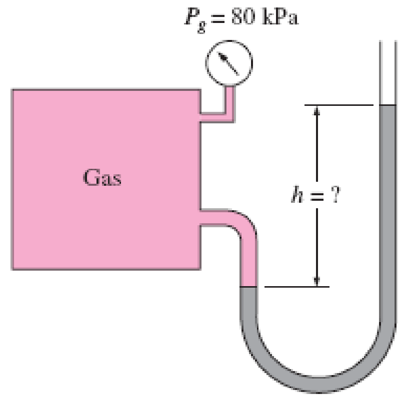
2-53 CD EES Both a gage and a manometer are attached to a gas to measure its pressure. For a specified reading of gage pressure, the difference between the fluid levels of the two arms of the manometer is to be determined for mercury and water.



Properties The densities of water and mercury are given to be ρ_{water} = 1000 kg/m^3 and be ρ_{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,

$$h = \frac{P_{\text{gage}}}{\rho_{H_2O} g} = \frac{80 \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1 \text{ kN}}\right) = \mathbf{8.16\ m}$$

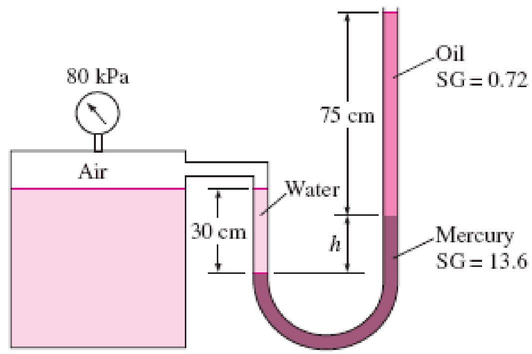
2-67 The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

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

$$P_1 + \rho_w gh_w - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{oil}} gh_{\text{oil}} = P_{atm}$$



Rearranging,

$$P_1 - P_{\text{atm}} = \rho_{\text{oil}} gh_{\text{oil}} + \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_w gh_w$$

or,

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

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.

2-85 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 ρ = 1.16 kg/m^3. The density of helium gas is 1/7th of this.

Analysis The buoyancy force acting on the balloon is

$$V_{balloon} = 4\pi r^3/3 = 4\pi(5\text{ m})^3/3 = 523.6\text{ m}^3$$

$$F_B = \rho_{\text{air}} g V_{\text{balloon}}$$
$$= (1.16\text{ kg/m}^3)(9.81\text{m/s}^2)(523.6\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 5958\text{ N}$$

The total mass is

$$m_{\text{He}} = \rho_{\text{He}} V = \left(\frac{1.16}{7}\text{kg/m}^3\right)(523.6\text{ m}^3) = 86.8\text{ kg}$$

$$m_{\text{total}} = m_{\text{He}} + m_{\text{people}} = 86.8 + 2\times 70 = 226.8\text{ kg}$$

The total weight is

$$W = m_{\text{total}} g = (226.8\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 2225\text{ N}$$

Thus the net force acting on the balloon is

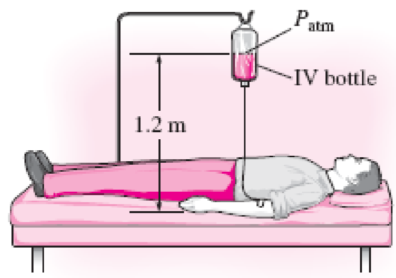
$$F_{\text{net}} = F_B - W = 5958 - 2225 = 3733\text{ N}$$

Then the acceleration becomes

$$a = \frac{F_{\text{net}}}{m_{\text{total}}} = \frac{3733\text{ N}}{226.8\text{ kg}}\left(\frac{1\text{kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{16.5\text{ m/s}^2}$$

2-96 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 ρ = 1020 kg/m^3.

Analysis (*a*) Noting that the IV fluid and the blood pressures balance each other when the bottle is 1.2 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 1.2 m,

$$\begin{aligned} P_{\text{gage, arm}} &= P_{\text{abs}} - P_{\text{atm}} = \rho g h_{\text{arm-bottle}} \\ &= (1020\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(1.20\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) \\ &= \mathbf{12.0\ kPa} \end{aligned}$$

(*b*) To provide a gage pressure of 20 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

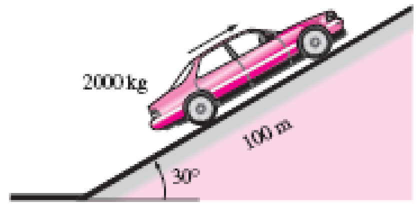
$$\begin{aligned} h_{\text{arm-bottle}} &= \frac{P_{\text{gage, arm}}}{\rho g} \\ &= \frac{20\ \text{kPa}}{(1020\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right) = \mathbf{2.0\ m} \end{aligned}$$

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.

3-40 A car is to climb a hill in 10 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,

$$\dot{W}_g = mg(z_2 - z_1)/\Delta t = (2000\text{ kg})(9.81\text{ m/s}^2)(50\text{ m})\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right)/(10\text{ s}) = 98.1\text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 0 + 98.1 = \mathbf{98.1\ kW}$

(*b*) The power needed to accelerate is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(2000\text{ kg})\left[(30\text{ m/s})^2 - 0\right]\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right)/(10\text{ s}) = 90\text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 90 + 98.1 = \mathbf{188.1\ kW}$

(*c*) The power needed to decelerate is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(2000\text{ kg})\left[(5\text{ m/s})^2 - (35\text{ m/s})^2\right]\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right)/(10\text{ s}) = -120\text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = -120 + 98.1 = \mathbf{-21.9\ kW}$ (breaking power)

3-54 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.

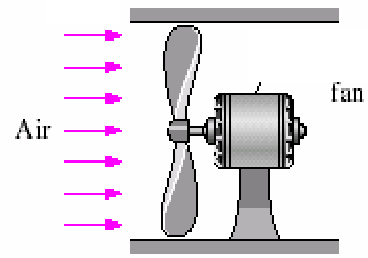
Assumptions The fan operates steadily.

Properties The density of air is given to be $\rho = 1.18\ \text{kg/m}^3$.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}} / dt^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$



where

$$\dot{m}_{\text{air}} = \rho \dot{\boldsymbol{V}} = (1.18\ \text{kg/m}^3)(4\ \text{m}^3/\text{s}) = 4.72\ \text{kg/s}$$

Substituting, the minimum power input required is determined

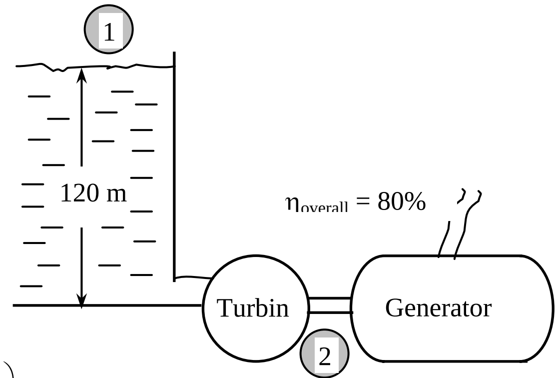
$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2} = (4.72\ \text{kg/s}) \frac{(10\ \text{m/s})^2}{2} \left(\frac{1\ \text{J/kg}}{1\ \text{m}^2/\text{s}^2} \right) = 236\ \text{J/s} = \mathbf{236\ W}$$

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.

3-105 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 ρ = 1000 kg/m^3 = 1 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.

$$e_{\text{mech}} = pe = gz = (9.81\,\text{m/s}^2)(120\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 1.177\ \text{kJ/kg}$$

The mass flow rate is

$$\dot{m} = \rho\dot{V} = (1000\ \text{kg/m}^3)(100\ \text{m}^3/\text{s}) = 100{,}000\ \text{kg/s}$$

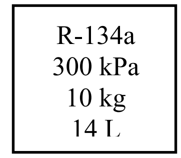
Then the maximum and actual electric power generation become

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (100{,}000\ \text{kg/s})(1.177\ \text{kJ/kg})\left(\frac{1\,\text{MW}}{1000\,\text{kJ/s}}\right) = 117.7\ \text{MW}$$

$$\dot{W}_{\text{electric}} = \eta_{\text{overall}}\dot{W}_{\max} = 0.80(117.7\ \text{MW}) = \mathbf{94.2\ MW}$$

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.

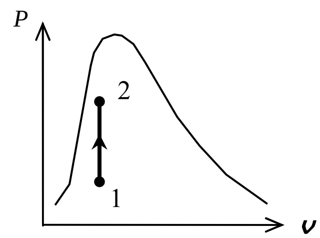
4-41 A rigid container that is filled with R-134a is heated. The temperature and total enthalpy are to be determined at the initial and final states.



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



$$T_1 = T_{\text{sat @ 300 kPa}} = \mathbf{0.61°C}$$

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,

$$T_2 = T_{\text{sat @ 600 kPa}} = \mathbf{21.55°C}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{(0.0014 - 0.0008199)\ \text{m}^3/\text{kg}}{(0.034295 - 0.0008199)\ \text{m}^3/\text{kg}} = 0.01733$$

$$h_2 = h_f + x_2 h_{fg} = 81.51 + (0.01733)(180.90) = 84.64\ \text{kJ/kg}$$

$$H_2 = mh_2 = (10\ \text{kg})(84.64\ \text{kJ/kg}) = \mathbf{846.4\ kJ}$$

4-54 CD EES 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 350°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,

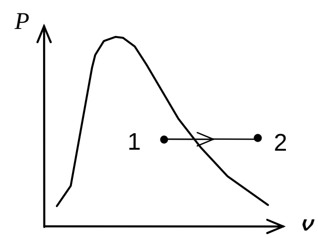
$$T = T_{\text{sat@800 kPa}} = \mathbf{170.41°C}$$

(b) The total mass in this case can easily be determined by adding the mass of each phase,

$$m_f = \frac{V_f}{v_f} = \frac{0.1 \text{ m}^3}{0.001115 \text{ m}^3/\text{kg}} = 89.704 \text{ kg}$$

$$m_g = \frac{V_g}{v_g} = \frac{0.9 \text{ m}^3}{0.24035 \text{ m}^3/\text{kg}} = 3.745 \text{ kg}$$

$$m_t = m_f + m_g = 89.704 + 3.745 = \mathbf{93.45 \text{ kg}}$$



(c) At the final state water is superheated vapor, and its specific volume is

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 350^\circ \text{C} \end{array}\right\} v_2 = 0.35442 \text{ m}^3/\text{kg} \qquad \text{(Table A-6)}$$

Then,

$$V_2 = m_t v_2 = (93.45 \text{ kg})(0.35442 \text{ m}^3/\text{kg}) = \mathbf{33.12 \text{ m}^3}$$

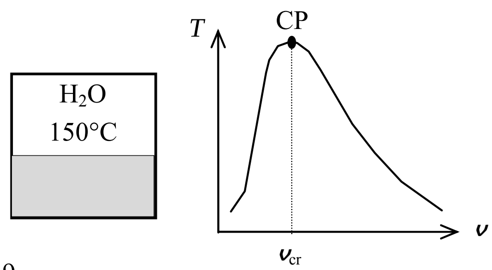
4-58 A rigid vessel that contains a saturated liquid-vapor mixture is heated until it reaches the critical state. The mass of the liquid water and the volume occupied by the liquid at the initial state are to be determined.

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,

$$v_1 = v_2 = v_{cr} = 0.003106 \text{ m}^3/\text{kg} \qquad \text{(last row of Table A-4)}$$

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

$$m_f = (1 - x_1)m_t = (1 - 0.005149)(96.60) = \mathbf{96.10 \ kg}$$

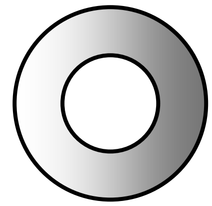
$$V_f = m_f v_f = (96.10 \text{ kg})(0.001091 \text{ m}^3/\text{kg}) = \mathbf{0.105 \ m^3}$$

4-79 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

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = 0.0906 \text{ kg}$$

$$m_2 = \frac{P_1 V}{RT_2} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(323 \text{ K})} = 0.0836 \text{ kg}$$

$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070\ kg}$$

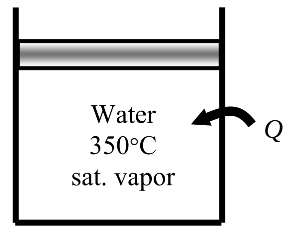
4-95 Water vapor is heated at constant pressure. The final temperature is to be determined using ideal gas equation, the compressibility charts, and the steam tables.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 647.1 \text{ K}, \qquad P_{cr} = 22.06 \text{ MPa}$$

Analysis (*a*) From the ideal gas equation,

$$T_2 = T_1 \frac{v_2}{v_1} = (350 + 273 \text{ K})(2) = \mathbf{1246\ K}$$



(*b*) The pressure of the steam is

$$P_1 = P_2 = P_{sat@350°C} = 16{,}529 \text{ kPa}$$

From the compressibility chart at the initial state (Fig. A-15),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{623 \text{ K}}{647.1 \text{ KR}} = 0.963 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{16.529 \text{ MPa}}{22.06 \text{ MPa}} = 0.749 \end{aligned}\right\} Z_1 = 0.593,\ v_{R1} = 0.75$$

At the final state,

$$\left.\begin{aligned} P_{R2} &= P_{R1} = 0.749 \\ v_{R2} &= 2v_{R1} = 2(0.75) = 1.50 \end{aligned}\right\} Z_2 = 0.88$$

Thus,

$$T_2 = \frac{P_2 v_2}{Z_2 R} = \frac{P_2}{Z_2}\frac{v_{R2} T_{cr}}{P_{cr}} = \frac{16{,}529 \text{ kPa}}{0.88}\frac{(1.50)(647.1 \text{ K})}{22{,}060 \text{ kPa}} = \mathbf{826\ K}$$

(*c*) From the superheated steam table,

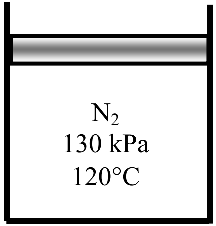
$$\left.\begin{aligned} T_1 &= 350°\text{C} \\ x_1 &= 1 \end{aligned}\right\} v_1 = 0.008806 \text{ m}^3\text{/kg} \qquad \text{(Table A-4)}$$

$$\left.\begin{aligned} P_2 &= 16{,}529 \text{ kPa} \\ v_2 &= 2v_1 = 0.01761 \text{ m}^3\text{/kg} \end{aligned}\right\} T_2 = 477°\text{C} = \mathbf{750\ K} \qquad \text{(from Table A-6 or EES)}$$

5-8 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



$$m = \frac{P_1 \boldsymbol{V}_1}{RT_1} = \frac{(130\text{ kPa})(0.07\text{ m}^3)}{(0.2968\text{ kJ/kg.K})(120+273\text{ K})} = 0.07802\text{ kg}$$

$$\boldsymbol{V}_2 = \frac{mRT_2}{P_2} = \frac{(0.07802\text{ kg})(0.2968\text{ kPa.m}^3\text{/kg.K})(100+273\text{ K})}{100\text{ kPa}} = 0.08637\text{ m}^3$$

The polytropic index is determined from

$$P_1 \boldsymbol{V}_1^n = P_2 \boldsymbol{V}_2^n \longrightarrow (130\text{ kPa})(0.07\text{ m}^3)^n = (100\text{ kPa})(0.08637\text{ m}^3)^n \longrightarrow n = 1.249$$

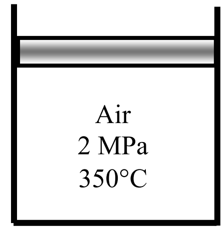
The boundary work is determined from

$$W_b = \frac{P_2 \boldsymbol{V}_2 - P_1 \boldsymbol{V}_1}{1-n} = \frac{(100\text{ kPa})(0.08637\text{ m}^3) - (130\text{ kPa})(0.07\text{ m}^3)}{1-1.249} = \mathbf{1.86\text{ kJ}}$$

5-25 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:



$$V_1 = \frac{mRT}{P_1} = \frac{(0.15\text{ kg})(0.287\text{ kJ/kg.K})(350+273\text{ K})}{(2000\text{ kPa})} = 0.01341\text{ m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.15\text{ kg})(0.287\text{ kJ/kg.K})(350+273\text{ K})}{(500\text{ kPa})} = 0.05364\text{ m}^3$$

$$W_{b,1-2} = P_1V_1\ln\left(\frac{V_2}{V_1}\right) = (2000\text{ kPa})(0.01341\text{ m}^3)\ln\left(\frac{0.05364\text{ m}^3}{0.01341\text{ m}^3}\right) = \mathbf{37.18\text{ kJ}}$$

For the polytropic compression process:

$$P_2V_2^n = P_3V_3^n \longrightarrow (500\text{ kPa})(0.05364\text{ m}^3)^{1.2} = (2000\text{ kPa})V_3^{1.2} \longrightarrow V_3 = 0.01690\text{ m}^3$$

$$W_{b,2-3} = \frac{P_3V_3 - P_2V_2}{1-n} = \frac{(2000\text{ kPa})(0.01690\text{ m}^3) - (500\text{ kPa})(0.05364\text{ m}^3)}{1-1.2} = \mathbf{-34.86\text{ kJ}}$$

For the constant pressure compression process:

$$W_{b,3-1} = P_3(V_1 - V_3) = (2000\text{ kPa})(0.01341 - 0.01690)\text{m}^3 = \mathbf{-6.97\text{ kJ}}$$

The net work for the cycle is the sum of the works for each process

$$W_{\text{net}} = W_{b,1-2} + W_{b,2-3} + W_{b,3-1} = 37.18 + (-34.86) + (-6.97) = \mathbf{-4.65\text{ kJ}}$$

Closed System Energy Analysis

5-29 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.

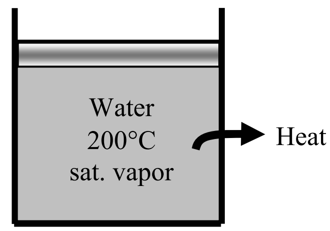
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,in} - Q_{out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{out} = W_{b,in} - m(u_2 - u_1)$$

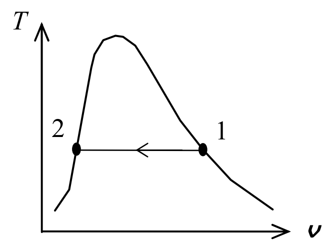


The properties at the initial and final states are (Table A-4)

$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} v_1 = v_g = 0.12721\ \text{m}^3/\text{kg} \\ u_1 = u_g = 2594.2\ \text{kJ/kg} \end{array}$$

$$P_1 = P_2 = 1554.9\ \text{kPa}$$

$$\left.\begin{array}{l} T_2 = 200°\text{C} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} v_2 = v_f = 0.001157\ \text{m}^3/\text{kg} \\ u_2 = u_f = 850.46\ \text{kJ/kg} \end{array}$$



The work done during this process is

$$w_{b,out} = \int_1^2 P d\mathbf{V} = P(v_2 - v_1) = (1554.9\ \text{kPa})(0.001157 - 0.12721)\ \text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = -196.0\ \text{kJ/kg}$$

That is,

$$w_{b,in} = \mathbf{196.0\ kJ/kg}$$

Substituting the energy balance equation, we get

$$q_{out} = w_{b,in} - (u_2 - u_1) = w_{b,in} + u_{fg} = 196.0 + 1743.7 = \mathbf{1940\ kJ/kg}$$

5-37 A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled at constant pressure. The amount of heat loss is to be determined, and the process is to be shown on a *T*-v diagram.

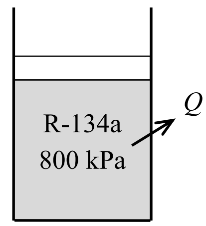
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{out} = m(h_2 - h_1)$$

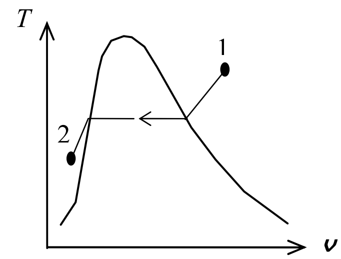


since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are

(Tables A-11 through A-13)

$$\left.\begin{aligned} P_1 &= 800 \text{ kPa} \\ T_1 &= 70°\text{C} \end{aligned}\right\} h_1 = 306.88 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 800 \text{ kPa} \\ T_2 &= 15°\text{C} \end{aligned}\right\} h_2 \cong h_{f@15°\text{C}} = 72.34 \text{ kJ/kg}$$



Substituting,

Q_{out} = - (5 kg)(72.34 - 306.88) kJ/kg = **1173 kJ**

5-40 CD EES 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.

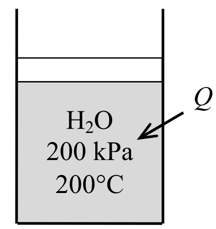
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium. **4** The spring is a linear spring.

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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(u_2 - u_1) + W_{b,out}$$



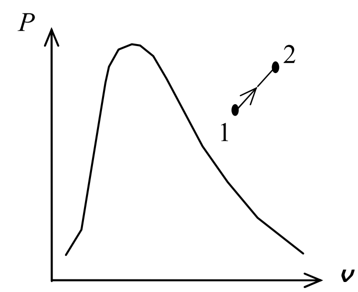
The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.4628 \text{ kg}$$

$$v_2 = \frac{V_2}{m} = \frac{0.6 \text{ m}^3}{0.4628 \text{ kg}} = 1.2966 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ v_2 = 1.2966 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{1132°C} \\ u_2 = 4325.2 \text{ kJ/kg} \end{array}$$



(*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,

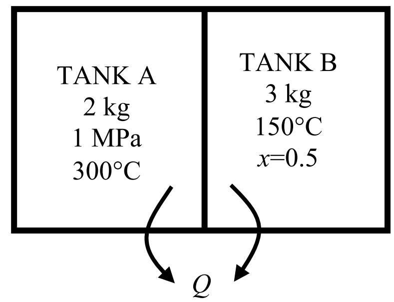
$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 500)\text{kPa}}{2}(0.6 - 0.5)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{35 \text{ kJ}}$$

(*c*) From the energy balance we have

$$Q_{in} = (0.4628 \text{ kg})(4325.2 - 2654.6)\text{kJ/kg} + 35 \text{ kJ} = \mathbf{808 \text{ kJ}}$$

5-42 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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

The properties of steam in both tanks at the initial state are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_{1,A} = 1000 \text{ kPa} \\ T_{1,A} = 300°\text{C} \end{array}\right\} \begin{array}{l} v_{1,A} = 0.25799 \text{ m}^3/\text{kg} \\ u_{1,A} = 2793.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_{1,B} = 150°\text{C} \\ x_1 = 0.50 \end{array}\right\} \begin{array}{ll} v_f = 0.001091, & v_g = 0.39248 \text{ m}^3/\text{kg} \\ u_f = 631.66, & u_{fg} = 1927.4 \text{ kJ/kg} \end{array}$$

$$v_{1,B} = v_f + x_1 v_{fg} = 0.001091 + [0.50 \times (0.39248 - 0.001091)] = 0.19679 \text{ m}^3/\text{kg}$$
$$u_{1,B} = u_f + x_1 u_{fg} = 631.66 + (0.50 \times 1927.4) = 1595.4 \text{ kJ/kg}$$

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

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ v_2 = 0.22127 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = T_{sat @ 300 \text{ kPa}} = \mathbf{133.5°C} \\ x_2 = \dfrac{v_2 - v_f}{v_g - v_f} = \dfrac{0.22127 - 0.001073}{0.60582 - 0.001073} = \mathbf{0.3641} \\ u_2 = u_f + x_2 u_{fg} = 561.11 + (0.3641 \times 1982.1) = 1282.8 \text{ kJ/kg} \end{array}$$

(*b*) Substituting,

$$-Q_{out} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B$$
$$= (2 \text{ kg})(1282.8 - 2793.7)\text{kJ/kg} + (3 \text{ kg})(1282.8 - 1595.4)\text{kJ/kg} = -3959 \text{ kJ}$$

or $\quad Q_{out} = \mathbf{3959 \text{ kJ}}$

5-63 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,

$$\begin{aligned}
\Delta\bar{u} &= \int_1^2 \bar{c}_v(T)dT = \int_1^2 \left[(a - R_u) + bT + cT^2 + dT^3\right]dT \\
&= (a - R_u)(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 + T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\
&= (29.11 - 8.314)(800 - 200) - \tfrac{1}{2}(0.1961\times10^{-2})(800^2 - 200^2) \\
&\quad + \tfrac{1}{3}(0.4003\times10^{-5})(800^3 - 200^3) - \tfrac{1}{4}(0.8704\times10^{-9})(800^4 - 200^4) \\
&= 12{,}487 \text{ kJ/kmol}
\end{aligned}$$

$$\Delta u = \frac{\Delta\bar{u}}{M} = \frac{12{,}487 \text{ kJ/kmol}}{2.016 \text{ kg/kmol}} = \mathbf{6194\ kJ/kg}$$

(*b*) Using a constant c_p value from Table A-2b at the average temperature of 500 K,

$$c_{v,\text{avg}} = c_{v@500\text{ K}} = 10.389 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta u = c_{v,\text{avg}}(T_2 - T_1) = (10.389 \text{ kJ/kg}\cdot\text{K})(800 - 200)\text{K} = \mathbf{6233\ kJ/kg}$$

(*c*) Using a constant c_p value from Table A-2a at room temperature,

$$c_{v,\text{avg}} = c_{v@300\text{ K}} = 10.183 \text{ kJ/kg}\cdot\text{K}$$

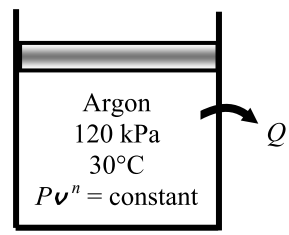
$$\Delta u = c_{v,\text{avg}}(T_2 - T_1) = (10.183 \text{ kJ/kg}\cdot\text{K})(800 - 200)\text{K} = \mathbf{6110\ kJ/kg}$$

5-74 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 $R = 0.2081 \text{ kJ/kg·K}$ and $c_v = 0.3122 \text{ kJ/kg·K}$ (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

$$w_{b,\text{out}} = \frac{RT_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] = \frac{(0.2081 \text{ kJ/kg}\cdot\text{K})(303 \text{ K})}{1\text{-}1.2}\left[\left(\frac{1200}{120}\right)^{0.2/1.2} - 1\right] = -147.5 \text{ kJ/kg}$$

Thus,

$$w_{b,\text{in}} = \mathbf{147.5 \text{ kJ/kg}}$$

The temperature at the final state is

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n\text{-}1)/n} = (303 \text{ K})\left(\frac{1200 \text{ kPa}}{120 \text{ kPa}}\right)^{0.2/1.2} = 444.7 \text{ K}$$

From the energy balance equation,

$$q_{\text{in}} = w_{b,\text{out}} + c_v(T_2 - T_1) = -147.5 \text{ kJ/kg} + (0.3122 \text{ kJ/kg}\cdot\text{K})(444.7 - 303)\text{K} = -103.3 \text{ kJ/kg}$$

Thus,

$$q_{\text{out}} = \mathbf{103.3 \text{ kJ/kg}}$$

5-84 A cylinder equipped with a set of stops for the piston is initially filled with air at a specified state. Heat is transferred to the air until the volume doubled. 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. **4** The compression or expansion process is quasi-equilibrium.

