Chapter 5

MASS AND ENERGY ANALYSIS OF CONTROL VOLUMES

Conservation of Mass

5-1C Mass, energy, momentum, and electric charge are conserved, and volume and entropy are not conserved during a process.

5-2C Mass flow rate is the amount of mass flowing through a cross-section per unit time whereas the volume flow rate is the amount of volume flowing through a cross-section per unit time.

5-3C The amount of mass or energy entering a control volume does not have to be equal to the amount of mass or energy leaving during an unsteady-flow process.

5-4C Flow through a control volume is steady when it involves no changes with time at any specified position.

5-5C No, a flow with the same volume flow rate at the inlet and the exit is not necessarily steady (unless the density is constant). To be steady, the mass flow rate through the device must remain constant.

5-6E A garden hose is used to fill a water bucket. The volume and mass flow rates of water, the filling time, and the discharge velocity are to be determined.

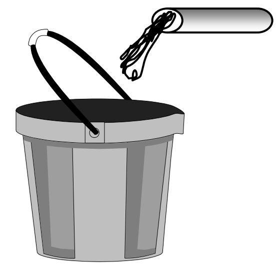
Assumptions **1** Water is an incompressible substance. **2** Flow through the hose is steady. **3** There is no waste of water by splashing.

Properties We take the density of water to be 62.4 lbm/ft^3 (Table A-3E).

Analysis (*a*) The volume and mass flow rates of water are

$$\dot{V} = AV = (\pi D^2/4)V = [\pi(1/12\text{ ft})^2/4](8\text{ ft/s}) = \mathbf{0.04363\ ft^3/s}$$

$$\dot{m} = \rho\dot{V} = (62.4\text{ lbm/ft}^3)(0.04363\text{ ft}^3/\text{s}) = \mathbf{2.72\ lbm/s}$$



(*b*) The time it takes to fill a 20-gallon bucket is

$$\Delta t = \frac{V}{\dot{V}} = \frac{20\text{ gal}}{0.04363\text{ ft}^3/\text{s}}\left(\frac{1\text{ ft}^3}{7.4804\text{ gal}}\right) = \mathbf{61.3\ s}$$

(*c*) The average discharge velocity of water at the nozzle exit is

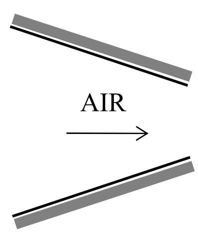
$$V_e = \frac{\dot{V}}{A_e} = \frac{\dot{V}}{\pi D_e^2/4} = \frac{0.04363\text{ ft}^3/\text{s}}{[\pi(0.5/12\text{ ft})^2/4]} = \mathbf{32\ ft/s}$$

Discussion Note that for a given flow rate, the average velocity is inversely proportional to the square of the velocity. Therefore, when the diameter is reduced by half, the velocity quadruples.

5-7 Air is accelerated in a nozzle. The mass flow rate and the exit area of the nozzle are to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 2.21 kg/m^3 at the inlet, and 0.762 kg/m^3 at the exit.



Analysis (a) The mass flow rate of air is determined from the inlet conditions to be

$$\dot{m} = \rho_1 A_1 V_1 = (2.21\text{ kg/m}^3)(0.009\text{ m}^2)(40\text{ m/s}) = \mathbf{0.796\ kg/s}$$

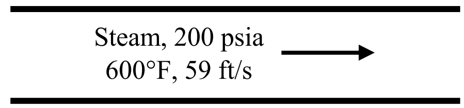
(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.
Then the exit area of the nozzle is determined to be

$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.796\text{ kg/s}}{(0.762\text{ kg/m}^3)(180\text{ m/s})} = 0.0058\text{ m}^2 = 58\ \mathbf{cm^2}$$

5-8E Steam flows in a pipe. The minimum diameter of the pipe for a given steam velocity is to be determined.

Assumptions Flow through the pipe is steady.

