (d) 290 kPa (e) 315 kPa

Answer (c) 216 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.33
Po=400 "kPa"
"The critical pressure is"
P_throat=Po*(2/(k+1))^(k/(k-1))
"The lowest pressure that will occur in the nozzle is the higher of the critical or atmospheric pressure."

"Some Wrong Solutions with Common Mistakes:"
W2_Pthroat=Po*(1/(k+1))^(k/(k-1)) "Using wrong relation"
W3_Pthroat=100 "Assuming atmospheric pressure"
```

17-164 ··· 17-166 Design and Essay Problems

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