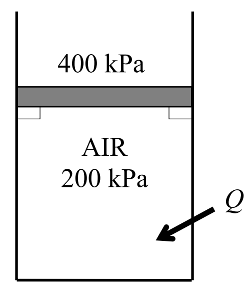
Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1).

Analysis We take the air in the cylinder as 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

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1)$$

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

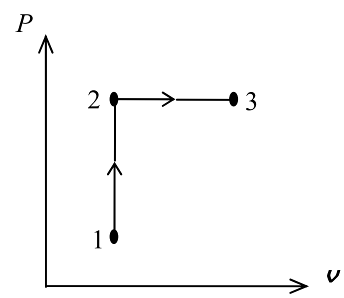


The initial and the final volumes and the final temperature of air are

$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2 \times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1 V_1}{T_1} = \frac{P_3 V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}} \times 2 \times (300\text{ K}) = 1200\text{ K}$$



No work is done during process 1-2 since $V_1 = V_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_{b,out} = \int_1^2 P dV = P_2(V_3 - V_2) = (400\text{ kPa})(2.58 - 1.29)\text{m}^3 = \mathbf{516\ kJ}$$

The initial and final internal energies of air are (Table A-21)

$$u_1 = u_{@300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_3 = u_{@1200\text{ K}} = 933.33\text{ kJ/kg}$$

Then from the energy balance,

Q_{in} = (3 kg)(933.33 - 214.07)kJ/kg + 516 kJ = **2674 kJ**

Alternative solution The specific heat of air at the average temperature of T_{avg} = (300 + 1200)/2 = 750 K is, from Table A-2b, $c_{v,avg}$ = 0.800 kJ/kg.K. Substituting,

$$Q_{in} = m(u_3 - u_1) + W_{b,out} \cong mc_v(T_3 - T_1) + W_{b,out}$$

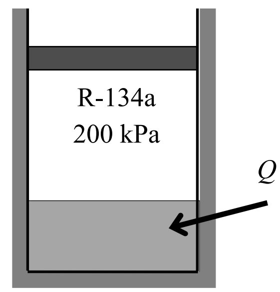
Q_{in} = (3 kg)(0.800 kJ/kg.K)(1200 - 300) K + 516 kJ = **2676 kJ**

5-109 A cylinder is initially filled with saturated liquid-vapor mixture of R-134a at a specified pressure. Heat is transferred to the cylinder until the refrigerant vaporizes completely at constant pressure. The initial volume, the work done, and the total 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 negligible. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis (a) Using property data from R-134a tables (Tables A-11 through A-13), the initial volume of the refrigerant is determined to be

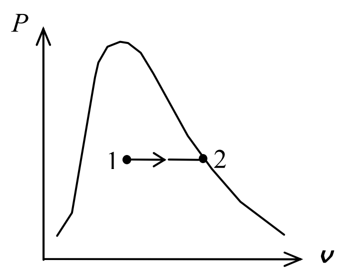
$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} v_f = 0.0007533, & v_g = 0.099867 \text{ m}^3/\text{kg} \\ u_f = 38.28, & u_{fg} = 186.21 \text{ kJ/kg} \end{array}$$



$$v_1 = v_f + x_1 v_{fg}$$
$$= 0.0007533 + 0.25 \times (0.099867 - 0.0007533) = 0.02553 \text{ m}^3/\text{kg}$$
$$u_1 = u_f + x_1 u_{fg} = 38.28 + 0.25 \times 186.21 = 84.83 \text{ kJ/kg}$$

$$V_1 = m v_1 = (0.2 \text{ kg})(0.02553 \text{ m}^3/\text{kg}) = \mathbf{0.005106 \text{ m}^3}$$

(b) The work done during this constant pressure process is



$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@200 \text{ kPa}} = 0.09987 \text{ m}^3/\text{kg} \\ u_2 = u_{g@200 \text{ kPa}} = 224.48 \text{ kJ/kg} \end{array}$$

$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = mP(v_2 - v_1)$$
$$= (0.2 \text{ kg})(200 \text{ kPa})(0.09987 - 0.02553)\text{m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{2.97 \text{ kJ}}$$

(c) 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

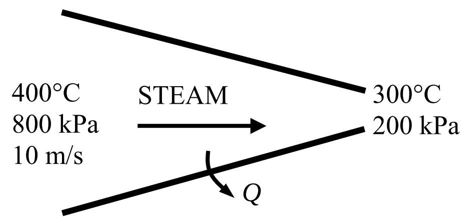
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$
$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U$$
$$Q_{\text{in}} = m(u_2 - u_1) + W_{\text{b,out}}$$

Substituting,

$$Q_{\text{in}} = (0.2 \text{ kg})(224.48 - 84.83)\text{kJ/kg} + 2.97 = \mathbf{30.9 \text{ kJ}}$$

6-44 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:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{out} \quad \text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

or

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + \frac{\dot{Q}_{out}}{\dot{m}}$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.38429 \text{ m}^3/\text{kg} \\ h_1 = 3267.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 1.31623 \text{ m}^3/\text{kg} \\ h_2 = 3072.1 \text{ kJ/kg} \end{array}$$

The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 = \frac{1}{0.38429 \text{ m}^3/\text{s}} (0.08 \text{ m}^2)(10 \text{ m/s}) = 2.082 \text{ kg/s}$$

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

$$\dot{\boldsymbol{V}}_2 = \dot{m}\boldsymbol{v}_2 = (2.082 \text{ kg/s})(1.31623 \text{ m}^3/\text{kg}) = \mathbf{2.74 \text{ m}^3/\text{s}}$$

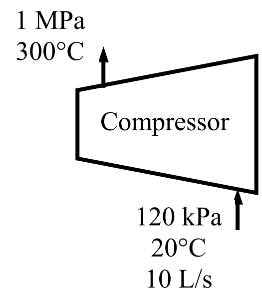
6-51 Air is compressed at a rate of 10 L/s by a compressor. The work required per unit mass and the power required are to be determined.

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

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273 \text{ K})}{120 \text{ kPa}} = 0.7008 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.010 \text{ m}^3/\text{s}}{0.7008 \text{ m}^3/\text{kg}} = 0.01427 \text{ kg/s}$$

Then the power input is determined from the energy balance equation to be

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = (0.01427 \text{ kg/s})(1.018 \text{ kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{4.068\ kW}$$

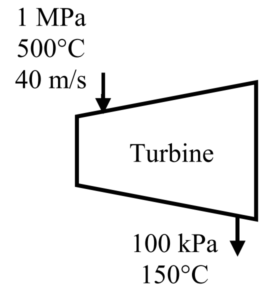
6-60 Air is expanded in an adiabatic turbine. The mass flow rate of the air and the power produced are to be determined.

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+150)/2=325°C=598 K is c_p = 1.051 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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}{}^{\nearrow 0 \text{ (steady)}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{W}_{out}$$

$$\dot{W}_{out} = \dot{m}\left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right) = \dot{m}\left(c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right)$$

The specific volume of air at the inlet and the mass flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500+273\text{ K})}{1000\text{ kPa}} = 0.2219\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\boldsymbol{v}_1} = \frac{(0.2\text{ m}^2)(40\text{ m/s})}{0.2219\text{ m}^3/\text{kg}} = \mathbf{36.06\ kg/s}$$

Similarly at the outlet,

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(150+273\text{ K})}{100\text{ kPa}} = 1.214\text{ m}^3/\text{kg}$$

$$V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{(36.06\text{ kg/s})(1.214\text{ m}^3/\text{kg})}{1\text{ m}^2} = 43.78\text{ m/s}$$

(*b*) Substituting into the energy balance equation gives

$$\dot{W}_{out} = \dot{m}\left(c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right)$$

$$= (36.06\text{ kg/s})\left[(1.051\text{ kJ/kg}\cdot\text{K})(500-150)\text{K} + \frac{(40\text{ m/s})^2 - (43.78\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{13{,}260\ kW}$$

6-80 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),

$$h_1 \cong h_{f\,@\,12°C} = 68.18 \text{ kJ/kg}$$

$$h_2 = h_{@\,1\,\text{MPa},\ 60°C} = 293.38 \text{ kJ/kg}$$

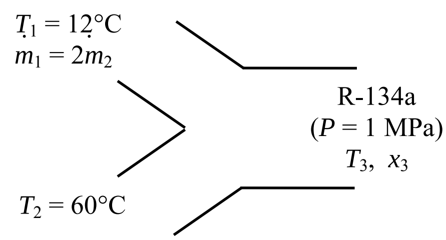
Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0\ (steady)} = 0$$

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2 \quad \text{since } \dot{m}_1 = 2\dot{m}_2$$



Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\ (steady)}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta ke \cong \Delta pe \cong 0)$$

Combining the two gives $2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3 \ \ or \ \ h_3 = (2h_1 + h_2)/3$

Substituting,

$$h_3 = (2\times 68.18 + 293.38)/3 = 143.25 \text{ kJ/kg}$$

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,

$$T_3 = T_{sat\,@\,1\,MPa} = \mathbf{39.37°C}$$

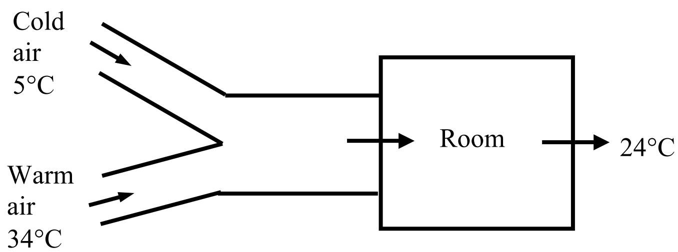
and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{143.25 - 107.32}{163.67} = \mathbf{0.220}$$

6-94 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-21) as



$$h_1 = h_{@278\text{ K}} = 278.13 \text{ kJ/kg}$$

$$h_2 = h_{@\,307\text{ K}} = 307.23 \text{ kJ/kg}$$

$$h_{\text{room}} = h_{@\,297\text{ K}} = 297.18 \text{ kJ/kg}$$

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 $\dot{m}_1 h_1 + 1.6\dot{m}_1 h_2 = 2.6\dot{m}_1 h_3$ *or* $h_3 = (h_1 + 1.6h_2)/2.6$

Substituting,

$$h_3 = (278.13 + 1.6 \times 307.23)/2.6 = 296.04 \text{ kJ/kg}$$

From air table at this enthalpy, the mixture temperature is

$$T_3 = T_{@\,h = 296.04 \text{ kJ/kg}} = 295.9 \text{ K} = \mathbf{22.9°C}$$

(*b*) The mass flow rates are determined as follows

$$v_1 = \frac{RT_1}{P} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(5 + 273 \text{ K})}{105 \text{ kPa}} = 0.7599 \text{ m}^3/\text{kg}$$

$$\dot{m}_1 = \frac{\dot{V}_1}{v_1} = \frac{1.25 \text{ m}^3/\text{s}}{0.7599 \text{ m}^3/\text{kg}} = 1.645 \text{ kg/s}$$

$$\dot{m}_3 = 2.6\dot{m}_1 = 2.6(1.645 \text{ kg/s}) = 4.277 \text{ kg/s}$$

The rate of heat gain of the room is determined from

$$\dot{Q}_{\text{cool}} = \dot{m}_3(h_{\text{room}} - h_3) = (4.277 \text{ kg/s})(297.18 - 296.04) \text{ kJ/kg} = \mathbf{4.88 \text{ kW}}$$

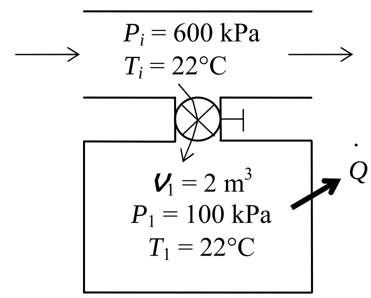
6-124 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until mechanical equilibrium is established. The mass of air that entered and the amount of heat transfer are to be determined.

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-21)

$$T_i = 295 \text{ K} \longrightarrow h_i = 295.17 \text{ kJ/kg}$$
$$T_1 = 295 \text{ K} \longrightarrow u_1 = 210.49 \text{ kJ/kg}$$
$$T_2 = 350 \text{ K} \longrightarrow u_2 = 250.02 \text{ kJ/kg}$$

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:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \to \quad m_i = m_2 - m_1$$

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,

$$m_i = m_2 - m_1 = 11.946 - 2.362 = \mathbf{9.584 \text{ kg}}$$

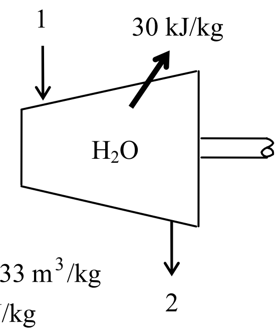
(*b*) The heat transfer during this process is determined from

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(9.584 \text{ kg})(295.17 \text{ kJ/kg}) + (11.946 \text{ kg})(250.02 \text{ kJ/kg}) - (2.362 \text{ kg})(210.49 \text{ kJ/kg}) \\ &= -339 \text{ kJ} \quad \to \quad Q_{\text{out}} = \mathbf{339 \text{ kJ}} \end{aligned}$$

Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reversed the direction.

6-168 CD EES 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)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 550°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.035655 \text{ m}^3/\text{kg} \\ h_1 = 3502.0 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 25 \text{ kPa} \\ x_2 = 0.95 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_f + x_2\boldsymbol{v}_{fg} = 0.00102 + (0.95)(6.2034 - 0.00102) = 5.8933 \text{ m}^3/\text{kg} \\ h_2 = h_f + x_2 h_{fg} = 271.96 + (0.95)(2345.5) = 2500.2 \text{ kJ/kg} \end{array}$$

Analysis (*a*) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} V_1 A_1 = \frac{1}{0.035655 \text{ m}^3/\text{kg}} (60 \text{ m/s})(0.015 \text{ m}^2) = \mathbf{25.24 \text{ kg/s}}$$

(*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

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} V_2 A_2 \longrightarrow V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{(25.24 \text{ kg/s})(5.8933 \text{ m}^3/\text{kg})}{0.14 \text{ m}^2} = \mathbf{1063 \text{ m/s}}$$

(*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

$$\dot{W}_{\text{out}} = -(25.24 \times 30) \text{ kJ/s} - (25.24 \text{ kg/s})\left(2500.2 - 3502.0 + \frac{(1063 \text{ m/s})^2 - (60 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

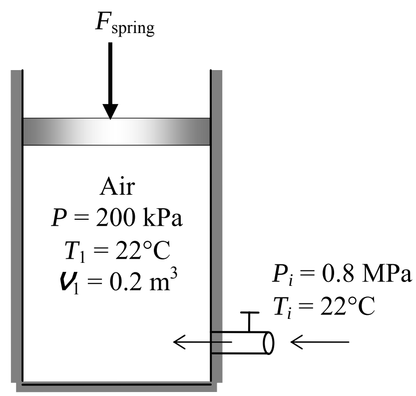
$$= \mathbf{10{,}330 \text{ kW}}$$

6-173 CD EES 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, $u = c_vT$ and $h = c_pT$.

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:

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{b,out} + m_2u_2 - m_1u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

Combining the two relations, $\quad (m_2 - m_1)h_i = W_{b,out} + m_2u_2 - m_1u_1$

or, $\quad (m_2 - m_1)c_pT_i = W_{b,out} + m_2c_vT_2 - m_1c_vT_1$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1V_1}{RT_1} = \frac{(200 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 0.472 \text{ kg}$$

$$m_2 = \frac{P_2V_2}{RT_2} = \frac{(600 \text{ kPa})(0.4 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{836.2}{T_2}$$

Then from the mass balance becomes $\quad m_i = m_2 - m_1 = \dfrac{836.2}{T_2} - 0.472$

The spring is a linear spring, and thus the boundary work for this process can be determined from

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200+600)\text{kPa}}{2}(0.4-0.2)\text{m}^3 = 80 \text{ kJ}$$

Substituting into the energy balance, the final temperature of air T_2 is determined to be

$$-80 = -\left(\frac{836.2}{T_2} - 0.472\right)(1.005)(295) + \left(\frac{836.2}{T_2}\right)(0.718)(T_2) - (0.472)(0.718)(295)$$

It yields $\quad T_2 = \mathbf{344.1\ K}$

Thus, $\quad m_2 = \dfrac{836.2}{T_2} = \dfrac{836.2}{344.1} = 2.430 \text{ kg}$

and $\quad m_i = m_2 - m_1 = 2.430 - 0.472 = \mathbf{1.96\ kg}$

7-88 A geothermal power plant uses geothermal liquid water at 160ºC 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, the source and the sink state enthalpies of geothermal water are (Table A-4)

$$\left.\begin{array}{l} T_{\text{source}} = 160°\text{C} \\ x_{\text{source}} = 0 \end{array}\right\} h_{\text{source}} = 675.47 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{\text{sink}} = 25°\text{C} \\ x_{\text{sink}} = 0 \end{array}\right\} h_{\text{sink}} = 104.83 \text{ kJ/kg}$$

Analysis (*a*) The rate of heat input to the plant may be taken as the enthalpy difference between the source and the sink for the power plant

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{geo}}(h_{\text{source}} - h_{\text{sink}}) = (440 \text{ kg/s})(675.47 - 104.83) \text{ kJ/kg} = 251{,}083 \text{ kW}$$

The actual thermal efficiency is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_{\text{in}}} = \frac{22 \text{ MW}}{251.083 \text{ MW}} = \mathbf{0.0876} = \mathbf{8.8\%}$$

(*b*) The maximum thermal efficiency is the thermal efficiency of a reversible heat engine operating between the source and sink temperatures

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(25+273) \text{ K}}{(160+273) \text{ K}} = \mathbf{0.312} = \mathbf{31.2\%}$$

(*c*) Finally, the rate of heat rejection is

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net,out}} = 251.1 - 22 = \mathbf{229.1 \text{ MW}}$$

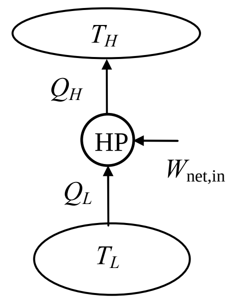
7-131 An expression for the COP of a completely reversible heat pump in terms of the thermal-energy reservoir temperatures, T_L and T_H is to be derived.

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

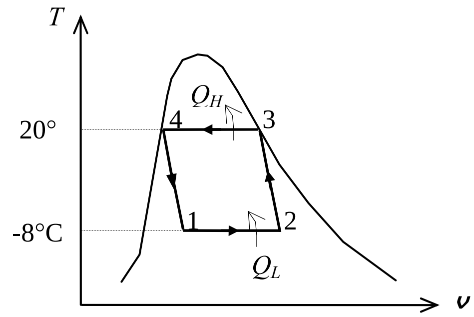
$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L / T_H} = \frac{T_H}{T_H - T_L}$$

7-135 A Carnot refrigeration cycle is executed in a closed system with a fixed mass of R-134a. The net work input and the maximum and minimum temperatures are given. The mass fraction of the refrigerant that vaporizes during the heat addition process, and the pressure at the end of the heat rejection process are to be determined.

Properties The enthalpy of vaporization of R-134a at -8°C is h_{fg} = 204.52 kJ/kg (Table A-12).

Analysis The coefficient of performance of the cycle is

$$\mathrm{COP_R} = \frac{1}{T_H / T_L - 1} = \frac{1}{293/265 - 1} = 9.464$$



and

$$Q_L = \mathrm{COP_R} \times W_{\mathrm{in}} = (9.464)(15\text{ kJ}) = 142\text{ kJ}$$

Then the amount of refrigerant that vaporizes during heat absorption is

$$Q_L = m h_{fg@T_L = -8^\circ\mathrm{C}} \longrightarrow m = \frac{142\text{ kJ}}{204.52\text{ kJ/kg}} = 0.695\text{ kg}$$

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, the fraction of mass that vaporized during heat addition process is

$$\frac{0.695\text{ kg}}{0.8\text{ kg}} = 0.868 \text{ or } \mathbf{86.8\%}$$

The pressure at the end of the heat rejection process is

$$P_4 = P_{\mathrm{sat@20°C}} = \mathbf{572.1 kPa}$$

8-24 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.

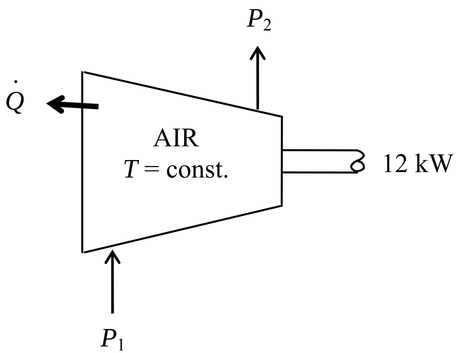
Properties Noting that $h = h(T)$ for ideal gases, we have $h_1 = h_2$ since $T_1 = T_2 = 25°C$.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} = \dot{Q}_{out}$$



Therefore,

$$\dot{Q}_{out} = \dot{W}_{in} = 12 \text{ kW}$$

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

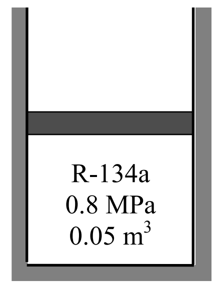
$$\Delta \dot{S}_{air} = -\frac{\dot{Q}_{out,air}}{T_{sys}} = -\frac{12 \text{ kW}}{298 \text{ K}} = \mathbf{-0.0403 \text{ kW/K}}$$

8-44 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 final temperature in the cylinder and the work done by the refrigerant 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 (*a*) 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),

$$\left.\begin{array}{l} P_1 = 0.8 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@0.8 \text{ MPa}} = 0.025621 \text{ m}^3\text{/kg} \\ u_1 = u_{g@0.8 \text{ MPa}} = 246.79 \text{ kJ/kg} \\ s_1 = s_{g@0.8 \text{ MPa}} = 0.91835 \text{ kJ/kg} \cdot \text{K} \end{array}$$



Also,

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.05 \text{ m}^3}{0.025621 \text{ m}^3\text{/kg}} = 1.952 \text{ kg}$$

and

$$\left.\begin{array}{l} P_2 = 0.4 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{0.91835 - 0.24761}{0.67929} = 0.9874 \\ u_2 = u_f + x_2 u_{fg} = 63.62 + (0.9874)(171.45) = 232.91 \text{ kJ/kg} \end{array}$$

$$T_2 = T_{\text{sat@0.4 MPa}} = \mathbf{8.91°C}$$

(*b*) 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

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U$$

$$W_{\text{b,out}} = m(u_1 - u_2)$$

Substituting, the work done during this isentropic process is determined to be

$$W_{\text{b,out}} = m(u_1 - u_2) = (1.952 \text{ kg})(246.79 - 232.91) \text{ kJ/kg} = \mathbf{27.09 \text{ kJ}}$$

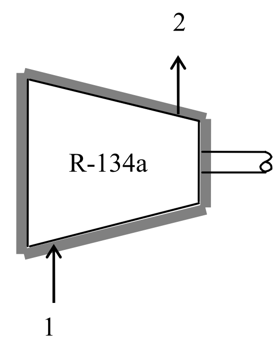
8-46 Saturated Refrigerant-134a vapor at 160 kPa is compressed steadily by an adiabatic compressor. The minimum power input to the compressor is to be determined.