Properties The specific volume of steam at the given state is (Table A-6E)



$$\left.\begin{array}{l} P = 200 \text{ psia} \\ T = 600°\text{F} \end{array}\right\} \boldsymbol{v}_1 = 3.0586 \text{ ft}^3/\text{lbm}$$

Analysis The cross sectional area of the pipe is

$$\dot{m} = \frac{1}{\boldsymbol{v}} A_c V \longrightarrow A = \frac{\dot{m}\boldsymbol{v}}{V} = \frac{(200 \text{ lbm/s})(3.0586 \text{ ft}^3/\text{lbm})}{59 \text{ ft/s}} = 10.37 \text{ ft}^2$$

Solving for the pipe diameter gives

$$A = \frac{\pi D^2}{4} \longrightarrow D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(10.37 \text{ ft}^2)}{\pi}} = \mathbf{3.63 \text{ ft}}$$

Therefore, the diameter of the pipe must be at least 3.63 ft to ensure that the velocity does not exceed 59 ft/s.

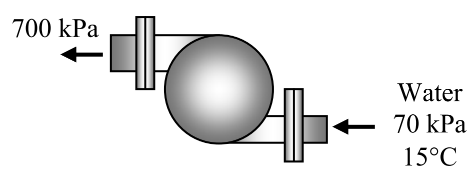
5-9 A water pump increases water pressure. The diameters of the inlet and exit openings are given. The velocity of the water at the inlet and outlet are to be determined.

Assumptions **1** Flow through the pump is steady. **2** The specific volume remains constant.

Properties The inlet state of water is compressed liquid. We approximate it as a saturated liquid at the given temperature. Then, at 15°C and 40°C, we have (Table A-4)

$$\left.\begin{array}{l} T = 15°\text{C} \\ x = 0 \end{array}\right\} \boldsymbol{v}_1 = 0.001001 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} T = 40°\text{C} \\ x = 0 \end{array}\right\} \boldsymbol{v}_1 = 0.001008 \text{ m}^3/\text{kg}$$



Analysis The velocity of the water at the inlet is

$$V_1 = \frac{\dot{m}\boldsymbol{v}_1}{A_1} = \frac{4\dot{m}\boldsymbol{v}_1}{\pi D_1^2} = \frac{4(0.5 \text{ kg/s})(0.001001 \text{ m}^3/\text{kg})}{\pi(0.01 \text{ m})^2} = \mathbf{6.37 \text{ m/s}}$$

Since the mass flow rate and the specific volume remains constant, the velocity at the pump exit is

$$V_2 = V_1\frac{A_1}{A_2} = V_1\left(\frac{D_1}{D_2}\right)^2 = (6.37 \text{ m/s})\left(\frac{0.01 \text{ m}}{0.015 \text{ m}}\right)^2 = \mathbf{2.83 \text{ m/s}}$$

Using the specific volume at 40°C, the water velocity at the inlet becomes

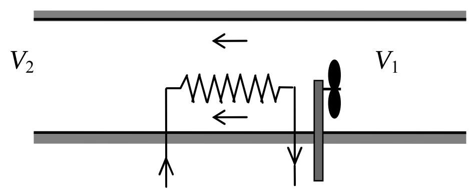
$$V_1 = \frac{\dot{m}\boldsymbol{v}_1}{A_1} = \frac{4\dot{m}\boldsymbol{v}_1}{\pi D_1^2} = \frac{4(0.5 \text{ kg/s})(0.001008 \text{ m}^3/\text{kg})}{\pi(0.01 \text{ m})^2} = \mathbf{6.42 \text{ m/s}}$$

which is a 0.8% increase in velocity.

5-10 Air is expanded and is accelerated as it is heated by a hair dryer of constant diameter. The percent increase in the velocity of air as it flows through the drier is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 1.20 kg/m^3 at the inlet, and 1.05 kg/m^3 at the exit.



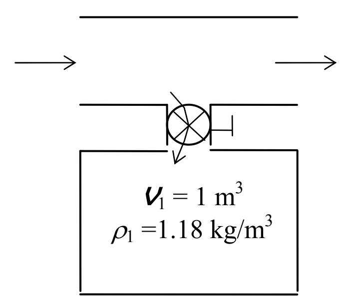
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\dot{m}_1 = \dot{m}_2$$
$$\rho_1 A V_1 = \rho_2 A V_2$$
$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20 \text{ kg/m}^3}{1.05 \text{ kg/m}^3} = 1.14 \quad \text{(or, and increase of } \mathbf{14\%}\text{)}$$

Therefore, the air velocity increases 14% as it flows through the hair drier.