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 The power input to an adiabatic compressor will be a minimum when the compression process is reversible. For the reversible adiabatic process we have $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 160 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@160 \text{ kPa}} = 0.12348 \text{ m}^3/\text{kg} \\ h_1 = h_{g@160 \text{ kPa}} = 241.11 \text{ kJ/kg} \\ s_1 = s_{g@160 \text{ kPa}} = 0.9419 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 277.06 \text{ kJ/kg}$$



Also,

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{2 \text{ m}^3/\text{min}}{0.12348 \text{ m}^3/\text{kg}} = 16.20 \text{ kg/min} = 0.27 \text{ kg/s}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

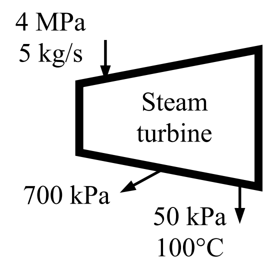
Substituting, the minimum power supplied to the compressor is determined to be

$$\dot{W}_{in} = (0.27 \text{ kg/s})(277.06 - 241.11) \text{ kJ/kg} = \mathbf{9.71 \ kW}$$

8-60 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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

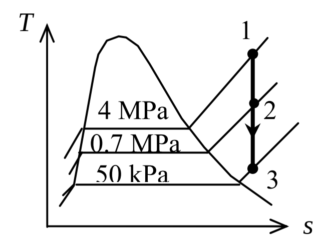
$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{out}$$

$$\dot{W}_{out} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$

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,

$$\begin{aligned} \dot{W}_{out} &= \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3 \\ &= (5 \text{ kg/s})(3979.3 \text{ kJ/kg}) - (0.25 \text{ kg/s})(3309.1 \text{ kJ/kg}) - (4.75 \text{ kg/s})(2682.4 \text{ kJ/kg}) \\ &= \mathbf{6328 \text{ kW}} \end{aligned}$$

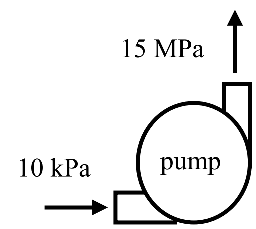
8-73 An adiabatic pump is used to compress saturated liquid water in a reversible manner. The work input is to be determined by different approaches.

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)

$$\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 191.81 \text{ kJ/kg} \\ s_1 = 0.6492 \text{ kJ/kg} \\ v_1 = 0.001010 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 15 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} h_2 = 206.90 \text{ kJ/kg} \\ v_2 = 0.001004 \text{ m}^3\text{/kg} \end{array}$$



(*a*) Using the entropy data from the compressed liquid water table

$$w_P = h_2 - h_1 = 206.90 - 191.81 = \mathbf{15.10\ kJ/kg}$$

(*b*) Using inlet specific volume and pressure values

$$w_P = v_1(P_2 - P_1) = (0.001010 \text{ m}^3\text{/kg})(15{,}000 - 10)\text{kPa} = \mathbf{15.14\ kJ/kg}$$

Error = **0.3%**

(*b*) Using average specific volume and pressure values

$$w_P = v_{avg}(P_2 - P_1) = \left[1/2(0.001010 + 0.001004) \text{ m}^3\text{/kg}\right](15{,}000 - 10)\text{kPa} = \mathbf{15.10\ kJ/kg}$$

Error = **0%**

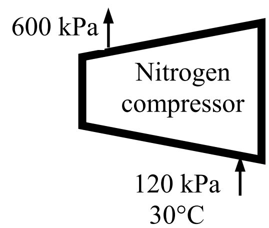
Discussion The results show that any of the method may be used to calculate reversible pump work.

8-99 Nitrogen is compressed in an adiabatic compressor. The minimum work input 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** Nitrogen is an ideal gas with constant specific heats.

Properties The properties of nitrogen at an anticipated average temperature of 400 K are c_p = 1.044 kJ/kg·K and k = 1.397 (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 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



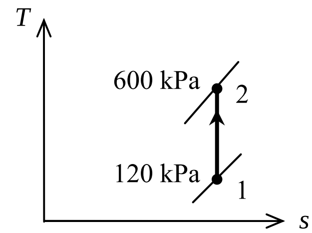
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 + \dot{W}_{in} = \dot{m}h_2$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

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



$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (303\ \text{K}) \left(\frac{600\ \text{kPa}}{120\ \text{kPa}} \right)^{0.397/1.397} = 479\ \text{K}$$

Substituting into the energy balance equation gives

$$w_{in} = h_2 - h_1 = c_p (T_2 - T_1) = (1.044\ \text{kJ/kg}\cdot\text{K})(479 - 303)\text{K} = \mathbf{184\ kJ/kg}$$

8-128 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.

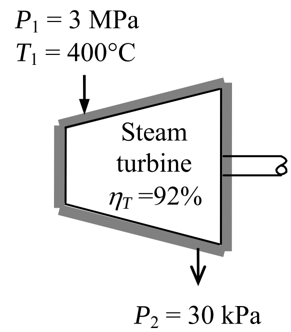
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2)$$



From the steam tables (Tables A-4 through A-6),

$$\left.\begin{aligned} P_1 &= 3 \text{ MPa} \\ T_1 &= 400°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 3231.7 \text{ kJ/kg} \\ s_1 &= 6.9235 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_{2s} &= 30 \text{ kPa} \\ s_{2s} &= s_1 \end{aligned}\right\} \begin{aligned} x_{2s} &= \frac{s_{2s} - s_f}{s_{fg}} = \frac{6.9235 - 0.9441}{6.8234} = 0.8763 \\ h_{2s} &= h_f + x_{2s} h_{fg} = 289.27 + (0.8763)(2335.3) = 2335.7 \text{ kJ/kg} \end{aligned}$$

The actual power output may be determined by multiplying the isentropic power output with the isentropic efficiency. Then,

$$\begin{aligned} \dot{W}_{a,\text{out}} &= \eta_T \dot{W}_{s,\text{out}} \\ &= \eta_T \dot{m}(h_1 - h_{2s}) \\ &= (0.92)(2 \text{ kg/s})(3231.7 - 2335.7)\text{kJ/kg} \\ &= \mathbf{1649\ kW} \end{aligned}$$

8-132 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),

$$\left.\begin{array}{l} P_1 = 7\text{ MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3650.6\text{ kJ/kg} \\ s_1 = 7.0910\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\text{ kPa} \\ T_2 = 150°\text{C} \end{array}\right\} h_{2a} = 2780.2\text{ kJ/kg}$$

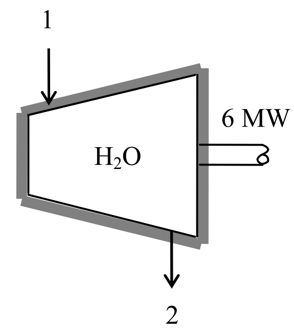
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{a,out} + \dot{m}(h_2 + V_1^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,out} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$



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

$$\dot{W}_{s,out} = -\dot{m}\left(h_{2s} - h_1 + \left\{\left(V_2^2 - V_1^2\right)/2\right\}\right)$$

$$\dot{W}_{s,out} = -(6.95\text{ kg/s})\left(2467.3 - 3650.6 + \frac{(140\text{ m/s})^2 - (80\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right)$$

$$= 8174\text{ kW}$$

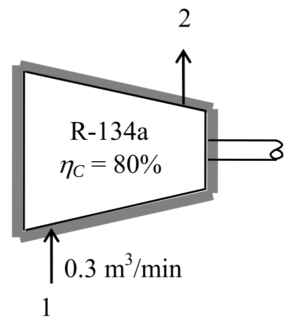
Then the isentropic efficiency of the turbine becomes

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{6000\text{ kW}}{8174\text{ kW}} = 0.734 = \mathbf{73.4\%}$$

8-135 CD EES 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),



$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@120\text{ kPa}} = 236.97\text{ kJ/kg} \\ s_1 = s_{g@120\text{ kPa}} = 0.94779\text{ kJ/kg}\cdot\text{K} \\ v_1 = v_{g@120\text{ kPa}} = 0.16212\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1\text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.21\text{ kJ/kg}$$

From the isentropic efficiency relation,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_C = 236.97 + (281.21 - 236.97)/0.80 = 292.26\text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 1\text{ MPa} \\ h_{2a} = 292.26\text{ kJ/kg} \end{array}\right\} T_{2a} = \mathbf{58.9^\circ C}$$

(b) The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60\text{ m}^3\text{/s}}{0.16212\text{ m}^3\text{/kg}} = 0.0308\text{ kg/s}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{a,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,in} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor becomes,

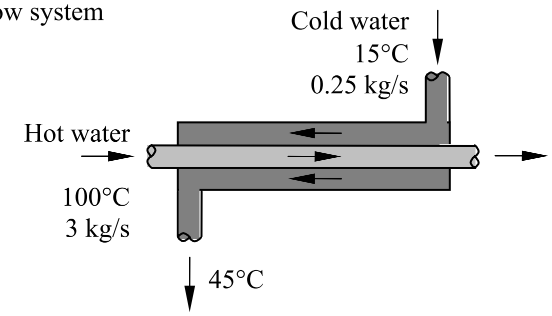
$$\dot{W}_{a,in} = (0.0308\text{ kg/s})(292.26 - 236.97)\text{kJ/kg} = \mathbf{1.70\text{ kW}}$$

8-148 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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q}_{in} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{cold water}} = (0.25 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) = \mathbf{31.35\ kW}$$

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

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 100°\text{C} - \frac{31.35 \text{ kW}}{(3 \text{ kg/s})(4.19 \text{ kJ/kg.°C})} = 97.5°\text{C}$$

(*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:

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{gen}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change of entropy}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{cold} s_1 + \dot{m}_{hot} s_3 - \dot{m}_{cold} s_2 - \dot{m}_{hot} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{cold}(s_2 - s_1) + \dot{m}_{hot}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{cold} c_p \ln\frac{T_2}{T_1} + \dot{m}_{hot} c_p \ln\frac{T_4}{T_3}$$

$$= (0.25 \text{ kg/s})(4.18 \text{ kJ/kg.K})\ln\frac{45+273}{15+273} + (3 \text{ kg/s})(4.19 \text{ kJ/kg.K})\ln\frac{97.5+273}{100+273}$$

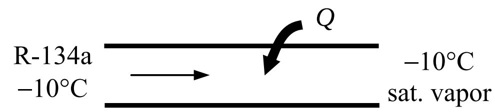
$$= \mathbf{0.0190\ \ kW/K}$$

8-152 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

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{gen}}_{\text{Rate of entropy generation}} = \underbrace{\Delta \dot{S}_{system}}_{\text{Rate of change of entropy}}^{\nearrow 0\text{ (steady)}}$$



$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{in}}{T_w} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_R (s_2 - s_1) - \frac{\dot{Q}_{in}}{T_w}$$

$$\dot{S}_{gen} = \dot{m}_R s_{fg} - \frac{\dot{Q}_{in}}{T_w}$$

The properties of the refrigerant are (Table A-11)

$$h_{fg\,@\,-10°C} = 205.96\text{ kJ/kg}$$
$$s_{fg\,@\,-10°C} = 0.78263\text{ kJ/kg}\cdot\text{K}$$

The rate of that must be removed from the water in order to freeze it at a rate of 4000 kg/h is

$$\dot{Q}_{in} = \dot{m}_w h_{if} = (4000/3600\text{ kg/s})(333.7\text{ kJ/kg}) = 370.8\text{ kW}$$

where the heat of fusion of water at 1 atm is 333.7 kJ/kg. The mass flow rate of R-134a is

$$\dot{m}_R = \frac{\dot{Q}_{in}}{h_{fg}} = \frac{370.8\text{ kJ/s}}{205.96\text{ kJ/kg}} = 1.800\text{ kg/s}$$

Substituting,

$$\dot{S}_{gen} = \dot{m}_R s_{fg} - \frac{\dot{Q}_{in}}{T_w} = (1.800\text{ kg/s})(0.78263\text{ kJ/kg}\cdot\text{K}) - \frac{370.8\text{ kW}}{273\text{ K}} = \mathbf{0.0505\ kW/K}$$

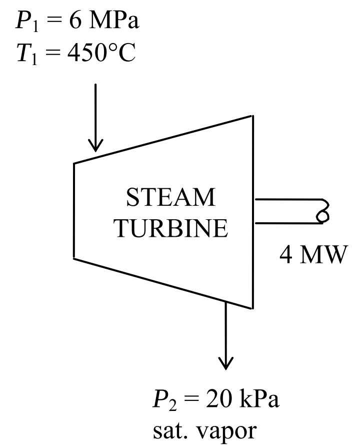
8-166 Steam expands in a turbine from a specified state to another specified state. The rate of entropy generation during this process is to be determined.

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)

$$\left.\begin{array}{l} P_1 = 6 \text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3302.9 \text{ kJ/kg} \\ s_1 = 6.7219 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 20 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_2 = 2608.9 \text{ kJ/kg} \\ s_2 = 7.9073 \text{ kJ/kg}\cdot\text{K} \end{array}$$



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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{W}_{out} + \dot{m}h_2$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2) - \dot{W}_{out}$$

Substituting,

$$\dot{Q}_{out} = (25{,}000/3600 \text{ kg/s})(3302.9 - 2608.9)\text{kJ/kg} - 4000 \text{ kJ/s} = 819.3 \text{ kJ/s}$$

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

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\text{Rate of net entropy transfer by heat and mass}} + \underbrace{\dot{S}_{gen}}_{\text{Rate of entropy generation}} = \underbrace{\Delta\dot{S}_{system}^{\nearrow 0}}_{\text{Rate of change of entropy}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_{b,surr}} + \dot{S}_{gen} = 0$$

Substituting, the rate of entropy generation during this process is determined to be

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{out}}{T_{b,surr}} = (25{,}000/3600 \text{ kg/s})(7.9073 - 6.7219)\text{kJ/kg}\cdot\text{K} + \frac{819.3 \text{ kW}}{298 \text{ K}} = \mathbf{11.0 \text{ kW/K}}$$

8-168 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.

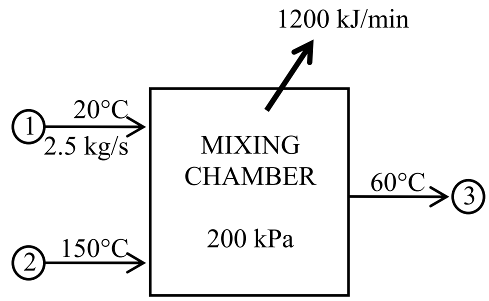
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

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,

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@20°\text{C}} = 83.91 \text{ kJ/kg} \\ s_1 \cong s_{f@20°\text{C}} = 0.2965 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ T_2 = 150°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 2769.1 \text{ kJ/kg} \\ s_2 = 7.2810 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 200 \text{ kPa} \\ T_3 = 60°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@60°\text{C}} = 251.18 \text{ kJ/kg} \\ s_3 \cong s_{f@60°\text{C}} = 0.8313 \text{ kJ/kg}\cdot\text{K} \end{array}$$



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:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{Q}_{\text{out}} + \dot{m}_3 h_3$$

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

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1 (h_1 - h_3)}{h_2 - h_3} = \frac{(1200/60 \text{kJ/s}) - (2.5 \text{ kg/s})(83.91 - 251.18)\text{kJ/kg}}{(2769.1 - 251.18)\text{kJ/kg}} = \mathbf{0.166 \text{ kg/s}}$$

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

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the rate of entropy generation during this process is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} \\ &= (2.666 \text{ kg/s})(0.8313 \text{ kJ/kg}\cdot\text{K}) - (0.166 \text{ kg/s})(7.2810 \text{ kJ/kg}\cdot\text{K}) \\ &\quad - (2.5 \text{ kg/s})(0.2965 \text{ kJ/kg}\cdot\text{K}) + \frac{(1200/60 \text{ kJ/s})}{298 \text{ K}} \\ &= \mathbf{0.333 \text{ kW/K}} \end{aligned}$$

8-189 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),

$$\left.\begin{array}{l} P_1 = 500 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = v_{g@500 \text{ kPa}} = 0.37483 \text{ m}^3/\text{kg} \\ u_1 = u_{g@500 \text{ kPa}} = 2560.7 \text{ kJ/kg} \\ s_1 = s_{g@500 \text{ kPa}} = 6.8207 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_{2,A} = T_{sat@150 \text{ kPa}} = \mathbf{111.35^\circ C}$$

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ s_2 = s_1 \\ (sat.mixture) \end{array}\right\} \begin{array}{l} x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{6.8207 - 1.4337}{5.7894} = 0.9305 \\ \boldsymbol{v}_{2,A} = \boldsymbol{v}_f + x_{2,A}\boldsymbol{v}_{fg} = 0.001053 + (0.9305)(1.1594 - 0.001053) = 1.0789 \text{ m}^3/\text{kg} \end{array}$$

$$u_{2,A} = u_f + x_{2,A}u_{fg} = 466.97 + (0.9305)(2052.3 \text{ kJ/kg}) = 2376.6 \text{ kJ/kg}$$

The initial and the final masses in tank A are

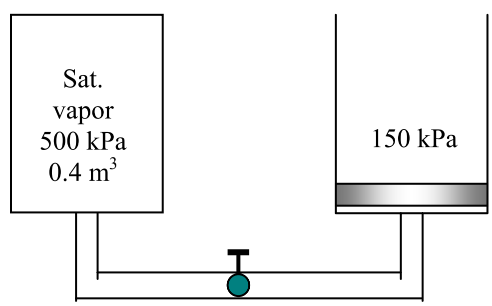
$$m_{1,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{1,A}} = \frac{0.4 \text{ m}^3}{0.37483 \text{ m}^3/\text{kg}} = 1.067 \text{ kg} \quad \text{and} \quad m_{2,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{2,A}} = \frac{0.4 \text{ m}^3}{1.0789 \text{ m}^3/\text{kg}} = 0.371 \text{ kg}$$

Thus,

$$m_{2,B} = m_{1,A} - m_{2,A} = 1.067 - 0.371 = 0.696 \text{ kg}$$

(*b*) The boundary work done during this process is

$$W_{b,out} = \int_1^2 P d\boldsymbol{V} = P_B(\boldsymbol{V}_{2,B} - 0) = P_B m_{2,B} \boldsymbol{v}_{2,B}$$



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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U = (\Delta U)_A + (\Delta U)_B$$

$$W_{\text{b,out}} + (\Delta U)_A + (\Delta U)_B = 0$$

or,

$$P_B m_{2,B} \boldsymbol{v}_{2,B} + (m_2u_2 - m_1u_1)_A + (m_2u_2)_B = 0$$

$$m_{2,B}h_{2,B} + (m_2u_2 - m_1u_1)_A = 0$$

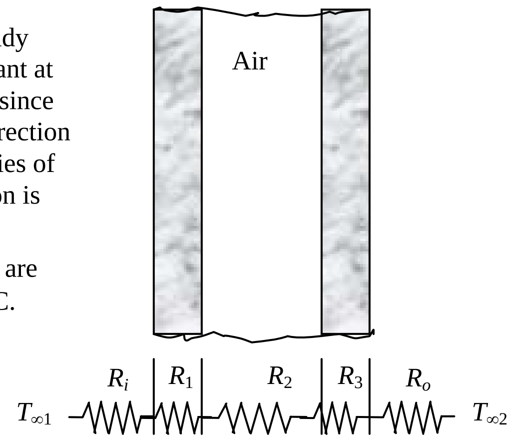
Thus,

$$h_{2,B} = \frac{(m_1u_1 - m_2u_2)_A}{m_{2,B}} = \frac{(1.067)(2560.7) - (0.371)(2376.6)}{0.696} = 2658.8 \text{ kJ/kg}$$

At 150 kPa, $h_f = 467.13$ and $h_g = 2693.1$ kJ/kg. Thus at the final state, the cylinder will contain a saturated liquid-vapor mixture since $h_f < h_2 < h_g$. Therefore,

$$T_{2,B} = T_{\text{sat@150 kPa}} = \mathbf{111.35^\circ C}$$

10-20 A double-pane window consists of two layers of glass separated by a stagnant air space. For specified indoors and outdoors temperatures, the rate of heat loss through the window and the inner surface temperature of the window are to be determined.



Assumptions **1** Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. **2** Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. **3** Thermal conductivities of the glass and air are constant. **4** Heat transfer by radiation is negligible.

Properties The thermal conductivity of the glass and air are given to be k_{glass} = 0.78 W/m·°C and k_{air} = 0.026 W/m·°C.

Analysis The area of the window and the individual resistances are

$$A = (1.2\text{ m}) \times (2\text{ m}) = 2.4\text{ m}^2$$

$$R_i = R_{conv,1} = \frac{1}{h_1 A} = \frac{1}{(10\text{ W/m}^2.\text{°C})(2.4\text{ m}^2)} = 0.0417\text{ °C/W}$$

$$R_1 = R_3 = R_{glass} = \frac{L_1}{k_1 A} = \frac{0.003\text{ m}}{(0.78\text{ W/m.°C})(2.4\text{ m}^2)} = 0.0016\text{ °C/W}$$

$$R_2 = R_{air} = \frac{L_2}{k_2 A} = \frac{0.012\text{ m}}{(0.026\text{ W/m.°C})(2.4\text{ m}^2)} = 0.1923\text{ °C/W}$$

$$R_o = R_{conv,2} = \frac{1}{h_2 A} = \frac{1}{(25\text{ W/m}^2.^\circ\text{C})(2.4\text{ m}^2)} = 0.0167\ ^\circ\text{C/W}$$

$$R_{total} = R_{conv,1} + 2R_1 + R_2 + R_{conv,2} = 0.0417 + 2(0.0016) + 0.1923 + 0.0167$$
$$= 0.2539\text{ °C/W}$$

The steady rate of heat transfer through window glass then becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[24-(-5)]\text{°C}}{0.2539\text{°C/W}} = \mathbf{114\ W}$$

The inner surface temperature of the window glass can be determined from

$$\dot{Q} = \frac{T_{\infty 1} - T_1}{R_{conv,1}} \longrightarrow T_1 = T_{\infty 1} - \dot{Q}R_{conv,1} = 24^\circ\text{C} - (114\text{ W})(0.0417\text{°C/W}) = \mathbf{19.2°C}$$

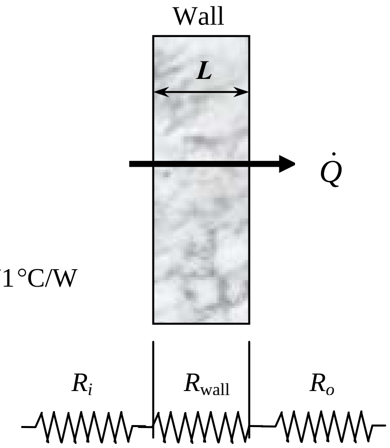
10-35 Two of the walls of a house have no windows while the other two walls have single- or double-pane windows. The average rate of heat transfer through each wall, and the amount of money this household will save per heating season by converting the single pane windows to double pane windows are to be determined.

Assumptions **1** Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. **2** Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. **3** Thermal conductivities of the glass and air are constant. **4** Heat transfer by radiation is disregarded.

Properties The thermal conductivities are given to be k = 0.026 W/m·°C for air, and 0.78 W/m·°C for glass.

Analysis The rate of heat transfer through each wall can be determined by applying thermal resistance network. The convection resistances at the inner and outer surfaces are common in all cases.