5-11 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until the density rises to a specified level. The mass of air that entered the tank is to be determined.

Properties The density of air is given to be 1.18 kg/m^3 at the beginning, and 7.20 kg/m^3 at the end.



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. The mass balance for this system can be expressed as

Mass balance:

$$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1 = \rho_2 \boldsymbol{V} - \rho_1 \boldsymbol{V}$$

Substituting,

$$m_i = (\rho_2 - \rho_1)\boldsymbol{V} = [(7.20\text{-}1.18) \text{ kg/m}^3](1 \text{ m}^3) = \mathbf{6.02 \text{ kg}}$$

Therefore, 6.02 kg of mass entered the tank.

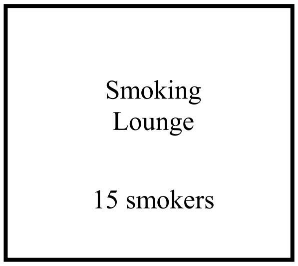
5-12 A smoking lounge that can accommodate 15 smokers is considered. The required minimum flow rate of air that needs to be supplied to the lounge and the diameter of the duct are to be determined.

Assumptions Infiltration of air into the smoking lounge is negligible.

Properties The minimum fresh air requirements for a smoking lounge is given to be 30 L/s per person.

Analysis The required minimum flow rate of air that needs to be supplied to the lounge is determined directly from

$$\dot{V}_{\text{air}} = \dot{V}_{\text{air per person}}(\text{No. of persons})$$
$$= (30\ \text{L/s}\cdot\text{person})(15\ \text{persons}) = 450\ \text{L/s} = \mathbf{0.45\ m^3/s}$$



The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2/4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.45\ \text{m}^3/\text{s})}{\pi(8\ \text{m/s})}} = \mathbf{0.268\ m}$$

Therefore, the diameter of the fresh air duct should be at least 26.8 cm if the velocity of air is not to exceed 8 m/s.

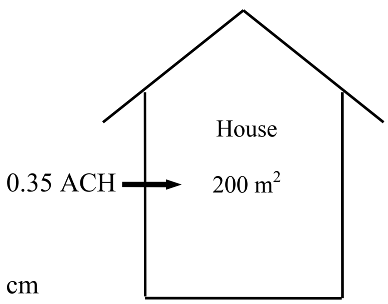
5-13 The minimum fresh air requirements of a residential building is specified to be 0.35 air changes per hour. The size of the fan that needs to be installed and the diameter of the duct are to be determined.

Analysis The volume of the building and the required minimum volume flow rate of fresh air are

$$V_{\text{room}} = (2.7\ \text{m})(200\ \text{m}^2) = 540\ \text{m}^3$$
$$\dot{V} = V_{\text{room}} \times \text{ACH} = (540\ \text{m}^3)(0.35/\text{h}) = 189\ \text{m}^3/\text{h} = 189{,}000\ \text{L/h} = \mathbf{3150\ \ L/min}$$

The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2/4)$$



Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(189/3600\ \text{m}^3/\text{s})}{\pi(6\ \text{m/s})}} = \mathbf{0.106\ m}$$

Therefore, the diameter of the fresh air duct should be at least 10.6 cm if the velocity of air is not to exceed 6 m/s.

5-14 A cyclone separator is used to remove fine solid particles that are suspended in a gas stream. The mass flow rates at the two outlets and the amount of fly ash collected per year are to be determined.

Assumptions Flow through the separator is steady.

Analysis Since the ash particles cannot be converted into the gas and vice-versa, the mass flow rate of ash into the control volume must equal that going out, and the mass flow rate of flue gas into the control volume must equal that going out. Hence, the mass flow rate of ash leaving is

$$\dot{m}_{\text{ash}} = y_{\text{ash}}\dot{m}_{\text{in}} = (0.001)(10\text{ kg/s}) = \mathbf{0.01\ kg/s}$$

The mass flow rate of flue gas leaving the separator is then

$$\dot{m}_{\text{flue gas}} = \dot{m}_{\text{in}} - \dot{m}_{\text{ash}} = 10 - 0.01 = \mathbf{9.99\ kg/s}$$

The amount of fly ash collected per year is

$$m_{\text{ash}} = \dot{m}_{\text{ash}}\Delta t = (0.01\text{ kg/s})(365\times 24\times 3600\text{ s/year}) = \mathbf{315{,}400\ kg/year}$$

5-15 Air flows through an aircraft engine. The volume flow rate at the inlet and the mass flow rate at the exit are to be determined.

Assumptions **1** Air is an ideal gas. **2** The flow is steady.

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

Analysis The inlet volume flow rate is

$$\dot{V}_1 = A_1 V_1 = (1\text{ m}^2)(180\text{ m/s}) = \mathbf{180\ m^3/s}$$

The specific volume at the inlet is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})}{100\text{ kPa}} = 0.8409\text{ m}^3/\text{kg}$$

Since the flow is steady, the mass flow rate remains constant during the flow. Then,

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{180\text{ m}^3/\text{s}}{0.8409\text{ m}^3/\text{kg}} = \mathbf{214.1\ kg/s}$$

5-16 A spherical hot-air balloon is considered. The time it takes to inflate the balloon is to be determined.

Assumptions **1** Air is an ideal gas.

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

Analysis The specific volume of air entering the balloon is

$$v = \frac{RT}{P} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(35+273\ \text{K})}{120\ \text{kPa}} = 0.7366\ \text{m}^3/\text{kg}$$

The mass flow rate at this entrance is

$$\dot{m} = \frac{A_c V}{v} = \frac{\pi D^2}{4}\frac{V}{v} = \frac{\pi(1\ \text{m})^2}{4}\frac{2\ \text{m/s}}{0.7366\ \text{m}^3/\text{kg}} = 2.132\ \text{kg/s}$$

The initial mass of the air in the balloon is

$$m_i = \frac{V_i}{v} = \frac{\pi D^3}{6v} = \frac{\pi(3\ \text{m})^3}{6(0.7366\ \text{m}^3/\text{kg})} = 19.19\ \text{kg}$$

Similarly, the final mass of air in the balloon is

$$m_f = \frac{V_f}{v} = \frac{\pi D^3}{6v} = \frac{\pi(15\ \text{m})^3}{6(0.7366\ \text{m}^3/\text{kg})} = 2399\ \text{kg}$$

The time it takes to inflate the balloon is determined from

$$\Delta t = \frac{m_f - m_i}{\dot{m}} = \frac{(2399-19.19)\ \text{kg}}{2.132\ \text{kg/s}} = 1116\ \text{s} = \mathbf{18.6\ min}$$

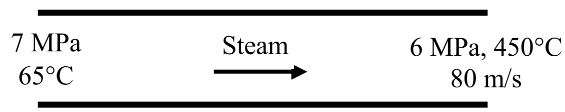
5-17 Water flows through the tubes of a boiler. The velocity and volume flow rate of the water at the inlet are to be determined.

Assumptions Flow through the boiler is steady.

Properties The specific volumes of water at the inlet and exit are (Tables A-6 and A-7)

$$\left.\begin{aligned} P_1 &= 7\text{ MPa} \\ T_1 &= 65°\text{C} \end{aligned}\right\} \boldsymbol{v}_1 = 0.001017\text{ m}^3/\text{kg}$$

$$\left.\begin{aligned} P_2 &= 6\text{ MPa} \\ T_2 &= 450°\text{C} \end{aligned}\right\} \boldsymbol{v}_2 = 0.05217\text{ m}^3/\text{kg}$$



Analysis The cross-sectional area of the tube is

$$A_c = \frac{\pi D^2}{4} = \frac{\pi(0.13\text{ m})^2}{4} = 0.01327\text{ m}^2$$