Walls without windows:



$$R_i = \frac{1}{h_i A} = \frac{1}{(7\ \text{W/m}^2\cdot{}^\circ\text{C})(10\times 4\ \text{m}^2)} = 0.003571\ {}^\circ\text{C/W}$$

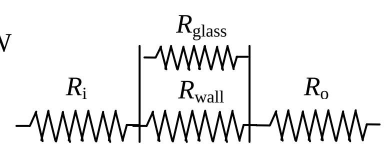
$$R_{\text{wall}} = \frac{L_{\text{wall}}}{kA} = \frac{R-value}{A} = \frac{2.31\ \text{m}^2\cdot{}^\circ\text{C/W}}{(10\times 4\ \text{m}^2)} = 0.05775\ {}^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(18\ \text{W/m}^2\cdot{}^\circ\text{C})(10\times 4\ \text{m}^2)} = 0.001389{}^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{wall}} + R_o = 0.003571 + 0.05775 + 0.001389 = 0.06271\ {}^\circ\text{C/W}$$

Then $\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(24-8){}^\circ\text{C}}{0.06271{}^\circ\text{C/W}} = \mathbf{255.1\ W}$

Wall with single pane windows:



$$R_i = \frac{1}{h_i A} = \frac{1}{(7\ \text{W/m}^2\cdot{}^\circ\text{C})(20\times 4\ \text{m}^2)} = 0.001786\ {}^\circ\text{C/W}$$

$$R_{\text{wall}} = \frac{L_{\text{wall}}}{kA} = \frac{R-value}{A} = \frac{2.31\ \text{m}^2\cdot{}^\circ\text{C/W}}{(20\times 4) - 5(1.2\times 1.8)\ \text{m}^2} = 0.033382\ {}^\circ\text{C/W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{kA} = \frac{0.005\ \text{m}}{(0.78\ \text{W/m}^2\cdot{}^\circ\text{C})(1.2\times 1.8)\text{m}^2} = 0.002968\ {}^\circ\text{C/W}$$

$$\frac{1}{R_{\text{eqv}}} = \frac{1}{R_{\text{wall}}} + 5\frac{1}{R_{\text{glass}}} = \frac{1}{0.033382} + 5\frac{1}{0.002968} \rightarrow R_{eqv} = 0.000583\ {}^\circ\text{C/W}$$

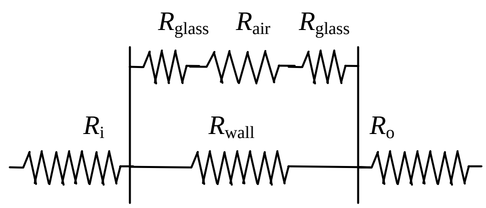
$$R_o = \frac{1}{h_o A} = \frac{1}{(18\ \text{W/m}^2\cdot{}^\circ\text{C})(20\times 4\ \text{m}^2)} = 0.000694\ {}^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{eqv}} + R_o = 0.001786 + 0.000583 + 0.000694 = 0.003063\ {}^\circ\text{C/W}$$

Then

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(24-8){}^\circ\text{C}}{0.003063{}^\circ\text{C/W}} = \mathbf{5224\ W}$$

4th wall with double pane windows:



$$R_{\text{wall}} = \frac{L_{\text{wall}}}{kA} = \frac{R-value}{A} = \frac{2.31\,\text{m}^2\cdot{}^\circ\text{C/W}}{(20\times 4)-5(1.2\times 1.8)\text{m}^2} = 0.033382\,^\circ\text{C/W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{kA} = \frac{0.005\,\text{m}}{(0.78\,\text{W/m}^2\cdot{}^\circ\text{C})(1.2\times 1.8)\text{m}^2} = 0.002968\,^\circ\text{C/W}$$

$$R_{\text{air}} = \frac{L_{\text{air}}}{kA} = \frac{0.015\,\text{m}}{(0.026\,\text{W/m}^2\cdot{}^\circ\text{C})(1.2\times 1.8)\text{m}^2} = 0.267094\,^\circ\text{C/W}$$

$$R_{\text{window}} = 2R_{\text{glass}} + R_{\text{air}} = 2\times 0.002968 + 0.267094 = 0.27303\,^\circ\text{C/W}$$

$$\frac{1}{R_{\text{eqv}}} = \frac{1}{R_{\text{wall}}} + 5\frac{1}{R_{\text{window}}} = \frac{1}{0.033382} + 5\frac{1}{0.27303} \longrightarrow R_{\text{eqv}} = 0.020717\,^\circ\text{C/W}$$

$$R_{\text{total}} = R_i + R_{\text{eqv}} + R_o = 0.001786 + 0.020717 + 0.000694 = 0.023197\,^\circ\text{C/W}$$

Then $\dot{Q} = \dfrac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \dfrac{(24-8)^\circ\text{C}}{0.023197^\circ\text{C/W}} = \mathbf{690\ W}$

The rate of heat transfer which will be saved if the single pane windows are converted to double pane windows is

$$\dot{Q}_{\text{save}} = \dot{Q}_{\substack{\text{single}\\ \text{pane}}} - \dot{Q}_{\substack{\text{double}\\ \text{pane}}} = 5224 - 690 = 4534\ \text{W}$$

The amount of energy and money saved during a 7-month long heating season by switching from single pane to double pane windows become

$$Q_{save} = \dot{Q}_{save}\Delta t = (4.534\ \text{kW})(7\times 30\times 24\ \text{h}) = 22{,}851\ \text{kWh}$$

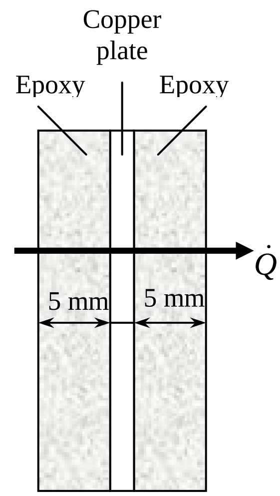
Money savings = (Energy saved)(Unit cost of energy) = (22,851 kWh)($0.08/kWh) = **$1828**

10-49 A thin copper plate is sandwiched between two epoxy boards. The error involved in the total thermal resistance of the plate if the thermal contact conductances are ignored is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is one-dimensional since the plate is large. **3** Thermal conductivities are constant.

Properties The thermal conductivities are given to be k = 386 W/m·°C for copper plates and k = 0.26 W/m·°C for epoxy boards. The contact conductance at the interface of copper-epoxy layers is given to be h_c = 6000 W/m²·°C.

Analysis The thermal resistances of different layers for unit surface area of 1 m² are



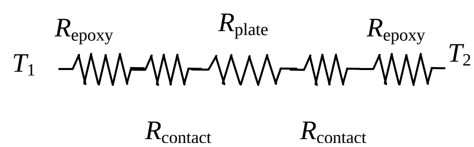
$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(6000\ \text{W/m}^2\cdot{}^\circ\text{C})(1\ \text{m}^2)} = 0.00017\ {}^\circ\text{C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.001\ \text{m}}{(386\ \text{W/m}\cdot{}^\circ\text{C})(1\ \text{m}^2)} = 2.6\times10^{-6}\ {}^\circ\text{C/W}$$

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.005\ \text{m}}{(0.26\ \text{W/m}\cdot{}^\circ\text{C})(1\ \text{m}^2)} = 0.01923\ {}^\circ\text{C/W}$$

The total thermal resistance is

$$\begin{aligned} R_{\text{total}} &= 2R_{\text{contact}} + R_{\text{plate}} + 2R_{\text{epoxy}} \\ &= 2\times0.00017 + 2.6\times10^{-6} + 2\times0.01923 = 0.03880\ {}^\circ\text{C/W} \end{aligned}$$



Then the percent error involved in the total thermal resistance of the plate if the thermal contact resistances are ignored is determined to be

$$\%\text{Error} = \frac{2R_{\text{contact}}}{R_{\text{total}}}\times100 = \frac{2\times0.00017}{0.03880}\times100 = \mathbf{0.88\%}$$

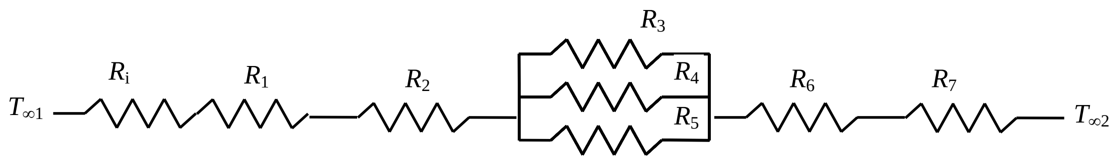
which is negligible.

10-54 A wall consists of horizontal bricks separated by plaster layers. There are also plaster layers on each side of the wall, and a rigid foam on the inner side of the wall. The rate of heat transfer through the wall is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer by radiation is disregarded.

Properties The thermal conductivities are given to be k = 0.72 W/m·°C for bricks, k = 0.22 W/m·°C for plaster layers, and k = 0.026 W/m·°C for the rigid foam.

Analysis We consider 1 m deep and 0.33 m high portion of wall which is representative of the entire wall. The thermal resistance network and individual resistances are



$$R_i = R_{conv,1} = \frac{1}{h_1 A} = \frac{1}{(10\ \text{W/m}^2\cdot°\text{C})(0.33\times 1\ \text{m}^2)} = 0.303\ °\text{C/W}$$

$$R_1 = R_{foam} = \frac{L}{kA} = \frac{0.02\ \text{m}}{(0.026\ \text{W/m}\cdot°\text{C})(0.33\times 1\ \text{m}^2)} = 2.33\ °\text{C/W}$$

$$R_2 = R_6 = R_{\substack{plaster\\side}} = \frac{L}{kA} = \frac{0.02\ \text{m}}{(0.22\ \text{W/m}\cdot°\text{C})(0.33\times 1\ \text{m}^2)} = 0.275\ °\text{C/W}$$

$$R_3 = R_5 = R_{\substack{plaster\\center}} = \frac{L}{h_o A} = \frac{0.18\ \text{m}}{(0.22\ \text{W/m}\cdot°\text{C})(0.015\times 1\ \text{m}^2)} = 54.55\ °\text{C/W}$$

$$R_4 = R_{brick} = \frac{L}{kA} = \frac{0.18\ \text{m}}{(0.72\ \text{W/m}\cdot°\text{C})(0.30\times 1\ \text{m}^2)} = 0.833\ °\text{C/W}$$

$$R_o = R_{conv,2} = \frac{1}{h_2 A} = \frac{1}{(20\ \text{W/m}\cdot°\text{C})(0.33\times 1\ \text{m}^2)} = 0.152\ °\text{C/W}$$

$$\frac{1}{R_{mid}} = \frac{1}{R_3} + \frac{1}{R_4} + \frac{1}{R_5} = \frac{1}{54.55} + \frac{1}{0.833} + \frac{1}{54.55} \longrightarrow R_{mid} = 0.81\ °\text{C/W}$$

$$R_{total} = R_i + R_1 + 2R_2 + R_{mid} + R_o = 0.303 + 2.33 + 2(0.275) + 0.81 + 0.152 = 4.145\ °\text{C/W}$$

The steady rate of heat transfer through the wall per $0.33\ \text{m}^2$ is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[(22-(-4)]°\text{C}}{4.145°\text{C/W}} = 6.27\ \text{W}$$

Then steady rate of heat transfer through the entire wall becomes

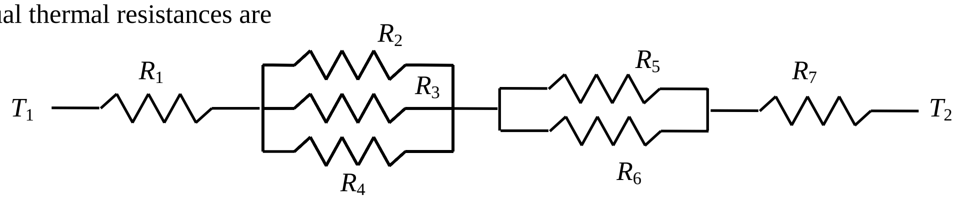
$$\dot{Q}_{total} = (6.27\ \text{W})\frac{(4\times 6)\text{m}^2}{0.33\ \text{m}^2} = \mathbf{456\ W}$$

10-59 A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop across the section F are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Thermal contact resistances at the interfaces are disregarded.

Properties The thermal conductivities are given to be $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, $k_E = 35$ W/m·°C.

Analysis (*a*) The representative surface area is $A = 0.12 \times 1 = 0.12 \text{ m}^2$. The thermal resistance network and the individual thermal resistances are



$$R_1 = R_A = \left(\frac{L}{kA}\right)_A = \frac{0.01 \text{ m}}{(2 \text{ W/m}\cdot°\text{C})(0.12 \text{ m}^2)} = 0.04\ °\text{C/W}$$

$$R_2 = R_4 = R_C = \left(\frac{L}{kA}\right)_C = \frac{0.05 \text{ m}}{(20 \text{ W/m}\cdot°\text{C})(0.04 \text{ m}^2)} = 0.06\ °\text{C/W}$$

$$R_3 = R_B = \left(\frac{L}{kA}\right)_B = \frac{0.05 \text{ m}}{(8 \text{ W/m}\cdot°\text{C})(0.04 \text{ m}^2)} = 0.16\ °\text{C/W}$$

$$R_5 = R_D = \left(\frac{L}{kA}\right)_D = \frac{0.1 \text{ m}}{(15 \text{ W/m}\cdot^{o}\text{C})(0.06 \text{ m}^2)} = 0.11\ °\text{C/W}$$

$$R_6 = R_E = \left(\frac{L}{kA}\right)_E = \frac{0.1 \text{ m}}{(35 \text{ W/m}\cdot°\text{C})(0.06 \text{ m}^2)} = 0.05\ ^{o}\text{C/W}$$

$$R_7 = R_F = \left(\frac{L}{kA}\right)_F = \frac{0.06 \text{ m}}{(2 \text{ W/m}\cdot°\text{C})(0.12 \text{ m}^2)} = 0.25\ °\text{C/W}$$

$$\frac{1}{R_{mid,1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06} \longrightarrow R_{mid,1} = 0.025\ °\text{C/W}$$

$$\frac{1}{R_{mid,2}} = \frac{1}{R_5} + \frac{1}{R_6} = \frac{1}{0.11} + \frac{1}{0.05} \longrightarrow R_{mid,2} = 0.034\ °\text{C/W}$$

$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 = 0.04 + 0.025 + 0.034 + 0.25 = 0.349\ °\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(300 - 100)°\text{C}}{0.349\ °\text{C/W}} = 572 \text{ W} \quad (\text{for a } 0.12 \text{ m} \times 1 \text{ m section})$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (572 \text{ W}) \frac{(5 \text{ m})(8 \text{ m})}{0.12 \text{ m}^2} = \mathbf{1.91 \times 10^5\ W}$$

(*b*) The total thermal resistance between left surface and the point where the sections B, D, and E meet is

$$R_{total} = R_1 + R_{mid,1} = 0.04 + 0.025 = 0.065\ °\text{C/W}$$

Then the temperature at the point where the sections B, D, and E meet becomes

$$\dot{Q} = \frac{T_1 - T}{R_{total}} \longrightarrow T = T_1 - \dot{Q}R_{total} = 300°\text{C} - (572 \text{ W})(0.065\ °\text{C/W}) = \mathbf{263°C}$$

(*c*) The temperature drop across the section F can be determined from

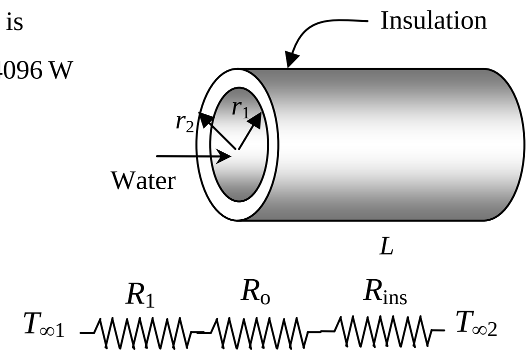
$$\dot{Q} = \frac{\Delta T}{R_F} \rightarrow \Delta T = \dot{Q}R_F = (572 \text{ W})(0.25\ °\text{C/W}) = \mathbf{143°C}$$

10-69 Chilled water is flowing inside a pipe. The thickness of the insulation needed to reduce the temperature rise of water to one-fourth of the original value is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal conductivities are constant. **4** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity is given to be k = 0.05 W/m·°C for insulation.

Analysis The rate of heat transfer without the insulation is



$$\dot{Q}_{old} = \dot{m}c_p\Delta T = (0.98\text{ kg/s})(4180\text{ J/kg}\cdot°\text{C})(8-7)°\text{C} = 4096\text{ W}$$

The total resistance in this case is

$$\dot{Q}_{old} = \frac{T_\infty - T_w}{R_{total}}$$

$$4096\text{ W} = \frac{(30-7.5)°\text{C}}{R_{total}} \longrightarrow R_{total} = 0.005493°\text{C/W}$$

The convection resistance on the outer surface is

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(9\text{ W/m}^2\cdot°\text{C})\pi(0.05\text{ m})(150\text{ m})} = 0.004716\text{ °C/W}$$

The rest of thermal resistances are due to convection resistance on the inner surface and the resistance of the pipe and it is determined from

$$R_1 = R_{total} - R_o = 0.005493 - 0.004716 = 0.0007769\text{ °C/W}$$

The rate of heat transfer with the insulation is

$$\dot{Q}_{new} = \dot{m}c_p\Delta T = (0.98\text{ kg/s})(4180\text{ J/kg}\cdot°\text{C})(0.25°\text{C}) = 1024\text{ W}$$

The total thermal resistance with the insulation is

$$\dot{Q}_{new} = \frac{T_\infty - T_w}{R_{total,new}} \longrightarrow 1024\text{ W} = \frac{[30-(7+7.25)/2)]°\text{C}}{R_{total,new}} \longrightarrow R_{total,new} = 0.02234°\text{C/W}$$

It is expressed by

$$R_{total,new} = R_1 + R_{o,new} + R_{ins} = R_1 + \frac{1}{h_o A_o} + \frac{\ln(D_2/D_1)}{2\pi k_{ins} L}$$

$$0.02234°\text{C/W} = 0.0007769 + \frac{1}{(9\text{ W/m}^2\cdot°\text{C})\pi D_2(150\text{ m})} + \frac{\ln(D_2/0.05)}{2\pi(0.05\text{ W/m}\cdot°\text{C})(150\text{ m})}$$

Solving this equation by trial-error or by using an equation solver such as EES, we obtain

$$D_2 = 0.1265\text{ m}$$

Then the required thickness of the insulation becomes

$$t_{ins} = (D_2 - D_1)/2 = (0.05 - 0.1265)/2 = 0.0382\text{ m} = \mathbf{3.8\text{ cm}}$$

10-71 A spherical container filled with iced water is subjected to convection and radiation heat transfer at its outer surface. The rate of heat transfer and the amount of ice that melts per day are to be determined.

Assumptions **1** Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** Thermal conductivity is constant.

Properties The thermal conductivity of steel is given to be k = 15 W/m·°C. The heat of fusion of water at 1 atm is $h_{if} = 333.7$ kJ/kg. The outer surface of the tank is black and thus its emissivity is ε = 1.

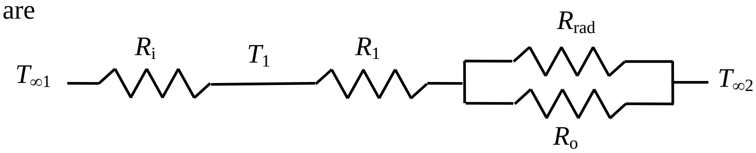
Analysis (a) The inner and the outer surface areas of sphere are

$$A_i = \pi D_i^2 = \pi(8\text{ m})^2 = 201.06\text{ m}^2 \qquad A_o = \pi D_o^2 = \pi(8.03\text{ m})^2 = 202.57\text{ m}^2$$

We assume the outer surface temperature T_2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surfaces of the tank. With this assumption, the radiation heat transfer coefficient can be determined from

$$\begin{aligned} h_{rad} &= \varepsilon\sigma(T_2^{\,2} + T_{surr}^{\,2})(T_2 + T_{surr}) \\ &= 1(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)[(273+5\text{ K})^2 + (273+25\text{ K})^2][(273+25\text{ K})(273+5\text{ K})] \\ &= 5.424\text{ W/m}^2.\text{K} \end{aligned}$$

The individual thermal resistances are



$$R_{conv,i} = \frac{1}{h_i A} = \frac{1}{(80\text{ W/m}^2\cdot°\text{C})(201.06\text{ m}^2)} = 0.000062\text{ °C/W}$$

$$R_1 = R_{sphere} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(4.015-4.0)\text{ m}}{4\pi(15\text{ W/m}\cdot°\text{C})(4.015\text{ m})(4.0\text{ m})} = 0.000005\text{ °C/W}$$

$$R_{conv,o} = \frac{1}{h_o A} = \frac{1}{(10\text{ W/m}^2\cdot°\text{C})(202.57\text{ m}^2)} = 0.000494\text{ °C/W}$$

$$R_{rad} = \frac{1}{h_{rad} A} = \frac{1}{(5.424\text{ W/m}^2\cdot°\text{C})(202.57\text{ m}^2)} = 0.000910\text{ °C/W}$$

$$\frac{1}{R_{eqv}} = \frac{1}{R_{conv,o}} + \frac{1}{R_{rad}} = \frac{1}{0.000494} + \frac{1}{0.000910} \longrightarrow R_{eqv} = 0.000320\text{ °C/W}$$

$$R_{total} = R_{conv,i} + R_1 + R_{eqv} = 0.000062 + 0.000005 + 0.000320 = 0.000387\text{ °C/W}$$

Then the steady rate of heat transfer to the iced water becomes

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{(25-0)°\text{C}}{0.000387\text{ °C/W}} = \mathbf{64{,}600\text{ W}}$$

(b) The total amount of heat transfer during a 24-hour period and the amount of ice that will melt during this period are

$$Q = \dot{Q}\Delta t = (64.600\text{ kJ/s})(24\times3600\text{ s}) = 5.581\times10^6\text{ kJ}$$

$$m_{ice} = \frac{Q}{h_{if}} = \frac{5.581\times10^6\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{16{,}730\text{ kg}}$$

Check: The outer surface temperature of the tank is

$$\dot{Q} = h_{conv+rad} A_o (T_{\infty1} - T_s)$$

$$\to T_s = T_{\infty1} - \frac{\dot{Q}}{h_{conv+rad} A_o} = 25°\text{C} - \frac{64{,}600\text{ W}}{(10+5.424\text{ W/m}^2\cdot°\text{C})(202.57\text{ m}^2)} = 4.3°\text{C}$$

which is very close to the assumed temperature of 5°C for the outer surface temperature used in the evaluation of the radiation heat transfer coefficient. Therefore, there is no need to repeat the calculations.

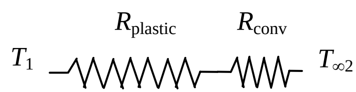
10-92 An electric wire is tightly wrapped with a 1-mm thick plastic cover. The interface temperature and the effect of doubling the thickness of the plastic cover on the interface temperature are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant. **4** The thermal contact resistance at the interface is negligible. **5** Heat transfer coefficient accounts for the radiation effects, if any.

Properties The thermal conductivity of plastic cover is given to be k = 0.15 W/m·°C.

Analysis In steady operation, the rate of heat transfer from the wire is equal to the heat generated within the wire,

$$\dot{Q} = \dot{W}_e = \mathbf{V}I = (8\text{ V})(13\text{ A}) = 104\text{ W}$$



The total thermal resistance is

$$R_{\text{conv}} = \frac{1}{h_o A_o} = \frac{1}{(24\text{ W/m}^2.°\text{C})[\pi(0.0042\text{ m})(10\text{ m})]} = 0.3158\text{ °C/W}$$

$$R_{\text{plastic}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(2.1/1.1)}{2\pi(0.15\text{ W/m.°C})(10\text{ m})} = 0.0686\text{ °C/W}$$

$$R_{\text{total}} = R_{\text{conv}} + R_{\text{plastic}} = 0.3158 + 0.0686 = 0.3844\text{ °C/W}$$

Then the interface temperature becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q}R_{\text{total}} = 30°\text{C} + (104\text{ W})(0.3844\text{ °C/W}) = \mathbf{70.0°C}$$

The critical radius of plastic insulation is

$$r_{cr} = \frac{k}{h} = \frac{0.15\text{ W/m.°C}}{24\text{ W/m}^2.°\text{C}} = 0.00625\text{ m} = 6.25\text{ mm}$$

Doubling the thickness of the plastic cover will increase the outer radius of the wire to 3 mm, which is less than the critical radius of insulation. Therefore, doubling the thickness of plastic cover will increase the rate of heat loss and decrease the interface temperature.

10-126 Two cast iron steam pipes are connected to each other through two 1-cm thick flanges exposed to cold ambient air. The average outer surface temperature of the pipe, the fin efficiency, the rate of heat transfer from the flanges, and the equivalent pipe length of the flange for heat transfer are to be determined.

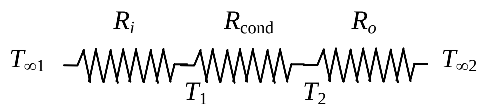
Assumptions **1** Steady operating conditions exist. **2** The temperature along the flanges (fins) varies in one direction only (normal to the pipe). **3** The heat transfer coefficient is constant and uniform over the entire fin surface. **4** The thermal properties of the fins are constant. **5** The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the cast iron is given to be k = 52 W/m·°C.