The mass flow rate through the tube is same at the inlet and exit. It may be determined from exit data to be

$$\dot{m} = \frac{A_c V_2}{\boldsymbol{v}_2} = \frac{(0.01327\text{ m}^2)(80\text{ m/s})}{0.05217\text{ m}^3/\text{kg}} = 20.35\text{ kg/s}$$

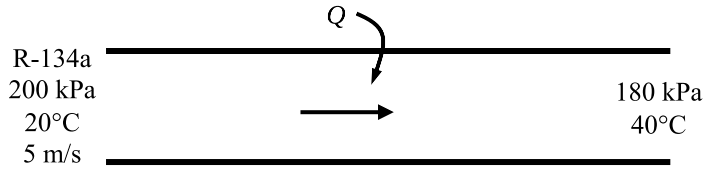
The water velocity at the inlet is then

$$V_1 = \frac{\dot{m}\boldsymbol{v}_1}{A_c} = \frac{(20.35\text{ kg/s})(0.001017\text{ m}^3/\text{kg})}{0.01327\text{ m}^2} = \mathbf{1.560\ m/s}$$

The volumetric flow rate at the inlet is

$$\dot{\boldsymbol{V}}_1 = A_c V_1 = (0.01327\text{ m}^2)(1.560\text{ m/s}) = \mathbf{0.0207\ m^3/s}$$

5-18 Refrigerant-134a flows through a pipe. Heat is supplied to R-134a. The volume flow rates of air at the inlet and exit, the mass flow rate, and the velocity at the exit are to be determined.



Properties The specific volumes of R-134a at the inlet and exit are (Table A-13)

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \boldsymbol{v}_1 = 0.1142\text{ m}^3/\text{kg} \qquad \left.\begin{array}{l} P_1 = 180\text{ kPa} \\ T_1 = 40°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.1374\text{ m}^3/\text{kg}$$

Analysis (*a*) (*b*) The volume flow rate at the inlet and the mass flow rate are

$$\dot{V}_1 = A_c V_1 = \frac{\pi D^2}{4} V_1 = \frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.3079\ m^3/s}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_c V_1 = \frac{1}{\boldsymbol{v}_1}\frac{\pi D^2}{4} V_1 = \frac{1}{0.1142\text{ m}^3/\text{kg}}\frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{2.696\ kg/s}$$

(c) Noting that mass flow rate is constant, the volume flow rate and the velocity at the exit of the pipe are determined from

$$\dot{V}_2 = \dot{m}\boldsymbol{v}_2 = (2.696\text{ kg/s})(0.1374\text{ m}^3/\text{kg}) = \mathbf{0.3705\ m^3/s}$$

$$V_2 = \frac{\dot{V}_2}{A_c} = \frac{0.3705\text{ m}^3/\text{s}}{\frac{\pi (0.28\text{ m})^2}{4}} = \mathbf{6.02\ m/s}$$

Flow Work and Energy Transfer by Mass

5-19C Energy can be transferred to or from a control volume as heat, various forms of work, and by mass.

5-20C Flow energy or flow work is the energy needed to push a fluid into or out of a control volume. Fluids at rest do not possess any flow energy.

5-21C Flowing fluids possess flow energy in addition to the forms of energy a fluid at rest possesses. The total energy of a fluid at rest consists of internal, kinetic, and potential energies. The total energy of a flowing fluid consists of internal, kinetic, potential, and flow energies.

5-22E Steam is leaving a pressure cooker at a specified pressure. The velocity, flow rate, the total and flow energies, and the rate of energy transfer by mass are to be determined.

Assumptions **1** The flow is steady, and the initial start-up period is disregarded. **2** The kinetic and potential energies are negligible, and thus they are not considered. **3** Saturation conditions exist within the cooker at all times so that steam leaves the cooker as a saturated vapor at 30 psia.

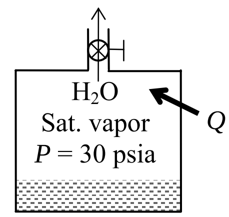
Properties The properties of saturated liquid water and water vapor at 30 psia are v_f = 0.01700 ft^3/lbm, v_g = 13.749 ft^3/lbm, u_g = 1087.8 Btu/lbm, and h_g = 1164.1 Btu/lbm (Table A-5E).