Analysis (a) We treat the flanges as fins. The individual thermal resistances are

$$A_i = \pi D_i L = \pi(0.092\text{ m})(6\text{ m}) = 1.73\text{ m}^2$$

$$A_o = \pi D_o L = \pi(0.1\text{ m})(6\text{ m}) = 1.88\text{ m}^2$$



$$R_{\text{i}} = \frac{1}{h_i A_i} = \frac{1}{(180\text{ W/m}^2.°\text{C})(1.73\text{ m}^2)} = 0.0032\text{ °C/W}$$

$$R_{\text{cond}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(5/4.6)}{2\pi(52\text{ W/m.°C})(6\text{ m})} = 0.00004\text{ °C/W}$$

$$R_{\text{o}} = \frac{1}{h_o A_o} = \frac{1}{(25\text{ W/m}^2.°\text{C})(1.88\text{ m}^2)} = 0.0213\text{ °C/W}$$

$$R_{\text{total}} = R_{\text{i}} + R_{\text{cond}} + R_{\text{o}} = 0.0032 + 0.00004 + 0.0213 = 0.0245\text{ °C/W}$$

The rate of heat transfer and average outer surface temperature of the pipe are

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(200-12)°\text{C}}{0.0245\text{ °C}} = 7673\text{ W}$$

$$\dot{Q} = \frac{T_2 - T_{\infty 2}}{R_{\text{o}}} \longrightarrow T_2 = T_{\infty 2} + \dot{Q}R_{\text{o}} = 12\text{ °C} + (7673\text{ W})(0.0213\text{ °C/W}) = \mathbf{175.4°C}$$

(b) The fin efficiency can be determined from (Fig. 10-43)

$$\left.\begin{aligned} &\frac{r_2 + \frac{t}{2}}{r_1} = \frac{0.1 + \frac{0.02}{2}}{0.05} = 2.2 \\ &\xi = L_c^{3/2}\left(\frac{h}{kA_p}\right)^{1/2} = \left(L + \frac{t}{2}\right)\sqrt{\frac{h}{kt}} = \left(0.05\text{ m} + \frac{0.02}{2}\text{ m}\right)\sqrt{\frac{25\text{ W/m}^2\,{}^{\text{o}}\text{C}}{(52\text{ W/m}^{\text{o}}\text{C})(0.02\text{ m})}} = 0.29 \end{aligned}\right\}\eta_{\text{fin}} = 0.88$$

$$A_{\text{fin}} = 2\pi(r_2^2 - r_1^2) + 2\pi r_2 t = 2\pi[(0.1\text{ m})^2 - (0.05\text{ m})^2] + 2\pi(0.1\text{ m})(0.02\text{ m}) = 0.0597\text{ m}^2$$

The heat transfer rate from the flanges is

$$\begin{aligned} \dot{Q}_{\text{finned}} &= \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} h A_{\text{fin}}(T_b - T_\infty) \\ &= 0.88(25\text{ W/m}^2.°\text{C})(0.0597\text{ m}^2)(175.4-12)°\text{C} = \mathbf{215\ W} \end{aligned}$$

(c) A 6-m long section of the steam pipe is losing heat at a rate of 7673 W or 7673/6 = 1279 W per m length. Then for heat transfer purposes the flange section is equivalent to

$$\text{Equivalent length} = \frac{215\text{ W}}{1279\text{ W/m}} = 0.168\text{ m} = \mathbf{16.8\ cm}$$

Therefore, the flange acts like a fin and increases the heat transfer by 16.8/2 = 8.4 times.

10-143 A cylindrical tank filled with liquid propane at 1 atm is exposed to convection and radiation. The time it will take for the propane to evaporate completely as a result of the heat gain from the surroundings for the cases of no insulation and 5-cm thick glass wool insulation are to be determined.

Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** The combined heat transfer coefficient is constant and uniform over the entire surface. **4** The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the propane inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

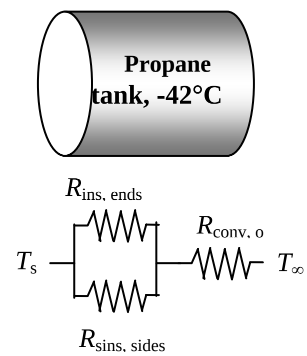
Properties The heat of vaporization and density of liquid propane at 1 atm are given to be 425 kJ/kg and 581 kg/m^3, respectively. The thermal conductivity of glass wool insulation is given to be $k = 0.038$ W/m·°C.

Analysis (*a*) If the tank is not insulated, the heat transfer rate is determined to be

$$A_{\text{tank}} = \pi DL + 2\pi(\pi D^2/4) = \pi(1.2\text{ m})(6\text{ m}) + 2\pi(1.2\text{ m})^2/4 = 24.88\text{ m}^2$$

$$\dot{Q} = hA_{\text{tank}}(T_{\infty 1} - T_{\infty 2}) = (25\text{ W/m}^2.°\text{C})(24.88\text{ m}^2)[30-(-42)]°\text{C} = 44{,}787\text{ W}$$

The volume of the tank and the mass of the propane are



$$V = \pi r^2 L = \pi(0.6\text{ m})^2(6\text{ m}) = 6.786\text{ m}^3$$

$$m = \rho V = (581\text{ kg/m}^3)(6.786\text{ m}^3) = 3942.6\text{ kg}$$

The rate of vaporization of propane is

$$\dot{Q} = \dot{m}h_{fg} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{44.787\text{ kJ/s}}{425\text{ kJ/kg}} = 0.1054\text{ kg/s}$$

Then the time period for the propane tank to empty becomes

$$\Delta t = \frac{m}{\dot{m}} = \frac{3942.6\text{ kg}}{0.1054\text{ kg/s}} = 37{,}413\text{ s} = \mathbf{10.4\text{ hours}}$$

(*b*) We now repeat calculations for the case of insulated tank with 5-cm thick insulation.

$$A_o = \pi DL + 2\pi(\pi D^2/4) = \pi(1.3\text{ m})(6\text{ m}) + 2\pi(1.3\text{ m})^2/4 = 27.16\text{ m}^2$$

$$R_{\text{conv,o}} = \frac{1}{h_o A_o} = \frac{1}{(25\text{ W/m}^2.°\text{C})(27.16\text{ m}^2)} = 0.001473\text{ °C/W}$$

$$R_{\text{insulation,side}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(65/60)}{2\pi(0.038\text{ W/m.°C})(6\text{ m})} = 0.05587\text{ °C/W}$$

$$R_{\text{insulation,ends}} = 2\frac{L}{kA_{avg}} = \frac{2\times 0.05\text{ m}}{(0.038\text{ W/m.°C})[\pi(1.25\text{ m})^2/4]} = 2.1444\text{ °C/W}$$

Noting that the insulation on the side surface and the end surfaces are in parallel, the equivalent resistance for the insulation is determined to be

$$R_{\text{insulation}} = \left(\frac{1}{R_{\text{insulation,side}}} + \frac{1}{R_{\text{insulation,ends}}}\right)^{-1} = \left(\frac{1}{0.05587\text{ °C/W}} + \frac{1}{2.1444\text{ °C/W}}\right)^{-1} = 0.05445\text{ °C/W}$$

Then the total thermal resistance and the heat transfer rate become

$$R_{\text{total}} = R_{\text{conv,o}} + R_{\text{insulation}} = 0.001473 + 0.05445 = 0.05592\text{ °C/W}$$

$$\dot{Q} = \frac{T_\infty - T_s}{R_{\text{total}}} = \frac{[30-(-42)]°\text{C}}{0.05592\text{ °C/W}} = 1288\text{ W}$$

Then the time period for the propane tank to empty becomes

$$\dot{Q} = \dot{m}h_{fg} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{1.288\text{ kJ/s}}{425\text{ kJ/kg}} = 0.003031\text{ kg/s}$$

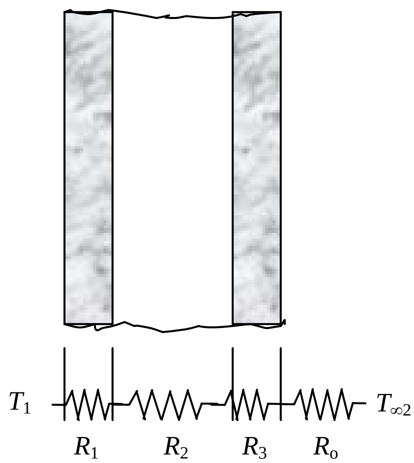
$$\Delta t = \frac{m}{\dot{m}} = \frac{3942.6\text{ kg}}{0.003031\text{ kg/s}} = 1.301\times 10^6\text{ s} = 361.4\text{ hours} = \mathbf{15.1\text{ days}}$$

10-157 A wall constructed of three layers is considered. The rate of hat transfer through the wall and temperature drops across the plaster, brick, covering, and surface-ambient air are to be determined.

Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer by radiation is accounted for in the heat transfer coefficient.

Properties The thermal conductivities of the plaster, brick, and covering are given to be k = 0.72 W/m·°C, k = 0.36 W/m·°C, k = 1.40 W/m·°C, respectively.

Analysis The surface area of the wall and the individual resistances are



$$A = (6\text{ m})\times(2.8\text{ m}) = 16.8\text{ m}^2$$

$$R_1 = R_{\text{plaster}} = \frac{L_1}{k_1 A} = \frac{0.01\text{ m}}{(0.36\text{ W/m.°C})(16.8\text{ m}^2)} = 0.00165\text{ °C/W}$$

$$R_2 = R_{\text{brick}} = \frac{L_2}{k_2 A} = \frac{0.20\text{ m}}{(0.72\text{ W/m.°C})(16.8\text{ m}^2)} = 0.01653\text{ °C/W}$$

$$R_3 = R_{\text{covering}} = \frac{L_3}{k_3 A} = \frac{0.02\text{ m}}{(1.4\text{ W/m.°C})(16.8\text{ m}^2)} = 0.00085\text{ °C/W}$$

$$R_o = R_{\text{conv,2}} = \frac{1}{h_2 A} = \frac{1}{(17\text{ W/m}^2\text{.°C})(16.8\text{ m}^2)} = 0.00350\text{°C/W}$$

$$\begin{aligned} R_{\text{total}} &= R_1 + R_2 + R_3 + R_{\text{conv,2}} \\ &= 0.00165 + 0.01653 + 0.00085 + 0.00350 = 0.02253\text{ °C/W} \end{aligned}$$

The steady rate of heat transfer through the wall then becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(23-8)\text{°C}}{0.02253\text{°C/W}} = \mathbf{665.8\text{ W}}$$

The temperature drops are

$$\Delta T_{\text{plaster}} = \dot{Q}R_{\text{plaster}} = (665.8\text{ W})(0.00165\text{°C/W}) = \mathbf{1.1\text{ °C}}$$

$$\Delta T_{\text{brick}} = \dot{Q}R_{\text{brick}} = (665.8\text{ W})(0.01653\text{°C/W}) = \mathbf{11.0\text{ °C}}$$

$$\Delta T_{\text{covering}} = \dot{Q}R_{\text{covering}} = (665.8\text{ W})(0.00085\text{°C/W}) = \mathbf{0.6\text{ °C}}$$

$$\Delta T_{\text{conv}} = \dot{Q}R_{\text{conv}} = (665.8\text{ W})(0.00350\text{°C/W}) = \mathbf{2.3\text{ °C}}$$

10-162 A 6-m-diameter spherical tank filled with liquefied natural gas (LNG) at -160°C is exposed to ambient air. The time for the LNG temperature to rise to -150°C is to be determined.

Assumptions **1** Heat transfer can be considered to be steady since the specified thermal conditions at the boundaries do not change with time significantly. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** Radiation is accounted for in the combined heat transfer coefficient. **3** The combined heat transfer coefficient is constant and uniform over the entire surface. **4** The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the LNG inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

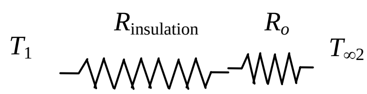
Properties The density and specific heat of LNG are given to be 425 kg/m^3 and 3.475 kJ/kg·°C, respectively. The thermal conductivity of super insulation is given to be k = 0.00008 W/m·°C.

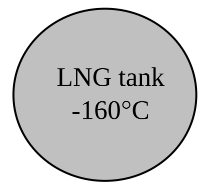
Analysis The inner and outer surface areas of the insulated tank and the volume of the LNG are

$$A_1 = \pi D_1^2 = \pi(4\text{ m})^2 = 50.27\text{ m}^2$$

$$A_2 = \pi D_2^2 = \pi(4.10\text{ m})^2 = 52.81\text{ m}^2$$

$$V_1 = \pi D_1^3/6 = \pi(4\text{ m})^3/6 = 33.51\text{ m}^3$$





The rate of heat transfer to the LNG is

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(2.05 - 2.0)\text{ m}}{4\pi(0.00008\text{ W/m.°C})(2.0\text{ m})(2.05\text{ m})} = 12.13071\text{ °C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(22\text{ W/m}^2\text{.°C})(52.81\text{ m}^2)} = 0.00086\text{ °C/W}$$

$$R_{total} = R_o + R_{insulation} = 0.00086 + 12.13071 = 12.13157\text{ °C/W}$$

$$\dot{Q} = \frac{T_{\infty 2} - T_{\text{LNG}}}{R_{\text{total}}} = \frac{[24 - (-155)]\text{°C}}{12.13157\text{ °C/W}} = 14.75\text{ W}$$

We used average LNG temperature in heat transfer rate calculation. The amount of heat transfer to increase the LNG temperature from -160°C to -150°C is

$$m = \rho V_1 = (425\text{ kg/m}^3)(33.51\text{ m}^3) = 14{,}242\text{ kg}$$

$$Q = mc_p\Delta T = (14{,}242\text{ kg})(3.475\text{ kJ/kg.°C})\left[(-150) - (-160)\text{°C}\right] = 4.95\times10^5\text{ kJ}$$

Assuming that heat will be lost from the LNG at an average rate of 15.17 W, the time period for the LNG temperature to rise to -150°C becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{4.95\times10^5\text{ kJ}}{0.01475\text{ kW}} = 3.355\times10^7\text{ s} = 9320\text{ h} = \mathbf{388\ days}$$

11-14 The temperature of a gas stream is to be measured by a thermocouple. The time it takes to register 99 percent of the initial ΔT is to be determined.

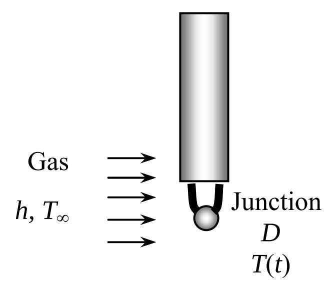
Assumptions **1** The junction is spherical in shape with a diameter of D = 0.0012 m. **2** The thermal properties of the junction are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** Radiation effects are negligible. **5** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified).

Properties The properties of the junction are given to be $k = 35\ \text{W/m.°C}$, $\rho = 8500\ \text{kg/m}^3$, and $c_p = 320\ \text{J/kg.°C}$.

Analysis The characteristic length of the junction and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{\pi D^3/6}{\pi D^2} = \frac{D}{6} = \frac{0.0012\ \text{m}}{6} = 0.0002\ \text{m}$$

$$Bi = \frac{hL_c}{k} = \frac{(90\ \text{W/m}^2.°\text{C})(0.0002\ \text{m})}{(35\ \text{W/m.°C})} = 0.00051 < 0.1$$



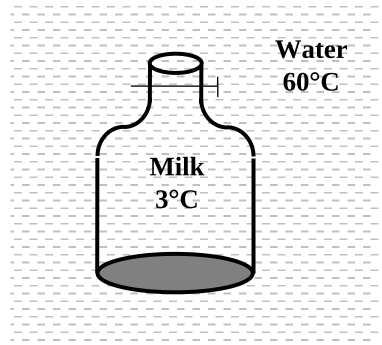
Since $Bi < 0.1$, the lumped system analysis is applicable. Then the time period for the thermocouple to read 99% of the initial temperature difference is determined from

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = 0.01$$

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{90\ \text{W/m}^2.°\text{C}}{(8500\ \text{kg/m}^3)(320\ \text{J/kg.°C})(0.0002\ \text{m})} = 0.1654\ \text{s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow 0.01 = e^{-(0.1654\ \text{s}^{-1})t} \longrightarrow t = \mathbf{27.8\ s}$$

11-17 Milk in a thin-walled glass container is to be warmed up by placing it into a large pan filled with hot water. The warming time of the milk is to be determined.



Assumptions **1** The glass container is cylindrical in shape with a radius of r_0 = 3 cm. **2** The thermal properties of the milk are taken to be the same as those of water. **3** Thermal properties of the milk are constant at room temperature. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Biot number in this case is large (much larger than 0.1). However, the lumped system analysis is still applicable since the milk is stirred constantly, so that its temperature remains uniform at all times.

Properties The thermal conductivity, density, and specific heat of the milk at 20°C are k = 0.598 W/m.°C, ρ = 998 kg/m^3, and c_p = 4.182 kJ/kg.°C (Table A-15).

Analysis The characteristic length and Biot number for the glass of milk are

$$L_c = \frac{V}{A_s} = \frac{\pi r_o^2 L}{2\pi r_o L + 2\pi r_o^2} = \frac{\pi (0.03\text{ m})^2 (0.07\text{ m})}{2\pi (0.03\text{ m})(0.07\text{ m}) + 2\pi (0.03\text{ m})^2} = 0.01050\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(120\text{ W/m}^2.°\text{C})(0.0105\text{ m})}{(0.598\text{ W/m.°C})} = 2.107 > 0.1$$

For the reason explained above we can use the lumped system analysis to determine how long it will take for the milk to warm up to 38°C:

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{120\text{ W/m}^2.°\text{C}}{(998\text{ kg/m}^3)(4182\text{ J/kg.°C})(0.0105\text{ m})} = 0.002738\text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{38 - 60}{3 - 60} = e^{-(0.002738\text{ s}^{-1})t} \longrightarrow t = \mathbf{348\text{ s} = 5.8\text{ min}}$$

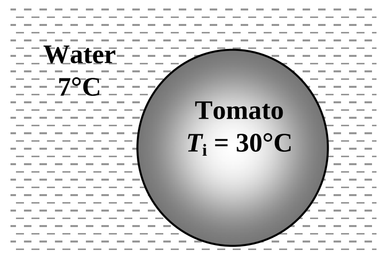
Therefore, it will take about 6 minutes to warm the milk from 3 to 38°C.

11-36 Tomatoes are placed into cold water to cool them. The heat transfer coefficient and the amount of heat transfer are to be determined.

Assumptions **1** The tomatoes are spherical in shape. **2** Heat conduction in the tomatoes is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the tomatoes are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the tomatoes are given to be k = 0.59 W/m.°C, $\alpha = 0.141\times10^{-6}$ m^2/s, ρ = 999 kg/m^3 and c_p = 3.99 kJ/kg.°C.

Analysis The Fourier number is



$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.141\times10^{-6}\text{ m}^2/\text{s})(2\times3600\text{ s})}{(0.04\text{ m})^2} = 0.635$$

which is greater than 0.2. Therefore one-term solution is applicable. The ratio of the dimensionless temperatures at the surface and center of the tomatoes are

$$\frac{\theta_{s,\text{sph}}}{\theta_{0,\text{sph}}} = \frac{\dfrac{T_s - T_\infty}{T_i - T_\infty}}{\dfrac{T_0 - T_\infty}{T_i - T_\infty}} = \frac{T_s - T_\infty}{T_0 - T_\infty} = \frac{A_1 e^{-\lambda_1^2\tau}\dfrac{\sin(\lambda_1)}{\lambda_1}}{A_1 e^{-\lambda_1^2\tau}} = \frac{\sin(\lambda_1)}{\lambda_1}$$

Substituting,

$$\frac{7.1-7}{10-7} = \frac{\sin(\lambda_1)}{\lambda_1} \longrightarrow \lambda_1 = 3.0401$$

From Table 11-2, the corresponding Biot number and the heat transfer coefficient are

$$\text{Bi} = 31.1$$

$$\text{Bi} = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.59\text{ W/m.°C})(31.1)}{(0.04\text{ m})} = \mathbf{459\ W/m^2.°C}$$

The maximum amount of heat transfer is

$$m = 8\rho V = 8\rho\pi D^3/6 = 8(999\text{ kg/m}^3)[\pi(0.08\text{ m})^3/6] = 2.143\text{ kg}$$
$$Q_{\max} = mc_p[T_i - T_\infty] = (2.143\text{ kg})(3.99\text{ kJ/kg.°C})(30-7)\text{°C} = 196.6\text{ kJ}$$

Then the actual amount of heat transfer becomes

$$\left(\frac{Q}{Q_{\max}}\right)_{\text{cyl}} = 1 - 3\left(\frac{T_0 - T_\infty}{T_i - T_\infty}\right)\frac{\sin\lambda_1 - \lambda_1\cos\lambda_1}{\lambda_1^3} = 1 - 3\left(\frac{10-7}{30-7}\right)\frac{\sin(3.0401) - (3.0401)\cos(3.0401)}{(3.0401)^3} = 0.9565$$

$$Q = 0.9565 Q_{\max}$$
$$Q = 0.9565(196.6\text{ kJ}) = \mathbf{188\ kJ}$$

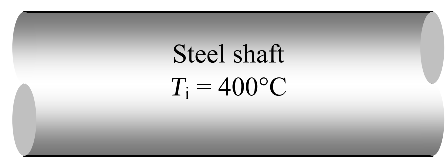
11-41 A long cylindrical shaft at 400°C is allowed to cool slowly. The center temperature and the heat transfer per unit length of the cylinder are to be determined.

Assumptions **1** Heat conduction in the shaft is one-dimensional since it is long and it has thermal symmetry about the center line. **2** The thermal properties of the shaft are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of stainless steel 304 at room temperature are given to be k = 14.9 W/m.°C, ρ = 7900 kg/m^3, c_p = 477 J/kg.°C, $\alpha = 3.95\times10^{-6}$ m^2/s

Analysis First the Biot number is calculated to be

$$Bi = \frac{hr_o}{k} = \frac{(60\ \text{W/m}^2.°\text{C})(0.175\ \text{m})}{(14.9\ \text{W/m.°C})} = 0.705$$



The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.0904 \quad \text{and} \quad A_1 = 1.1548$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(3.95\times10^{-6}\ \text{m}^2/\text{s})(20\times60\ \text{s})}{(0.175\ \text{m})^2} = 0.1548$$

which is very close to the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the temperature at the center of the shaft becomes

$$\theta_{0,cyl} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} = (1.1548)e^{-(1.0904)^2(0.1548)} = 0.9607$$

$$\frac{T_0 - 150}{400-150} = 0.9607 \longrightarrow T_0 = \mathbf{390\ °C}$$

The maximum heat can be transferred from the cylinder per meter of its length is

$$m = \rho \boldsymbol{V} = \rho\pi r_o^2 L = (7900\ \text{kg/m}^3)[\pi(0.175\ \text{m})^2(1\ \text{m})] = 760.1\ \text{kg}$$

$$Q_{\max} = mc_p[T_\infty - T_i] = (760.1\ \text{kg})(0.477\ \text{kJ/kg.°C})(400-150)°\text{C} = 90{,}640\ \text{kJ}$$

Once the constant J_1= 0.4679 is determined from Table 11-3 corresponding to the constant λ_1=1.0904, the actual heat transfer becomes

$$\left(\frac{Q}{Q_{\max}}\right)_{cyl} = 1 - 2\left(\frac{T_o - T_\infty}{T_i - T_\infty}\right)\frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2\left(\frac{390-150}{400-150}\right)\frac{0.4679}{1.0904} = 0.1761$$

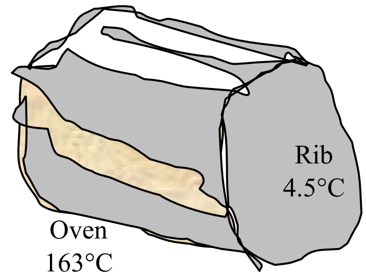
$$Q = 0.1761(90{,}640\ \text{kJ}) = \mathbf{15{,}960\ kJ}$$

11-46 A rib is roasted in an oven. The heat transfer coefficient at the surface of the rib, the temperature of the outer surface of the rib and the amount of heat transfer when it is rare done are to be determined. The time it will take to roast this rib to medium level is also to be determined.

Assumptions **1** The rib is a homogeneous spherical object. **2** Heat conduction in the rib is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the rib are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the rib are given to be $k = 0.45$ W/m.°C, $\rho = 1200$ kg/m^3, $c_p = 4.1$ kJ/kg.°C, and $\alpha = 0.91\times10^{-7}$ m^2/s.

Analysis (*a*) The radius of the roast is determined to be



$$m = \rho V \longrightarrow V = \frac{m}{\rho} = \frac{3.2\text{ kg}}{1200\text{ kg/m}^3} = 0.002667\text{ m}^3$$

$$V = \frac{4}{3}\pi r_o^3 \longrightarrow r_o = \sqrt[3]{\frac{3V}{4\pi}} = \sqrt[3]{\frac{3(0.002667\text{ m}^3)}{4\pi}} = 0.08603\text{ m}$$

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.91\times10^{-7}\text{ m}^2\text{/s})(2\times3600+45\times60)\text{s}}{(0.08603\text{ m})^2} = 0.1217$$

which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the one-term solution can be written in the form

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \longrightarrow \frac{60-163}{4.5-163} = 0.65 = A_1 e^{-\lambda_1^2(0.1217)}$$

It is determined from Table 11-2 by trial and error that this equation is satisfied when $Bi = 30$, which corresponds to $\lambda_1 = 3.0372$ and $A_1 = 1.9898$. Then the heat transfer coefficient can be determined from

$$Bi = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.45\text{ W/m.°C})(30)}{(0.08603\text{ m})} = \mathbf{156.9\ W/m^2.°C}$$

This value seems to be larger than expected for problems of this kind. This is probably due to the Fourier number being less than 0.2.

(*b*) The temperature at the surface of the rib is

$$\theta(r_o,t)_{sph} = \frac{T(r_o,t)-T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o} = (1.9898)e^{-(3.0372)^2(0.1217)}\frac{\sin(3.0372\text{ rad})}{3.0372}$$

$$\frac{T(r_o,t)-163}{4.5-163} = 0.0222 \longrightarrow T(r_o,t) = \mathbf{159.5\ °C}$$

(*c*) The maximum possible heat transfer is

$$Q_{\max} = mc_p(T_\infty - T_i) = (3.2\text{ kg})(4.1\text{ kJ/kg.°C})(163-4.5)\text{°C} = 2080\text{ kJ}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{o,sph}\frac{\sin(\lambda_1)-\lambda_1\cos(\lambda_1)}{\lambda_1^3} = 1-3(0.65)\frac{\sin(3.0372)-(3.0372)\cos(3.0372)}{(3.0372)^3} = 0.783$$

$$Q = 0.783Q_{\max} = (0.783)(2080\text{ kJ}) = \mathbf{1629\ kJ}$$

(*d*) The cooking time for medium-done rib is determined to be

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \longrightarrow \frac{71-163}{4.5-163} = (1.9898)e^{-(3.0372)^2\tau} \longrightarrow \tau = 0.1336$$

$$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.1336)(0.08603\text{ m})^2}{(0.91\times10^{-7}\text{ m}^2\text{/s})} = 10{,}866\text{ s} = 181\text{ min} \cong \mathbf{3\ hr}$$

This result is close to the listed value of 3 hours and 20 minutes. The difference between the two results is due to the Fourier number being less than 0.2 and thus the error in the one-term approximation.

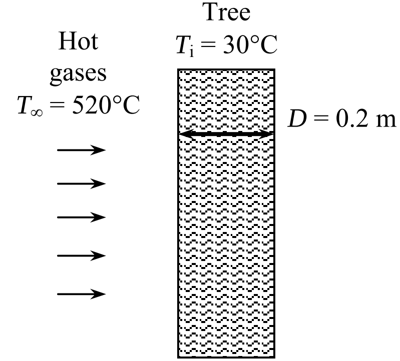
Discussion The temperature of the outer parts of the rib is greater than that of the inner parts of the rib after it is taken out of the oven. Therefore, there will be a heat transfer from outer parts of the rib to the inner parts as a result of this temperature difference. The recommendation is logical.

11-97 CD EES The trunks of some dry oak trees are exposed to hot gases. The time for the ignition of the trunks is to be determined.

Assumptions **1** Heat conduction in the trunks is one-dimensional since it is long and it has thermal symmetry about the center line. **2** The thermal properties of the trunks are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the trunks are given to be $k = 0.17$ W/m.°C and $\alpha = 1.28\times10^{-7}$ m^2/s.

Analysis We treat the trunks of the trees as an infinite cylinder since heat transfer is primarily in the radial direction. Then the Biot number becomes



$$Bi = \frac{hr_o}{k} = \frac{(65\ \text{W/m}^2.°\text{C})(0.1\ \text{m})}{(0.17\ \text{W/m.°C})} = 38.24$$

The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 2.3420 \quad \text{and} \quad A_1 = 1.5989$$

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(1.28\times10^{-7}\ \text{m}^2/\text{s})(4\ \text{h}\times3600\ \text{s/h})}{(0.1\ \text{m})^2} = 0.184$$

which is slightly below 0.2 but close to it. Therefore, assuming the one-term approximate solution for transient heat conduction to be applicable, the temperature at the surface of the trees in 4 h becomes

$$\theta(r_o,t)_{cyl} = \frac{T(r_o,t)-T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} J_0(\lambda_1 r/r_o)$$

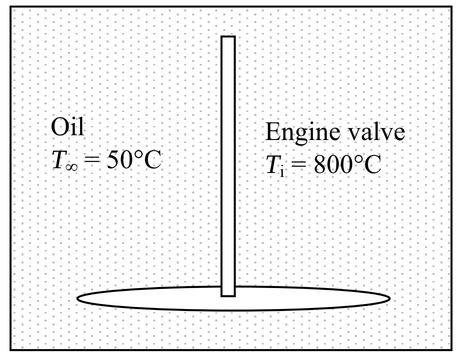
$$\frac{T(r_o,t)-520}{30-520} = (1.5989)e^{-(2.3420)^2(0.184)}(0.0332) = 0.01935 \longrightarrow T(r_o,t) = \mathbf{511\ °C} \ > \ 410°\text{C}$$

Therefore, the trees will ignite. (Note: J_0 is read from Table 11-3).

11-104 Internal combustion engine valves are quenched in a large oil bath. The time it takes for the valve temperature to drop to specified temperatures and the maximum heat transfer are to be determined.

Assumptions **1** The thermal properties of the valves are constant. **2** The heat transfer coefficient is constant and uniform over the entire surface. **3** Depending on the size of the oil bath, the oil bath temperature will increase during quenching. However, an average canstant temperature as specified in the problem will be used. **4** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified).

Properties The thermal conductivity, density, and specific heat of the balls are given to be $k = 48$ W/m.°C, $\rho = 7840$ kg/m^3, and $c_p = 440$ J/kg.°C.



Analysis (*a*) The characteristic length of the balls and the Biot number are

$$L_c = \frac{V}{A_s} = \frac{1.8(\pi D^2 L/4)}{2\pi DL} = \frac{1.8D}{8} = \frac{1.8(0.008\text{ m})}{8} = 0.0018\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(800\text{ W/m}^2.°\text{C})(0.0018\text{ m})}{48\text{ W/m.°C}} = 0.03 < 0.1$$

Therefore, we can use lumped system analysis. Then the time for a final valve temperature of 400°C becomes

$$b = \frac{hA_s}{\rho c_p V} = \frac{8h}{1.8\rho c_p D} = \frac{8(800\text{ W/m}^2.°\text{C})}{1.8(7840\text{ kg/m}^3)(440\text{ J/kg.°C})(0.008\text{ m})} = 0.1288\text{ s}^{-1}$$

$$\frac{T(t)-T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{400-50}{800-50} = e^{-(0.1288\text{ s}^{-1})t} \longrightarrow t = \mathbf{5.9\text{ s}}$$

(*b*) The time for a final valve temperature of 200°C is

$$\frac{T(t)-T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{200-50}{800-50} = e^{-(0.1288\text{ s}^{-1})t} \longrightarrow t = \mathbf{12.5\text{ s}}$$

(*c*) The time for a final valve temperature of 51°C is

$$\frac{T(t)-T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{51-50}{800-50} = e^{-(0.1288\text{ s}^{-1})t} \longrightarrow t = \mathbf{51.4\text{ s}}$$

(*d*) The maximum amount of heat transfer from a single valve is determined from

$$m = \rho V = \rho\frac{1.8\pi D^2 L}{4} = (7840\text{ kg/m}^3)\frac{1.8\pi(0.008\text{ m})^2(0.10\text{ m})}{4} = 0.0709\text{ kg}$$

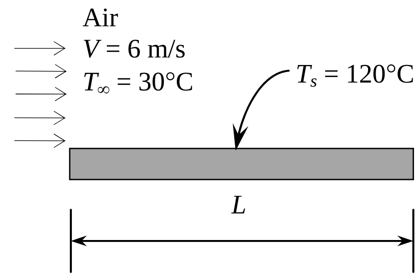
$$Q = mc_p[T_f - T_i] = (0.0709\text{ kg})(440\text{ J/kg.°C})(800-50)°\text{C} = 23{,}400\text{ J} = \mathbf{23.4\text{ kJ}}\ \text{(per valve)}$$

12-40 The top surface of a hot block is to be cooled by forced air. The rate of heat transfer is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

Properties The atmospheric pressure in atm is

$$P = (83.4\,\text{kPa})\frac{1\,\text{atm}}{101.325\,\text{kPa}} = 0.823\,\text{atm}$$



For an ideal gas, the thermal conductivity and the Prandtl number are independent of pressure, but the kinematic viscosity is inversely proportional to the pressure. With these considerations, the properties of air at 0.823 atm and at the film temperature of (120+30)/2=75°C are (Table A-22)

$$k = 0.02917\text{ W/m.°C}$$
$$\nu = \nu_{@1atm} / P_{atm} = (2.046\times10^{-5}\text{ m}^2\text{/s}) / 0.823 = 2.486\times10^{-5}\text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7166$$

Analysis (*a*) If the air flows parallel to the 8 m side, the Reynolds number in this case becomes

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(6\text{ m/s})(8\text{ m})}{2.486\times10^{-5}\text{ m}^2\text{/s}} = 1.931\times10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{0.8} - 871)\,\text{Pr}^{1/3} = [0.037(1.931\times10^6)^{0.8} - 871](0.7166)^{1/3} = 2757$$
$$h = \frac{k}{L}Nu = \frac{0.02917\text{ W/m.°C}}{8\text{ m}}(2757) = 10.05\text{ W/m}^2\text{.°C}$$
$$A_s = wL = (2.5\text{ m})(8\text{ m}) = 20\text{ m}^2$$
$$\dot{Q} = hA_s(T_\infty - T_s) = (10.05\text{ W/m}^2\text{.°C})(20\text{ m}^2)(120-30)\text{°C} = 18{,}100\text{ W} = \mathbf{18.10\ kW}$$

(*b*) If the air flows parallel to the 2.5 m side, the Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(6\text{ m/s})(2.5\text{ m})}{2.486\times10^{-5}\text{ m}^2\text{/s}} = 6.034\times10^5$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

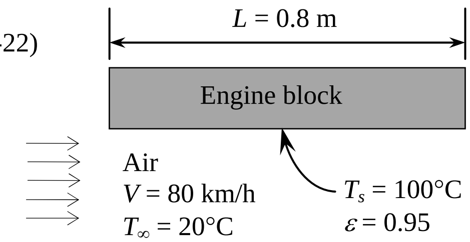
$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{0.8} - 871)\,\text{Pr}^{1/3} = [0.037(6.034\times10^5)^{0.8} - 871](0.7166)^{1/3} = 615.1$$
$$h = \frac{k}{L}Nu = \frac{0.02917\text{ W/m.°C}}{2.5\text{ m}}(615.1) = 7.177\text{ W/m}^2\text{.°C}$$
$$\dot{Q} = hA_s(T_\infty - T_s) = (7.177\text{ W/m}^2\text{.°C})(20\text{ m}^2)(120-30)\text{°C} = 12{,}920\text{ W} = \mathbf{12.92\ kW}$$

12-49 A car travels at a velocity of 80 km/h. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas with constant properties. **4** The flow is turbulent over the entire surface because of the constant agitation of the engine block.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (100+20)/2 = 60°\text{C}$ are (Table A-22)

$$k = 0.02808 \text{ W/m.°C}$$
$$\nu = 1.896\times10^{-5} \text{ m}^2/\text{s}$$
$$\Pr = 0.7202$$



Analysis Air flows parallel to the 0.4 m side. The Reynolds number in this case is

$$\mathrm{Re}_L = \frac{V_\infty L}{\nu} = \frac{[(80\times1000/3600)\text{ m/s}](0.8\text{ m})}{1.896\times10^{-5}\text{ m}^2/\text{s}} = 9.376\times10^5$$

which is greater than the critical Reynolds number and thus the flow is laminar + turbulent. But the flow is assumed to be turbulent over the entire surface because of the constant agitation of the engine block. Using the proper relations, the Nusselt number, the heat transfer coefficient, and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.037\,\mathrm{Re}_L^{0.8}\,\Pr^{1/3} = 0.037(9.376\times10^5)^{0.8}(0.7202)^{1/3} = 1988$$

$$h = \frac{k}{L}Nu = \frac{0.02808\text{ W/m.°C}}{0.8\text{ m}}(1988) = 69.78\text{ W/m}^2.°\text{C}$$

$$A_s = wL = (0.8\text{ m})(0.4\text{ m}) = 0.32\text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (69.78\text{ W/m}^2.°\text{C})(0.32\text{ m}^2)(100-20)°\text{C} = \mathbf{1786\ W}$$

The radiation heat transfer from the same surface is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$
$$= (0.95)(0.32\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)[(100+273\text{ K})^4 - (25+273\text{ K})^4] = \mathbf{198\ W}$$

Then the total rate of heat transfer from that surface becomes

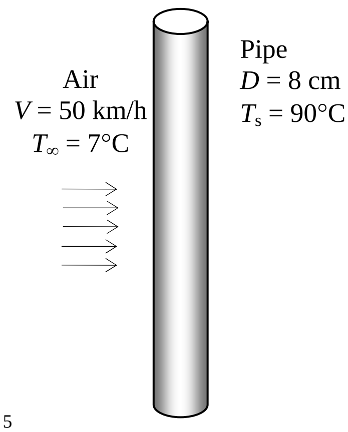
$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = (1786 + 198)\text{W} = \mathbf{1984\ W}$$

12-68 A steam pipe is exposed to windy air. The rate of heat loss from the steam is to be determined.√

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (90+7)/2 = 48.5$°C are (Table A-22)

$$k = 0.02724 \text{ W/m.°C}$$
$$\nu = 1.784\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7232$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(50 \text{ km/h})(1000 \text{ m/km})/(3600 \text{ s/h})](0.08 \text{ m})}{1.784\times10^{-5} \text{ m}^2\text{/s}} = 6.228\times10^4$$

The Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(6.228\times10^4)^{0.5}(0.7232)^{1/3}}{\left[1+(0.4/0.7232)^{2/3}\right]^{1/4}}\left[1+\left(\frac{6.228\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 159.1$$

The heat transfer coefficient and the heat transfer rate become

$$h = \frac{k}{D}Nu = \frac{0.02724 \text{ W/m.°C}}{0.08 \text{ m}}(159.1) = 54.17 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL = \pi(0.08 \text{ m})(1 \text{ m}) = 0.2513 \text{ m}^2$$

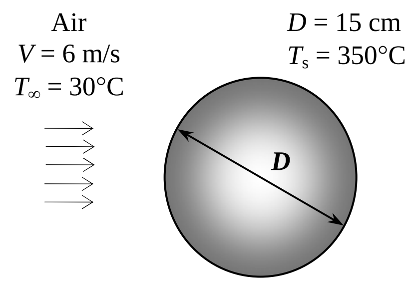
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (54.17 \text{ W/m}^2\text{.°C})(0.2513 \text{ m}^2)(90-7)\text{°C} = \mathbf{1130\ W} \text{ (per m length)}$$

12-70 A hot stainless steel ball is cooled by forced air. The average convection heat transfer coefficient and the cooling time are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The outer surface temperature of the ball is uniform at all times.

Properties The average surface temperature is (350+250)/2 = 300°C, and the properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@\,300\,°\text{C}} = 2.934\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7282$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6 \text{ m/s})(0.15 \text{ m})}{1.608\times10^{-5} \text{ m}^2\text{/s}} = 5.597\times10^4$$

The Nusselt number corresponding to this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(5.597\times10^4)^{0.5} + 0.06(5.597\times10^4)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{2.934\times10^{-5}}\right)^{1/4} = 145.6$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02588 \text{ W/m.°C}}{0.15 \text{ m}}(145.6) = \mathbf{25.12\ W/m^2.°C}$$

The average rate of heat transfer can be determined from Newton's law of cooling by using average surface temperature of the ball

$$A_s = \pi D^2 = \pi(0.15 \text{ m})^2 = 0.07069 \text{ m}^2$$
$$\dot{Q}_{avg} = hA_s(T_s - T_\infty) = (25.12 \text{ W/m}^2.°\text{C})(0.07069 \text{ m}^2)(300-30)°\text{C} = 479.5 \text{ W}$$

Assuming the ball temperature to be nearly uniform, the total heat transferred from the ball during the cooling from 350°C to 250°C can be determined from

$$Q_{\text{total}} = mc_p(T_1 - T_2)$$

where $m = \rho \boldsymbol{V} = \rho\frac{\pi D^3}{6} = (8055 \text{ kg/m}^3)\frac{\pi(0.15 \text{ m})^3}{6} = 14.23 \text{ kg}$

Therefore, $Q_{\text{total}} = mc_p(T_1 - T_2) = (14.23 \text{ kg})(480 \text{ J/kg.°C})(350-250)°\text{C} = 683{,}250 \text{ J}$

Then the time of cooling becomes

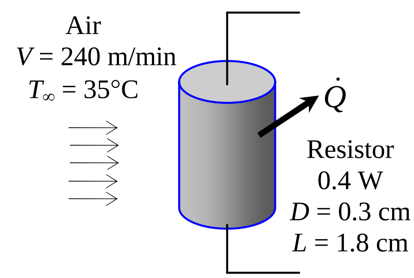
$$\Delta t = \frac{Q}{\dot{Q}_{avg}} = \frac{683{,}250 \text{ J}}{479.5 \text{ J/s}} = 1425 \text{ s} = \mathbf{23.7\ min}$$

12-87 A cylindrical electronic component mounted on a circuit board is cooled by air flowing across it. The surface temperature of the component is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 50°C. The properties of air at 1 atm and at this temperature are (Table A-22)

$$k = 0.02735 \text{ W/m.°C}$$
$$\nu = 1.798\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7228$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(240/60 \text{ m/s})(0.003 \text{ m})}{1.798\times10^{-5} \text{ m}^2\text{/s}} = 667.4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\Pr^{1/3}}{\left[1+(0.4/\Pr)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(667.4)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{667.4}{282{,}000}\right)^{5/8}\right]^{4/5} = 13.17$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02735 \text{ W/m.°C}}{0.003 \text{ m}}(13.17) = 120.0 \text{ W/m}^2\text{.°C}$$

Then the surface temperature of the component becomes

$$A_s = \pi DL = \pi(0.003 \text{ m})(0.018 \text{ m}) = 0.0001696 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 35\text{ °C} + \frac{0.4 \text{ W}}{(120.0 \text{ W/m}^2\text{.°C})(0.0001696 \text{ m}^2)} = \mathbf{54.6°C}$$

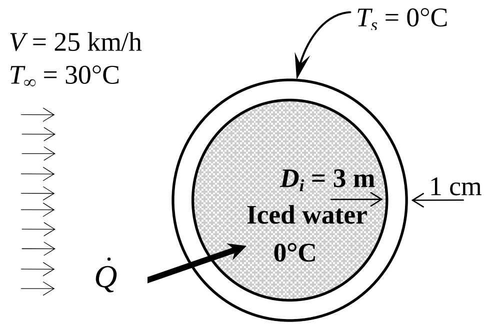
The film temperature is (54.6+35)/2=44.8°C, which is sufficiently close to the assumed value of 50°C. Therefore, there is no need to repeat calculations.

12-98 A spherical tank used to store iced water is subjected to winds. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Thermal resistance of the tank is negligible. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties. **5** The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@\,0°\text{C}} = 1.729\times10^{-5} \text{ kg/m.s}$$
$$\text{Pr} = 0.7282$$



Analysis (*a*) The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25\times1000/3600)\text{ m/s}](3.02\text{ m})}{1.608\times10^{-5}\text{ m}^2\text{/s}} = 1.304\times10^6$$

The Nusselt number corresponding to this Reynolds number is determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(1.304\times10^6)^{0.5} + 0.06(1.304\times10^6)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 1056$$

and $$h = \frac{k}{D}Nu = \frac{0.02588\text{ W/m.°C}}{3.02\text{ m}}(1056) = 9.05\text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the iced water is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (9.05\text{ W/m}^2\text{.°C})[\pi(3.02\text{ m})^2](30-0)\text{°C} = \mathbf{7779\ W}$$

(*b*) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q}\Delta t = (7.779\text{ kJ/s})(24\times3600\text{ s}) = 672{,}000\text{ kJ}$$

Then the amount of ice that melts during this period becomes

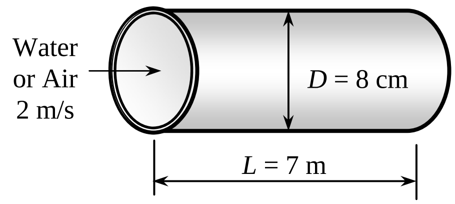
$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{672{,}000\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{2014\ kg}$$

13-39 The convection heat transfer coefficients for the flow of air and water are to be determined under similar conditions.

Assumptions **1** Steady flow conditions exist. **2** The surface heat flux is uniform. **3** The inner surfaces of the tube are smooth.

Properties The properties of air at 25°C are (Table A-22)

$$k = 0.02551 \text{ W/m.°C}$$
$$\nu = 1.562\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7296$$



The properties of water at 25°C are (Table A-15)

$$\rho = 997 \text{ kg/m}^3$$
$$k = 0.607 \text{ W/m.°C}$$
$$\nu = \mu/\rho = 0.891\times10^{-3}/997 = 8.937\times10^{-7} \text{ m}^2\text{/s}$$
$$\Pr = 6.14$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.08 \text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 10{,}243$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.08 \text{ m}) = 0.8 \text{ m}$$

which is much shorter than the total length of the tube. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.4} = 0.023(10{,}243)^{0.8}(0.7296)^{0.4} = 32.76$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02551 \text{ W/m.°C}}{0.08 \text{ m}}(32.76) = \mathbf{10.45\ W/m^2.°C}$$

Repeating calculations for water:

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.08 \text{ m})}{8.937\times10^{-7} \text{ m}^2\text{/s}} = 179{,}035$$

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.4} = 0.023(179{,}035)^{0.8}(6.14)^{0.4} = 757.4$$

$$h = \frac{k}{D}Nu = \frac{0.607 \text{ W/m.°C}}{0.08 \text{ m}}(757.4) = \mathbf{5747\ W/m^2.°C}$$

Discussion The heat transfer coefficient for water is 550 times that of air.