Analysis (*a*) Saturation conditions exist in a pressure cooker at all times after the steady operating conditions are established. Therefore, the liquid has the properties of saturated liquid and the exiting steam has the properties of saturated vapor at the operating pressure. The amount of liquid that has evaporated, the mass flow rate of the exiting steam, and the exit velocity are

$$m = \frac{\Delta V_{\text{liquid}}}{v_f} = \frac{0.4\text{ gal}}{0.01700\text{ ft}^3/\text{lbm}}\left(\frac{0.13368\text{ ft}^3}{1\text{ gal}}\right) = 3.145\text{ lbm}$$

$$\dot{m} = \frac{m}{\Delta t} = \frac{3.145\text{ lbm}}{45\text{ min}} = 0.0699\text{ lbm/min} = \mathbf{1.165\times 10^{-3}\ lbm/s}$$

$$V = \frac{\dot{m}}{\rho_g A_c} = \frac{\dot{m} v_g}{A_c} = \frac{(1.165\times 10^{-3}\text{ lbm/s})(13.749\text{ ft}^3/\text{lbm})}{0.15\text{ in}^2}\left(\frac{144\text{ in}^2}{1\text{ ft}^2}\right) = \mathbf{15.4\ ft/s}$$



(*b*) Noting that $h = u + Pv$ and that the kinetic and potential energies are disregarded, the flow and total energies of the exiting steam are

$$e_{\text{flow}} = Pv = h - u = 1164.1 - 1087.8 = \mathbf{76.3\ Btu/lbm}$$
$$\theta = h + ke + pe \cong h = \mathbf{1164.1\ Btu/lbm}$$

Note that the kinetic energy in this case is ke = $V^2/2$ = (15.4 ft/s)^2 = 237 ft^2/s^2 = 0.0095 Btu/lbm, which is very small compared to enthalpy.

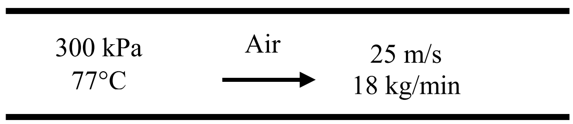
(*c*) The rate at which energy is leaving the cooker by mass is simply the product of the mass flow rate and the total energy of the exiting steam per unit mass,

$$\dot{E}_{\text{mass}} = \dot{m}\theta = (1.165\times 10^{-3}\text{ lbm/s})(1164.1\text{ Btu/lbm}) = \mathbf{1.356\ Btu/s}$$

Discussion The numerical value of the energy leaving the cooker with steam alone does not mean much since this value depends on the reference point selected for enthalpy (it could even be negative). The significant quantity is the difference between the enthalpies of the exiting vapor and the liquid inside (which is h_{fg}) since it relates directly to the amount of energy supplied to the cooker.

5-23 Air flows steadily in a pipe at a specified state. The diameter of the pipe, the rate of flow energy, and the rate of energy transport by mass are to be determined. Also, the error involved in the determination of energy transport by mass is to be determined.

Properties The properties of air are R = 0.287 kJ/kg.K and c_p = 1.008 kJ/kg.K (at 350 K from Table A-2b)



Analysis (*a*) The diameter is determined as follows

$$\boldsymbol{v} = \frac{RT}{P} = \frac{(0.287\ \text{kJ/kg.K})(77+273\ \text{K})}{(300\ \text{kPa})} = 0.3349\ \text{m}^3/\text{kg}$$

$$A = \frac{\dot{m}\boldsymbol{v}}{V} = \frac{(18/60\ \text{kg/s})(0.3349\ \text{m}^3/\text{kg})}{25\ \text{m/s}} = 0.004018\ \text{m}^2$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(0.004018\ \text{m}^2)}{\pi}} = \mathbf{0.0715\ m}$$

(*b*) The rate of flow energy is determined from

$$\dot{W}_{\text{flow}} = \dot{m}P\boldsymbol{v} = (18/60\ \text{kg/s})(300\ \text{kPa})(0.3349\ \text{m}^3/\text{kg}) = \mathbf{30.14\ kW}$$