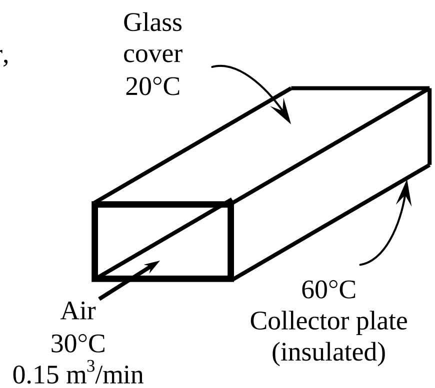
13-47 Air enters the constant spacing between the glass cover and the plate of a solar collector. The net rate of heat transfer and the temperature rise of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The inner surfaces of the spacing are smooth. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and estimated average temperature of 35°C are (Table A-22)

$$\rho = 1.145 \text{kg/m}^3, \quad k = 0.02625 \text{ W/m.°C}$$

$$\nu = 1.655\times10^{-5} \text{ m}^2\text{/s}, \quad c_p = 1007 \text{ J/kg.°C}, \quad \text{Pr} = 0.7268$$



Analysis Mass flow rate, cross sectional area, hydraulic diameter, mean velocity of air and the Reynolds number are

$$\dot{m} = \rho\dot{V} = (1.145 \text{ kg/m}^3)(0.15 \text{ m}^3\text{/s}) = 0.1718 \text{ kg/s}$$

$$A_c = (1 \text{ m})(0.03 \text{ m}) = 0.03 \text{ m}^2$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.03 \text{ m}^2)}{2(1 \text{ m} + 0.03 \text{ m})} = 0.05825 \text{ m}$$

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.15 \text{ m}^3\text{/s}}{0.03 \text{ m}^2} = 5 \text{ m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(5 \text{ m/s})(0.05825 \text{ m})}{1.655\times10^{-5} \text{ m}^2\text{/s}} = 17{,}600$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D_h = 10(0.05825 \text{ m}) = 0.5825 \text{ m}$$

which are much shorter than the total length of the collector. Therefore, we can assume fully developed turbulent flow in the entire collector, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} = 0.023(17{,}600)^{0.8}(0.7268)^{0.4} = 50.43$$

and $$h = \frac{k}{D_h} Nu = \frac{0.02625 \text{ W/m.°C}}{0.05825 \text{ m}}(50.43) = 22.73 \text{ W/m}^2\text{.°C}$$

The exit temperature of air can be calculated using the "average" surface temperature as

$$A_s = 2(5 \text{ m})(1 \text{ m}) = 10 \text{ m}^2, \qquad T_{s,avg} = \frac{60+20}{2} = 40\text{°C}$$

$$T_e = T_{s,avg} - (T_{s,avg} - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) = 40 - (40-30)\exp\left(-\frac{22.73\times10}{0.1718\times1007}\right) = 37.31\text{°C}$$

The temperature rise of air is

$$\Delta T = 37.3\text{°C} - 30\text{°C} = \mathbf{7.3°C}$$

The logarithmic mean temperature difference and the heat loss from the glass are

$$\Delta T_{\ln,glass} = \frac{T_e - T_i}{\ln\frac{T_s - T_e}{T_s - T_i}} = \frac{37.31-30}{\ln\frac{20-37.31}{20-30}} = 13.32\text{°C}$$

$$\dot{Q}_{glass} = hA_s\Delta T_{\ln} = (22.73 \text{ W/m}^2\text{.°C})(5 \text{ m}^2)(13.32\text{°C}) = 1514 \text{ W}$$

The logarithmic mean temperature difference and the heat gain of the absorber are

$$\Delta T_{\ln,absorber} = \frac{T_e - T_i}{\ln\frac{T_s - T_e}{T_s - T_i}} = \frac{37.31-30}{\ln\frac{60-37.31}{60-30}} = 26.17\text{°C}$$

$$\dot{Q}_{absorber} = hA\Delta T_{\ln} = (22.73 \text{ W/m}^2\text{.°C})(5 \text{ m}^2)(26.17\text{°C}) = 2975 \text{ W}$$

Then the net rate of heat transfer becomes

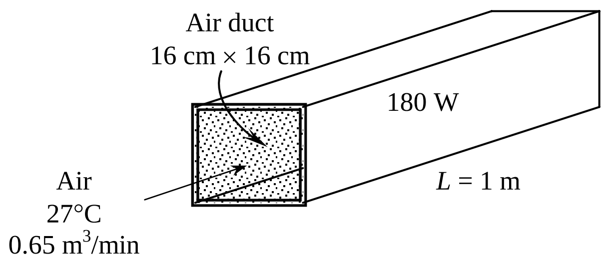
$$\dot{Q}_{net} = 2975 - 1514 = \mathbf{1461 \ W}$$

13-59 The components of an electronic system located in a rectangular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the duct are smooth. **3** The thermal resistance of the duct is negligible. **4** Air is an ideal gas with constant properties. **5** The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 35°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-22)

$$\rho = 1.145\ \text{kg/m}^3$$
$$k = 0.02625\ \text{W/m.°C}$$
$$\nu = 1.655\times10^{-5}\ \text{m}^2\text{/s}$$
$$c_p = 1007\ \text{J/kg.°C}$$
$$\Pr = 0.7268$$



Analysis (*a*) The mass flow rate of air and the exit temperature are determined from

$$\dot{m} = \rho\dot{V} = (1.145\ \text{kg/m}^3)(0.65\ \text{m}^3\text{/min}) = 0.7443\ \text{kg/min} = 0.0124\ \text{kg/s}$$

$$\dot{Q} = \dot{m}c_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}c_p} = 27°\text{C} + \frac{(0.85)(180\ \text{W})}{(0.0124\ \text{kg/s})(1007\ \text{J/kg.°C})} = \mathbf{39.3°C}$$

(*b*) The mean fluid velocity and hydraulic diameter are

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.65\ \text{m/min}}{(0.16\ \text{m})(0.16\ \text{m})} = 25.4\ \text{m/min} = 0.4232\ \text{m/s}$$

$$D_h = \frac{4A_c}{p} = \frac{4(0.16\ \text{m})(0.16\ \text{m})}{4(0.16\ \text{m})} = 0.16\ \text{m}$$

Then

$$\text{Re} = \frac{V_{\text{avg}}D_h}{\nu} = \frac{(0.4232\ \text{m/s})(0.16\ \text{m})}{1.655\times10^{-5}\ \text{m}^2\text{/s}} = 4091$$

which is not greater than 10,000 but the components will cause turbulence and thus we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.4} = 0.023(4091)^{0.8}(0.7268)^{0.4} = 15.69$$

and

$$h = \frac{k}{D_h}Nu = \frac{0.02625\ \text{W/m.°C}}{0.16\ \text{m}}(15.69) = 2.574\ \text{W/m}^2\text{.°C}$$

The highest component surface temperature will occur at the exit of the duct. Assuming uniform surface heat flux, its value is determined from

$$\dot{Q}/A_s = h(T_{s,highest} - T_e)$$

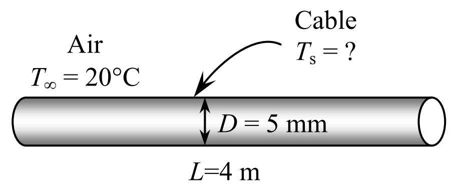
$$T_{s,highest} = T_e + \frac{\dot{Q}/A_s}{h} = 39.2°\text{C} + \frac{(0.85)(180\ \text{W})/[4(0.16\ \text{m})(1\ \text{m})]}{2.574\ \text{W/m}^2\text{.°C}} = \mathbf{132°C}$$

14-18 Heat generated by the electrical resistance of a bare cable is dissipated to the surrounding air. The surface temperature of the cable is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The local atmospheric pressure is 1 atm. **4** The temperature of the surface of the cable is constant.

Properties We assume the surface temperature to be 100°C. Then the properties of air at 1 atm and the film temperature of $(T_s+T_\infty)/2$ = (100+20)/2 = 60°C are (Table A-22)

$$k = 0.02808 \text{ W/m.°C}$$
$$\nu = 1.896\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7202$$
$$\beta = \frac{1}{T_f} = \frac{1}{(60+273)\text{K}} = 0.003003 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the outer diameter of the pipe, $L_c = D = 0.005$ m. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003003 \text{ K}^{-1})(100-20 \text{ K})(0.005 \text{ m})^3}{(1.896\times10^{-5} \text{ m}^2\text{/s})^2}(0.7202) = 590.2$$

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+\left(0.559/\text{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(590.2)^{1/6}}{\left[1+\left(0.559/0.7202\right)^{9/16}\right]^{8/27}}\right\}^2 = 2.346$$

$$h = \frac{k}{D}Nu = \frac{0.02808 \text{ W/m.°C}}{0.005 \text{ m}}(2.346) = 13.17 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL = \pi(0.005 \text{ m})(4 \text{ m}) = 0.06283 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$(60 \text{ V})(1.5 \text{ A}) = (13.17 \text{ W/m}^2\text{.°C})(0.06283 \text{ m}^2)(T_s - 20)\text{°C}$$
$$T_s = 128.8\text{°C}$$

which is not close to the assumed value of 100°C. Repeating calculations for an assumed surface temperature of 120°C, [$T_f = (T_s+T_\infty)/2$ = (120+20)/2 = 70°C]

$$k = 0.02881 \text{ W/m.°C}$$
$$\nu = 1.995\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7177$$
$$\beta = \frac{1}{T_f} = \frac{1}{(70+273)\text{K}} = 0.002915 \text{ K}^{-1}$$

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.002915 \text{ K}^{-1})(120-20 \text{ K})(0.005 \text{ m})^3}{(1.995\times10^{-5} \text{ m}^2\text{/s})^2}(0.7177) = 644.6$$

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+\left(0.559/\text{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(644.6)^{1/6}}{\left[1+\left(0.559/0.7177\right)^{9/16}\right]^{8/27}}\right\}^2 = 2.387$$

$$h = \frac{k}{D}Nu = \frac{0.02881 \text{ W/m.°C}}{0.005 \text{ m}}(2.387) = 13.76 \text{ W/m}^2\text{.°C}$$

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$(60 \text{ V})(1.5A) = (13.76 \text{ W/m}^2\text{.°C})(0.06283 \text{ m}^2)(T_s - 20)\text{°C}$$
$$T_s = \mathbf{124.1°C}$$

which is sufficiently close to the assumed value of 120°C.

14-24 A cylindrical resistance heater is placed horizontally in a fluid. The outer surface temperature of the resistance wire is to be determined for two different fluids.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The local atmospheric pressure is 1 atm. **4** Any heat transfer by radiation is ignored. **5** Properties are evaluated at 500°C for air and 40°C for water.

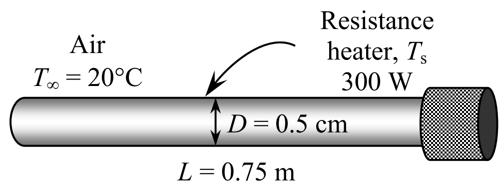
Properties The properties of air at 1 atm and 500°C are (Table A-22)

$$k = 0.05572 \text{ W/m.°C}$$

$$\nu = 7.804\times10^{-5} \text{ m}^2\text{/s}$$

$$\text{Pr} = 0.6986,$$

$$\beta = \frac{1}{T_f} = \frac{1}{(500+273)\text{K}} = 0.001294 \text{ K}^{-1}$$



The properties of water at 40°C are (Table A-15)

$$k = 0.631 \text{ W/m.°C}, \quad \nu = \mu/\rho = 0.6582\times10^{-6} \text{ m}^2\text{/s}$$

$$\text{Pr} = 4.32, \quad \beta = 0.000377 \text{ K}^{-1}$$

Analysis (*a*) The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by "guessing" the surface temperature to be 1200°C for the calculation of *h*. We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the outer diameter of the wire, $L_c = D = 0.005$ m. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.001294 \text{ K}^{-1})(1200-20)\text{°C}(0.005 \text{ m})^3}{(7.804\times10^{-5} \text{ m}^2\text{/s})^2}(0.6986) = 214.7$$

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(214.7)^{1/6}}{\left[1+(0.559/0.6986)^{9/16}\right]^{8/27}}\right\}^2 = 1.919$$

$$h = \frac{k}{D}Nu = \frac{0.05572 \text{ W/m.°C}}{0.005 \text{ m}}(1.919) = 21.38 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL = \pi(0.005 \text{ m})(0.75 \text{ m}) = 0.01178 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow 300 \text{ W} = (21.38 \text{ W/m}^2\text{.°C})(0.01178 \text{ m}^2)(T_s - 20)\text{°C} \rightarrow T_s = \mathbf{1211°C}$$

which is sufficiently close to the assumed value of 1200°C used in the evaluation of *h*, and thus it is not necessary to repeat calculations.

(*b*) For the case of water, we "guess" the surface temperature to be 40°C. The characteristic length in this case is the outer diameter of the wire, $L_c = D = 0.005$ m. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.000377 \text{ K}^{-1})(40-20 \text{ K})(0.005 \text{ m})^3}{(0.6582\times10^{-6} \text{ m}^2\text{/s})^2}(4.32) = 92{,}197$$

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(92{,}197)^{1/6}}{\left[1+(0.559/4.32)^{9/16}\right]^{8/27}}\right\}^2 = 8.986$$

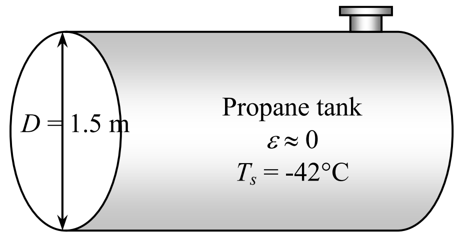
$$h = \frac{k}{D}Nu = \frac{0.631 \text{ W/m.°C}}{0.005 \text{ m}}(8.986) = 1134 \text{ W/m}^2\text{.°C}$$

and $$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow 300 \text{ W} = (1134 \text{ W/m}^2\text{.°C})(0.01178 \text{ m}^2)(T_s - 20)\text{°C} \longrightarrow T_s = \mathbf{42.5°C}$$

which is sufficiently close to the assumed value of 40°C in the evaluation of the properties and *h*. The film temperature in this case is $(T_s+T_\infty)/2 = (42.5+20)/2 = 31.3$°C, which is close to the value of 40°C used in the evaluation of the properties.

14-45 A cylindrical propane tank is exposed to calm ambient air. The propane is slowly vaporized due to a crack developed at the top of the tank. The time it will take for the tank to empty is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The local atmospheric pressure is 1 atm. **4** Radiation heat transfer is negligible.



Properties The properties of air at 1 atm and the film temperature of $(T_s+T_\infty)/2$ = (-42+25)/2 = -8.5°C are (Table A-22)

$$k = 0.02299 \text{ W/m.°C}$$

$$\nu = 1.265\times10^{-5} \text{ m}^2/\text{s}$$

$$\Pr = 0.7383$$

$$\beta = \frac{1}{T_f} = \frac{1}{(-8.5+273)\text{K}} = 0.003781 \text{ K}^{-1}$$

Analysis The tank gains heat through its cylindrical surface as well as its circular end surfaces. For convenience, we take the heat transfer coefficient at the end surfaces of the tank to be the same as that of its side surface. (The alternative is to treat the end surfaces as a vertical plate, but this will double the amount of calculations without providing much improvement in accuracy since the area of the end surfaces is much smaller and it is circular in shape rather than being rectangular). The characteristic length in this case is the outer diameter of the tank, $L_c = D = 1.5$ m. Then,

$$Ra = \frac{g\beta(T_\infty - T_s)D^3}{\nu^2}\Pr = \frac{(9.81 \text{ m/s}^2)(0.003781 \text{ K}^{-1})[(25-(-42)\text{ K}](1.5\text{ m})^3}{(1.265\times10^{-5}\text{ m}^2/\text{s})^2}(0.7383) = 3.869\times10^{10}$$

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+\left(0.559/\Pr\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(3.869\times10^{10})^{1/6}}{\left[1+\left(0.559/0.7383\right)^{9/16}\right]^{8/27}}\right\}^2 = 374.1$$

$$h = \frac{k}{D}Nu = \frac{0.02299 \text{ W/m.°C}}{1.5 \text{ m}}(374.1) = 5.733 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL + 2\pi D^2/4 = \pi(1.5\text{ m})(4\text{ m}) + 2\pi(1.5\text{ m})^2/4 = 22.38 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_\infty - T_s) = (5.733 \text{ W/m}^2\text{.°C})(22.38 \text{ m}^2)[(25-(-42)]\text{°C} = 8598 \text{ W}$$

The total mass and the rate of evaporation of propane are

$$m = \rho \boldsymbol{V} = \rho\frac{\pi D^2}{4}L = (581 \text{ kg/m}^3)\frac{\pi(1.5\text{ m})^2}{4}(4\text{ m}) = 4107 \text{ kg}$$

$$\dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{8.598 \text{ kJ/s}}{425 \text{ kJ/kg}} = 0.02023 \text{ kg/s}$$

and it will take

$$\Delta t = \frac{m}{\dot{m}} = \frac{4107 \text{ kg}}{0.02023 \text{ kg/s}} = 202{,}996 \text{ s} = \mathbf{56.4 \text{ hours}}$$

for the propane tank to empty.

14-82 An ice chest filled with ice at 0°C is exposed to ambient air. The time it will take for the ice in the chest to melt completely is to be determined for natural and forced convection cases.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** Heat transfer from the base of the ice chest is disregarded. **4** Radiation effects are negligible. **5** Heat transfer coefficient is the same for all surfaces considered. **6** The local atmospheric pressure is 1 atm.

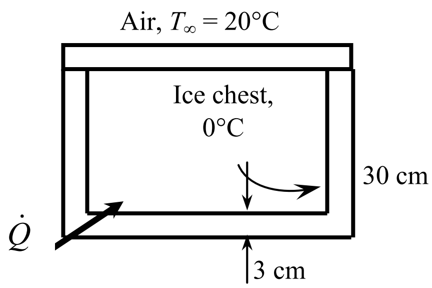
Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s+T_\infty)/2 = (15+20)/2 = 17.5$°C are (Table A-22)

$$k = 0.02495 \text{ W/m.°C}$$

$$\nu = 1.493\times10^{-5} \text{ m}^2\text{/s}$$

$$\text{Pr} = 0.7316$$

$$\beta = \frac{1}{T_f} = \frac{1}{(17.5+273)\text{K}} = 0.003442 \text{ K}^{-1}$$



Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by "guessing" the surface temperature to be 15°C for the evaluation of the properties and *h*. We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length for the side surfaces is the height of the chest, $L_c = L = 0.3$ m Then,

$$Ra = \frac{g\beta(T_\infty - T_s)L^3}{\nu^2}\text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.003442 \text{ K}^{-1})(20-15 \text{ K})(0.3 \text{ m})^3}{(1.493\times10^{-5} \text{ m}^2\text{/s})^2}(0.7316) = 1.495\times10^7$$

$$Nu = \left\{0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+\left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.825 + \frac{0.387(1.495\times10^7)^{1/6}}{\left[1+\left(\frac{0.492}{0.7316}\right)^{9/16}\right]^{8/27}}\right\}^2 = 35.15$$

$$h = \frac{k}{L}Nu = \frac{0.02495 \text{ W/m.°C}}{0.3 \text{ m}}(35.15) = 2.923 \text{ W/m}^2\text{.°C}$$

The heat transfer coefficient at the top surface can be determined similarly. However, the top surface constitutes only about one-fourth of the heat transfer area, and thus we can use the heat transfer coefficient for the side surfaces for the top surface also for simplicity. The heat transfer surface area is

$$A_s = 4(0.3 \text{ m})(0.4 \text{ m}) + (0.4 \text{ m})(0.4 \text{ m}) = 0.64 \text{ m}^2$$

Then the rate of heat transfer becomes

$$\dot{Q} = \frac{T_\infty - T_{s,i}}{R_{wall} + R_{conv,o}} = \frac{T_\infty - T_{s,i}}{\frac{L}{kA_s} + \frac{1}{hA_s}} = \frac{(20-0)\text{°C}}{\frac{0.03 \text{ m}}{(0.033 \text{ W/m.°C})(0.64 \text{ m}^2)} + \frac{1}{(2.923 \text{ W/m}^2\text{.°C})(0.64 \text{ m}^2)}} = 10.23 \text{ W}$$

The outer surface temperature of the ice chest is determined from Newton's law of cooling to be

$$\dot{Q} = hA_s(T_\infty - T_s) \rightarrow T_s = T_\infty - \frac{\dot{Q}}{hA_s} = 20\text{°C} - \frac{10.23 \text{ W}}{(2.923 \text{ W/m}^2\text{.C})(0.64 \text{ m}^2)} = 14.53\text{°C}$$

which is almost identical to the assumed value of 15°C used in the evaluation of properties and *h*. Therefore, there is no need to repeat the calculations.

The rate at which the ice will melt is

$$\dot{Q} = \dot{m}h_{if} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{10.23\times10^{-3} \text{ kJ/s}}{333.7 \text{ kJ/kg}} = 3.066\times10^{-5} \text{ kg/s}$$

Therefore, the melting of the ice in the chest completely will take

$$m = \dot{m}\Delta t \longrightarrow \Delta t = \frac{m}{\dot{m}} = \frac{30 \text{ kg}}{3.066\times10^{-5} \text{ kg/s}} = 9.786\times10^{5} \text{ s} = \mathbf{271.8\ h} = \mathbf{11.3\ days}$$

(b) The temperature drop across the styrofoam will be much greater in this case than that across thermal boundary layer on the surface. Thus we assume outer surface temperature of the styrofoam to be 19 °C. Radiation heat transfer will be neglected. The properties of air at 1 atm and the film temperature of $(T_s+T_\infty)/2$ = (19+20)/2 = 19.5°C are (Table A-22)

$$k = 0.0251 \text{ W/m.°C}$$

$$\nu = 1.511\times10^{-5} \text{ m}^2\text{/s}$$

$$\text{Pr} = 0.7310$$

$$\beta = \frac{1}{T_f} = \frac{1}{(19.5+273)\text{K}} = 0.00342 \text{ K}^{-1}$$

The characteristic length in this case is the width of the chest, $L_c = W$ =0.4 m. Then,

$$\text{Re} = \frac{VW}{\nu} = \frac{(50\times1000/3600 \text{ m/s})(0.4 \text{ m})}{1.511\times10^{-5} \text{ m}^2\text{/s}} = 367{,}700$$

which is less than critical Reynolds number (5×10^5). Therefore the flow is laminar, and the Nusselt number is determined from

$$Nu = \frac{hW}{k} = 0.664\,\text{Re}^{0.5}\,\text{Pr}^{1/3} = 0.664(367{,}700)^{0.5}(0.7310)^{1/3} = 362.7$$

$$h = \frac{k}{W}Nu = \frac{0.0251 \text{ W/m.°C}}{0.4 \text{ m}}(362.7) = 22.76 \text{ W/m}^2\text{.°C}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \frac{T_\infty - T_{s,i}}{R_{wall} + R_{conv,o}} = \frac{T_\infty - T_{s,i}}{\frac{L}{kA_s} + \frac{1}{hA_s}} = \frac{(20-0)\text{°C}}{\frac{0.03 \text{ m}}{(0.033 \text{ W/m.°C})(0.64 \text{ m}^2)} + \frac{1}{(22.76 \text{ W/m}^2\text{.°C})(0.64 \text{ m}^2)}} = 13.43 \text{ W}$$

The outer surface temperature of the ice chest is determined from Newton's law of cooling to be

$$\dot{Q} = hA_s(T_\infty - T_s) \rightarrow T_s = T_\infty - \frac{\dot{Q}}{hA_s} = 20\text{°C} - \frac{13.43 \text{ W}}{(22.76 \text{ W/m}^2\text{.C})(0.64 \text{ m}^2)} = 19.1\text{°C}$$

which is almost identical to the assumed value of 19°C used in the evaluation of properties and *h*. Therefore, there is no need to repeat the calculations. Then the rate at which the ice will melt becomes

$$\dot{Q} = \dot{m}h_{if} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{13.43\times10^{-3} \text{ kJ/s}}{333.7 \text{ kJ/kg}} = 4.025\times10^{-5} \text{ kg/s}$$

Therefore, the melting of the ice in the chest completely will take

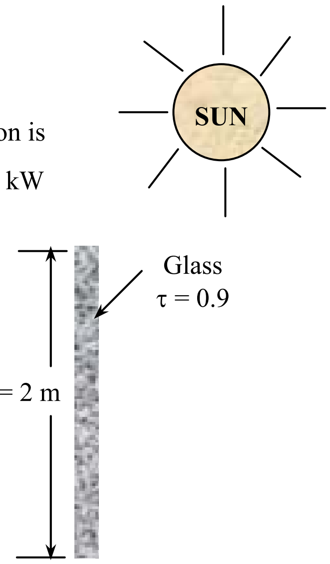
$$m = \dot{m}\Delta t \longrightarrow \Delta t = \frac{m}{\dot{m}} = \frac{30}{4.025\times10^{-5}} = 7.454\times10^{5} \text{ s} = \mathbf{207.05\ h} = \mathbf{8.6\ days}$$

15-27 A glass window transmits 90% of the radiation in a specified wavelength range and is opaque for radiation at other wavelengths. The rate of radiation transmitted through this window is to be determined for two cases.

Assumptions The sources behave as a black body.

Analysis The surface area of the glass window is

$$A_s = 4\ \text{m}^2$$



(*a*) For a blackbody source at 5800 K, the total blackbody radiation emission is

$$E_b(T) = \sigma T^4 A_s = (5.67\times10^{-8}\ \text{kW/m}^2.\text{K})^4(5800\ \text{K})^4(4\ \text{m}^2) = 2.567\times10^5\ \text{kW}$$

The fraction of radiation in the range of 0.3 to 3.0 μm is

$$\lambda_1 T = (0.30\ \mu\text{m})(5800\ \text{K}) = 1740\ \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.03345$$

$$\lambda_2 T = (3.0\ \mu\text{m})(5800\ \text{K}) = 17{,}400\ \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.97875$$

$$\Delta f = f_{\lambda_2} - f_{\lambda_1} = 0.97875 - 0.03345 = 0.9453$$

Noting that 90% of the total radiation is transmitted through the window,

$$E_{\text{transmit}} = 0.90\Delta f E_b(T)$$

$$= (0.90)(0.9453)(2.567\times10^5\ \text{kW}) = \mathbf{2.184\times10^5\ kW}$$

(*b*) For a blackbody source at 1000 K, the total blackbody emissive power is

$$E_b(T) = \sigma T^4 A_s = (5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1000\ \text{K})^4(4\ \text{m}^2) = 226.8\ \text{kW}$$

The fraction of radiation in the visible range of 0.3 to 3.0 μm is

$$\lambda_1 T = (0.30\ \mu\text{m})(1000\ \text{K}) = 300\ \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.0000$$

$$\lambda_2 T = (3.0\ \mu\text{m})(1000\ \text{K}) = 3000\ \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.273232$$

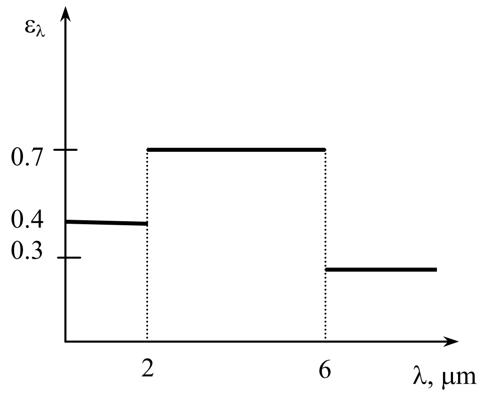
$$\Delta f = f_{\lambda_2} - f_{\lambda_1} = 0.273232 - 0$$

and

$$E_{\text{transmit}} = 0.90\Delta f E_b(T) = (0.90)(0.273232)(226.8\ \text{kW}) = \mathbf{55.8\ kW}$$

15-33 The variation of emissivity of a surface at a specified temperature with wavelength is given. The average emissivity of the surface and its emissive power are to be determined.

Analysis The average emissivity of the surface can be determined from



$$\varepsilon(T) = \frac{\varepsilon_1 \int_0^{\lambda_1} E_{b_\lambda}(T)d\lambda}{\sigma T^4} + \frac{\varepsilon_2 \int_{\lambda_1}^{\lambda_2} E_{b_\lambda}(T)d\lambda}{\sigma T^4} + \frac{\varepsilon_3 \int_{\lambda_2}^{\infty} E_{b_\lambda}(T)d\lambda}{\sigma T^4}$$

$$= \varepsilon_1 f_{0\text{-}\lambda_1} + \varepsilon_2 f_{\lambda_1\text{-}\lambda_2} + \varepsilon_3 f_{\lambda_2\text{-}\infty}$$

$$= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (f_{\lambda_2} - f_{\lambda_1}) + \varepsilon_3 (1 - f_{\lambda_2})$$

where f_{λ_1} and f_{λ_2} are blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$, determined from

$$\lambda_1 T = (2\ \mu\text{m})(1000\ \text{K}) = 2000\ \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.066728$$

$$\lambda_2 T = (6\ \mu\text{m})(1000\ \text{K}) = 6000\ \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.737818$$

$$f_{0-\lambda_1} = f_{\lambda 1} - f_0 = f_{\lambda_1} \text{ since } f_0 = 0 \text{ and } f_{\lambda_2-\infty} = f_\infty - f_{\lambda_2} \text{ since } f_\infty = 1.$$

and,

$$\varepsilon = (0.4)0.066728 + (0.7)(0.737818 - 0.066728) + (0.3)(1 - 0.737818) = \mathbf{0.575}$$

Then the emissive power of the surface becomes

$$E = \varepsilon\sigma T^4 = 0.575(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1000\ \text{K})^4 = \mathbf{32.6\ kW/m^2}$$

15-50 A cylindrical enclosure is considered. The view factor from the side surface of this cylindrical enclosure to its base surface is to be determined.

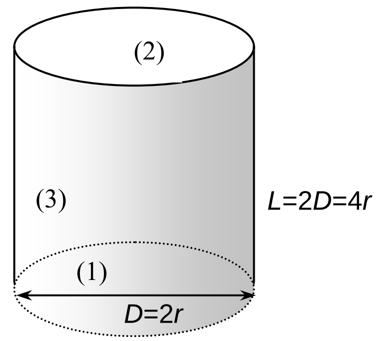
Assumptions The surfaces are diffuse emitters and reflectors.

Analysis We designate the surfaces as follows:

Base surface by (1),

top surface by (2), and

side surface by (3).



Then from Fig. 15-7

$$\left.\begin{array}{l} \dfrac{L}{r_1} = \dfrac{4r_1}{r_1} = 4 \\ \dfrac{r_2}{L} = \dfrac{r_2}{4r_2} = 0.25 \end{array}\right\} F_{12} = F_{21} = 0.05$$

$$\text{summation rule}: F_{11} + F_{12} + F_{13} = 1$$

$$0 + 0.05 + F_{13} = 1 \longrightarrow F_{13} = 0.95$$

$$\text{reciprocity rule}: A_1 F_{13} = A_3 F_{31} \longrightarrow F_{31} = \frac{A_1}{A_3} F_{13} = \frac{\pi r_1^2}{2\pi r_1 L} F_{13} = \frac{\pi r_1^2}{8\pi r_1^2} F_{13} = \frac{1}{8}(0.95) = \mathbf{0.119}$$

Discussion This problem can be solved more accurately by using the view factor relation from Table 15-3 to be

$$R_1 = \frac{r_1}{L} = \frac{r_1}{4r_1} = 0.25$$

$$R_2 = \frac{r_2}{L} = \frac{r_2}{4r_2} = 0.25$$

$$S = 1 + \frac{1 + R_2^2}{R_1^2} = 1 + \frac{1 + 0.25^2}{0.25^2} = 18$$

$$F_{12} = \tfrac{1}{2}\left\{ S - \left[S^2 - 4\left(\frac{R_2}{R_1}\right)^2 \right]^{0.5} \right\} = \tfrac{1}{2}\left\{ 18 - \left[18^2 - 4\left(\frac{1}{1}\right)^2 \right]^{0.5} \right\} = 0.056$$

$$F_{13} = 1 - F_{12} = 1 - 0.056 = 0.944$$

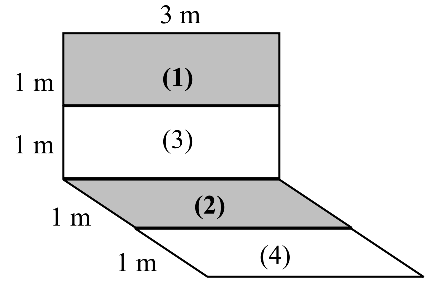
$$\text{reciprocity rule}: A_1 F_{13} = A_3 F_{31} \longrightarrow F_{31} = \frac{A_1}{A_3} F_{13} = \frac{\pi r_1^2}{2\pi r_1 L} F_{13} = \frac{\pi r_1^2}{8\pi r_1^2} F_{13} = \frac{1}{8}(0.944) = \mathbf{0.118}$$

15-57 The view factors between the rectangular surfaces shown in the figure are to be determined.

Assumptions The surfaces are diffuse emitters and reflectors.

Analysis We designate the different surfaces as follows:

shaded part of perpendicular surface by (1),
bottom part of perpendicular surface by (3),
shaded part of horizontal surface by (2), and
front part of horizontal surface by (4).



(*a*) From Fig.15-6

$$\left.\begin{array}{l}\frac{L_2}{W}=\frac{1}{3}\\ \frac{L_1}{W}=\frac{1}{3}\end{array}\right\}F_{23}=0.25 \quad \text{and} \quad \left.\begin{array}{l}\frac{L_2}{W}=\frac{2}{3}\\ \frac{L_1}{W}=\frac{1}{3}\end{array}\right\}F_{2\to(1+3)}=0.32$$

$$\text{superposition rule}: F_{2\to(1+3)}=F_{21}+F_{23}\longrightarrow F_{21}=F_{2\to(1+3)}-F_{23}=0.32-0.25=0.07$$

$$\text{reciprocity rule}: A_1=A_2\longrightarrow A_1F_{12}=A_2F_{21}\longrightarrow F_{12}=F_{21}=\mathbf{0.07}$$

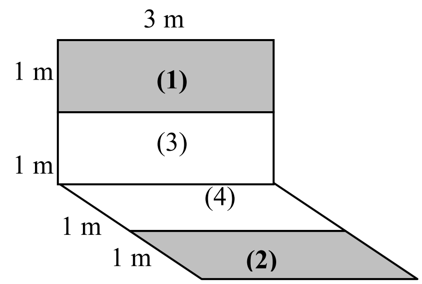
(*b*) From Fig.15-6,

$$\frac{L_2}{W}=\frac{1}{3} \text{ and } \left.\frac{L_1}{W}=\frac{2}{3}\right\}F_{(4+2)\to 3}=0.15 \quad \text{and} \quad \frac{L_2}{W}=\frac{2}{3} \text{ and } \left.\frac{L_1}{W}=\frac{2}{3}\right\}F_{(4+2)\to(1+3)}=0.22$$

$$\text{superposition rule}: F_{(4+2)\to(1+3)}=F_{(4+2)\to 1}+F_{(4+2)\to 3}\longrightarrow F_{(4+2)\to 1}=0.22-0.15=0.07$$

$$\text{reciprocity rule}: A_{(4+2)}F_{(4+2)\to 1}=A_1F_{1\to(4+2)}$$

$$\longrightarrow F_{1\to(4+2)}=\frac{A_{(4+2)}}{A_1}F_{(4+2)\to 1}=\frac{6}{3}(0.07)=0.14$$



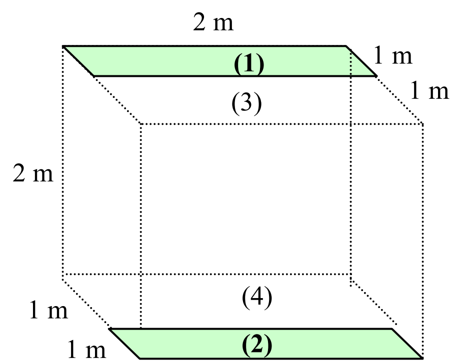
$$\text{superposition rule}: F_{1\to(4+2)}=F_{14}+F_{12}$$

$$\longrightarrow F_{14}=0.14-0.07=\mathbf{0.07}$$

since $F_{12}=0.07$ (from part *a*). Note that F_{14} in part (*b*) is equivalent to F_{12} in part (*a*).

(*c*) We designate

shaded part of top surface by (1),
remaining part of top surface by (3),
remaining part of bottom surface by (4), and
shaded part of bottom surface by (2).



From Fig.15-5,

$$\left.\begin{array}{l}\frac{L_2}{D}=\frac{2}{2}\\ \frac{L_1}{D}=\frac{2}{2}\end{array}\right\}F_{(2+4)\to(1+3)}=0.20 \quad \text{and} \quad \left.\begin{array}{l}\frac{L_2}{D}=\frac{2}{2}\\ \frac{L_1}{D}=\frac{1}{2}\end{array}\right\}F_{14}=0.12$$

$$\text{superposition rule}: F_{(2+4)\to(1+3)}=F_{(2+4)\to 1}+F_{(2+4)\to 3}$$

$$\text{symmetry rule}: F_{(2+4)\to 1}=F_{(2+4)\to 3}$$

Substituting symmetry rule gives

$$F_{(2+4)\to 1}=F_{(2+4)\to 3}=\frac{F_{(2+4)\to(1+3)}}{2}=\frac{0.20}{2}=0.10$$

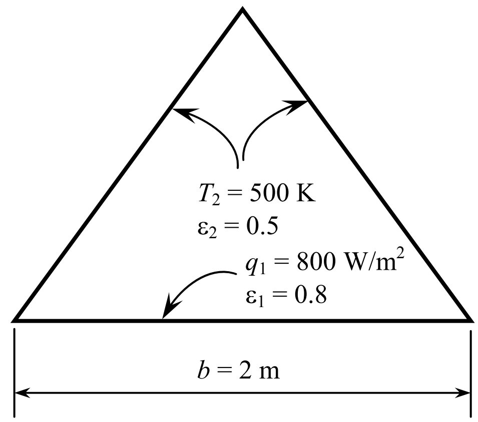
$$\text{reciprocity rule}: A_1F_{1\to(2+4)}=A_{(2+4)}F_{(2+4)\to 1}\longrightarrow(2)F_{1\to(2+4)}=(4)(0.10)\longrightarrow F_{1\to(2+4)}=0.20$$

$$\text{superposition rule}: F_{1\to(2+4)}=F_{12}+F_{14}\longrightarrow 0.20=F_{12}+0.12\longrightarrow F_{12}=0.20-0.12=\mathbf{0.08}$$

15-77 A furnace shaped like a long equilateral-triangular duct is considered. The temperature of the base surface is to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered. **4** End effects are neglected.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.8$ and $\varepsilon_2 = 0.5$.



Analysis This geometry can be treated as a two surface enclosure since two surfaces have identical properties. We consider base surface to be surface 1 and other two surface to be surface 2. Then the view factor between the two becomes $F_{12} = 1$. The temperature of the base surface is determined from

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

$$800 \text{ W} = \frac{(5.67\times10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_1)^4 - (500 \text{ K})^4]}{\frac{1-0.8}{(1 \text{ m}^2)(0.8)} + \frac{1}{(1 \text{ m}^2)(1)} + \frac{1-0.5}{(2 \text{ m}^2)(0.5)}}$$

$$T_1 = \mathbf{543\ K}$$

Note that $A_1 = 1 \text{ m}^2$ and $A_2 = 2 \text{ m}^2$.

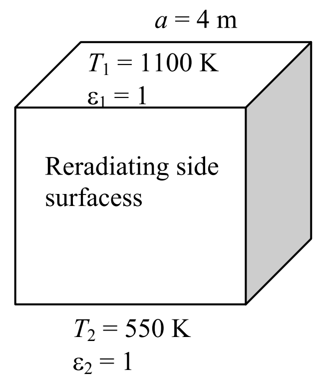
15-79 The floor and the ceiling of a cubical furnace are maintained at uniform temperatures. The net rate of radiation heat transfer between the floor and the ceiling is to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

Properties The emissivities of all surfaces are $\varepsilon = 1$ since they are black or reradiating.

Analysis We consider the ceiling to be surface 1, the floor to be surface 2 and the side surfaces to be surface 3. The furnace can be considered to be three-surface enclosure. We assume that steady-state conditions exist. Since the side surfaces are reradiating, there is no heat transfer through them, and the entire heat lost by the ceiling must be gained by the floor. The view factor from the ceiling to the floor of the furnace is $F_{12} = 0.2$. Then the rate of heat loss from the ceiling can be determined from

$$\dot{Q}_1 = \frac{E_{b1} - E_{b2}}{\left(\dfrac{1}{R_{12}} + \dfrac{1}{R_{13} + R_{23}}\right)^{-1}}$$



where

$$E_{b1} = \sigma T_1^{\,4} = (5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1100\ \text{K})^4 = 83{,}015\ \text{W/m}^2$$

$$E_{b2} = \sigma T_2^{\,4} = (5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(550\ \text{K})^4 = 5188\ \text{W/m}^2$$

and

$$A_1 = A_2 = (4\ \text{m})^2 = 16\ \text{m}^2$$

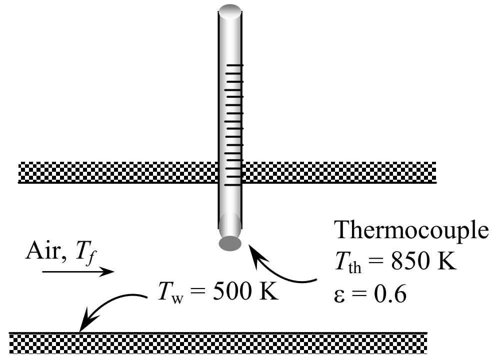
$$R_{12} = \frac{1}{A_1 F_{12}} = \frac{1}{(16\ \text{m}^2)(0.2)} = 0.3125\ \text{m}^{-2}$$

$$R_{13} = R_{23} = \frac{1}{A_1 F_{13}} = \frac{1}{(16\ \text{m}^2)(0.8)} = 0.078125\ \text{m}^{-2}$$

Substituting,

$$\dot{Q}_{12} = \frac{(83{,}015 - 5188)\ \text{W/m}^2}{\left(\dfrac{1}{0.3125\ \text{m}^{-2}} + \dfrac{1}{2(0.078125\ \text{m}^{-2})}\right)^{-1}} = 7.47\times10^5\ \text{W} = \textbf{747 kW}$$

15-96 The temperature of air in a duct is measured by a thermocouple. The radiation effect on the temperature measurement is to be quantified, and the actual air temperature is to be determined.



Assumptions The surfaces are opaque, diffuse, and gray.

Properties The emissivity of thermocouple is given to be ε=0.6.

Analysis The actual temperature of the air can be determined from

$$T_f = T_{th} + \frac{\varepsilon_{th}\sigma(T_{th}^4 - T_w^4)}{h}$$

$$= 850\text{ K} + \frac{(0.6)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)[(850\text{ K})^4 - (500\text{ K})^4]}{60\text{ W/m}^2\cdot°\text{C}}$$

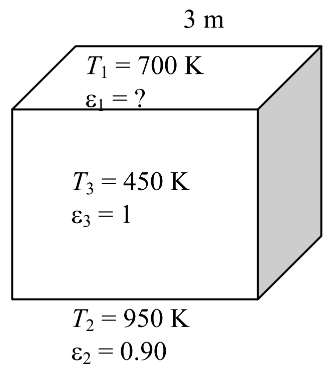
$$= \mathbf{1111K}$$

15-107 A cylindrical furnace with specified top and bottom surface temperatures and specified heat transfer rate at the bottom surface is considered. The emissivity of the top surface and the net rates of heat transfer between the top and the bottom surfaces, and between the bottom and the side surfaces are to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

Properties The emissivity of the bottom surface is 0.90.

Analysis We consider the top surface to be surface 1, the base surface to be surface 2, and the side surface to be surface 3. This system is a three-surface enclosure. The view factor from the base to the top surface of the cube is from Fig. 15-5 $F_{12} = 0.2$. The view factor from the base or the top to the side surfaces is determined by applying the summation rule to be



$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow F_{13} = 1 - F_{12} = 1 - 0.2 = 0.8$$

since the base surface is flat and thus $F_{11} = 0$. Other view factors are

$$F_{21} = F_{12} = 0.20, \qquad F_{23} = F_{13} = 0.80, \qquad F_{31} = F_{32} = 0.20$$

We now apply Eq. 9-35 to each surface

Surface 1:

$$\sigma T_1^{\,4} = J_1 + \frac{1-\varepsilon_1}{\varepsilon_1}\left[F_{12}(J_1 - J_2) + F_{13}(J_1 - J_3)\right]$$

$$(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)(700\text{ K})^4 = J_1 + \frac{1-\varepsilon_1}{\varepsilon_1}\left[0.20(J_1 - J_2) + 0.80(J_1 - J_3)\right]$$

Surface 2:

$$\sigma T_2^{\,4} = J_2 + \frac{1-\varepsilon_2}{\varepsilon_2}\left[F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)\right]$$

$$(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)(950\text{ K})^4 = J_2 + \frac{1-0.90}{0.90}\left[0.20(J_2 - J_1) + 0.80(J_2 - J_3)\right]$$

Surface 3:

$$\sigma T_3^{\,4} = J_3$$

$$(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)(450\text{ K})^4 = J_3$$

We now apply Eq. 9-34 to surface 2

$$\dot{Q}_2 = A_2\left[F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)\right] = (9\text{ m}^2)\left[0.20(J_2 - J_1) + 0.80(J_2 - J_3)\right]$$

Solving the above four equations, we find

$$\varepsilon_1 = \mathbf{0.44}, \quad J_1 = 11{,}736\text{ W/m}^2, \quad J_2 = 41{,}985\text{ W/m}^2, \quad J_3 = 2325\text{ W/m}^2$$

The rate of heat transfer between the bottom and the top surface is

$$A_1 = A_2 = (3\text{ m})^2 = 9\text{ m}^2$$

$$\dot{Q}_{21} = A_2 F_{21}(J_2 - J_1) = (9\text{ m}^2)(0.20)(41{,}985 - 11{,}736)\text{W/m}^2 = \mathbf{54.4\ kW}$$

The rate of heat transfer between the bottom and the side surface is

$$A_3 = 4A_1 = 4(9\text{ m}^2) = 36\text{ m}^2$$

$$\dot{Q}_{23} = A_2 F_{23}(J_2 - J_3) = (9\text{ m}^2)(0.8)(41{,}985 - 2325)\text{W/m}^2 = \mathbf{285.6\ kW}$$

Discussion The sum of these two heat transfer rates are 54.4 + 285.6 = 340 kW, which is equal to 340 kW heat supply rate from surface 2.

15-108 Radiation heat transfer occurs between two square parallel plates. The view factors, the rate of radiation heat transfer and the temperature of a third plate to be inserted are to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

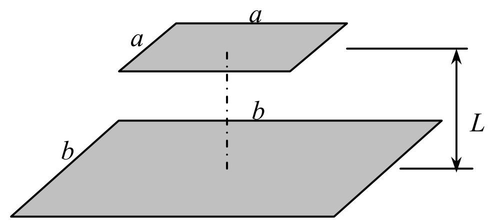
Properties The emissivities of plate a, b, and c are given to be $\varepsilon_a = 0.8$, $\varepsilon_b = 0.4$, and $\varepsilon_c = 0.1$, respectively.

Analysis (*a*) The view factor from surface a to surface b is determined as follows

$$A = \frac{a}{L} = \frac{20}{40} = 0.5,\ \ B = \frac{b}{L} = \frac{60}{40} = 1.5$$

$$F_{ab} = \frac{1}{2A}\left\{\left[(B+A)^2+4\right]^{0.5} - \left[(B-A)^2+4\right]^{0.5}\right\} = \frac{1}{2(0.5)}\left\{\left[(1.5+0.5)^2+4\right]^{0.5} - \left[(1.5-0.5)^2+4\right]^{0.5}\right\} = \mathbf{0.592}$$

The view factor from surface b to surface a is determined from reciprocity relation:



$$A_a = (0.2\text{ m})(0.2\text{ m}) = 0.04\text{ m}^2$$

$$A_b = (0.6\text{ m})(0.6\text{ m}) = 0.36\text{ m}^2$$

$$A_a F_{ab} = A_b F_{ba}$$

$$(0.04)(0.592) = (0.36)F_{ba} \longrightarrow F_{ba} = \mathbf{0.0658}$$

(b) The net rate of radiation heat transfer between the surfaces can be determined from

$$\dot{Q}_{ab} = \frac{\sigma\left(T_a^{\,4} - T_b^{\,4}\right)}{\dfrac{1-\varepsilon_a}{A_a\varepsilon_a} + \dfrac{1}{A_a F_{ab}} + \dfrac{1-\varepsilon_b}{A_b\varepsilon_b}} = \frac{(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)\left[(1073\text{ K})^4 - (473\text{ K})^4\right]}{\dfrac{1-0.8}{(0.04\text{ m}^2)(0.8)} + \dfrac{1}{(0.04\text{ m}^2)(0.592)} + \dfrac{1-0.4}{(0.36\text{ m}^2)(0.4)}} = \mathbf{1374\text{ W}}$$

(c) In this case we have

$$A = \frac{a}{L} = \frac{0.2\text{ m}}{0.2\text{ m}} = 1,\ \ C = \frac{c}{L} = \frac{2.0\text{ m}}{0.2\text{ m}} = 10$$

$$F_{ac} = \frac{1}{2A}\left\{\left[(C+A)^2+4\right]^{0.5} - \left[(C-A)^2+4\right]^{0.5}\right\} = \frac{1}{2(0.5)}\left\{\left[(10+0.5)^2+4\right]^{0.5} - \left[(10-0.5)^2+4\right]^{0.5}\right\} = 0.981$$

$$B = \frac{b}{L} = \frac{0.6\text{ m}}{0.2\text{ m}} = 3,\ \ C = \frac{c}{L} = \frac{2.0\text{ m}}{0.2\text{ m}} = 10$$

$$F_{bc} = \frac{1}{2A}\left\{\left[(C+B)^2+4\right]^{0.5} - \left[(C-B)^2+4\right]^{0.5}\right\}$$

$$= \frac{1}{2(3)}\left\{\left[(10+3)^2+4\right]^{0.5} - \left[(10-3)^2+4\right]^{0.5}\right\} = 0.979$$

$$A_b F_{bc} = A_c F_{cb}$$

$$(0.36)(0.979) = (4.0)F_{cb} \longrightarrow F_{ba} = 0.0881$$

An energy balance gives

$$\dot{Q}_{ac} = \dot{Q}_{cb}$$

$$\frac{\sigma\left(T_a^{\,4} - T_c^{\,4}\right)}{\dfrac{1-\varepsilon_a}{A_a\varepsilon_a} + \dfrac{1}{A_a F_{ac}} + \dfrac{1-\varepsilon_c}{A_c\varepsilon_c}} = \frac{\sigma\left(T_c^{\,4} - T_b^{\,4}\right)}{\dfrac{1-\varepsilon_c}{A_c\varepsilon_c} + \dfrac{1}{A_c F_{cb}} + \dfrac{1-\varepsilon_b}{A_b\varepsilon_b}}$$

$$\frac{(1073\text{ K})^4 - T_c^{\,4}}{\dfrac{1-0.8}{(0.04\text{ m}^2)(0.8)} + \dfrac{1}{(0.04\text{ m}^2)(0.981)} + \dfrac{1-0.1}{(4\text{ m}^2)(0.1)}} = \frac{T_c^{\,4} - (473\text{ K})^4}{\dfrac{1-0.1}{(4\text{ m}^2)(0.1)} + \dfrac{1}{(4\text{ m}^2)(0.0881)} + \dfrac{1-0.4}{(0.36\text{ m}^2)(0.4)}}$$

Solving the equation with an equation solver such as EES, we obtain $T_c = 754$ K **= 481°C**