(*c*) The rate of energy transport by mass is

$$\dot{E}_{\text{mass}} = \dot{m}(h+ke) = \dot{m}\left(c_pT + \frac{1}{2}V^2\right)$$

$$= (18/60\ \text{kg/s})\left[(1.008\ \text{kJ/kg.K})(77+273\ \text{K}) + \frac{1}{2}(25\ \text{m/s})^2\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{105.94\ kW}$$

(*d*) If we neglect kinetic energy in the calculation of energy transport by mass

$$\dot{E}_{\text{mass}} = \dot{m}h = \dot{m}c_pT = (18/60\ \text{kg/s})(1.005\ \text{kJ/kg.K})(77+273\ \text{K}) = 105.84\ \text{kW}$$

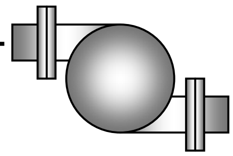
Therefore, the error involved if neglect the kinetic energy is only **0.09%**.

5-24E A water pump increases water pressure. The flow work required by the pump is to be determined.

Assumptions **1** Flow through the pump is steady. **2** The state of water at the pump inlet is saturated liquid. **3** The specific volume remains constant.

Properties The specific volume of saturated liquid water at 10 psia is

$$v = v_{f@10\text{ psia}} = 0.01659\text{ ft}^3/\text{lbm} \quad \text{(Table A-5E)}$$



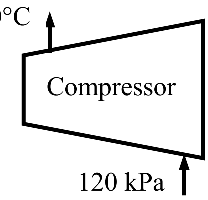
Then the flow work relation gives

$$w_{\text{flow}} = P_2 v_2 - P_1 v_1 = v(P_2 - P_1)$$
$$= (0.01659\text{ ft}^3/\text{lbm})(50-10)\text{psia}\left(\frac{1\text{ Btu}}{5.404\text{ psia}\cdot\text{ft}^3}\right)$$
$$= \mathbf{0.1228\ Btu/lbm}$$

5-25 An air compressor compresses air. The flow work required by the compressor is to be determined.

Assumptions **1** Flow through the compressor is steady. **2** Air is an ideal gas.

Properties Combining the flow work expression with the ideal gas equation of state gives



$$w_{\text{flow}} = P_2 v_2 - P_1 v_1$$
$$= R(T_2 - T_1)$$
$$= (0.287\text{ kJ/kg}\cdot\text{K})(300-20)\text{K}$$
$$= \mathbf{80.36\ kJ/kg}$$

Steady Flow Energy Balance: Nozzles and Diffusers

5-26C A steady-flow system involves no changes with time anywhere within the system or at the system boundaries

5-27C No.

5-28C It is mostly converted to internal energy as shown by a rise in the fluid temperature.

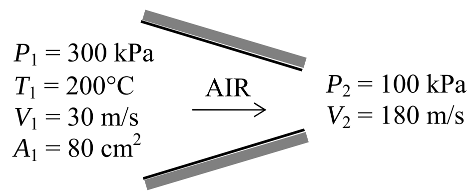
5-29C The kinetic energy of a fluid increases at the expense of the internal energy as evidenced by a decrease in the fluid temperature.

5-30C Heat transfer to the fluid as it flows through a nozzle is desirable since it will probably increase the kinetic energy of the fluid. Heat transfer from the fluid will decrease the exit velocity.

5-31 Air is accelerated in a nozzle from 30 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is c_p = 1.02 kJ/kg.°C (Table A-2).



Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473 \text{ K})}{300 \text{ kPa}} = 0.4525 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 = \frac{1}{0.4525 \text{ m}^3/\text{kg}}(0.008 \text{ m}^2)(30 \text{ m/s}) = \mathbf{0.5304 \text{ kg/s}}$$

(*b*) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,ave}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Substituting, $$0 = (1.02 \text{ kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) + \frac{(180 \text{ m/s})^2 - (30 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

It yields T_2 = **184.6°C**

(*c*) The specific volume of air at the nozzle exit is

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.6 + 273 \text{ K})}{100 \text{ kPa}} = 1.313 \text{ m}^3/\text{kg}$$

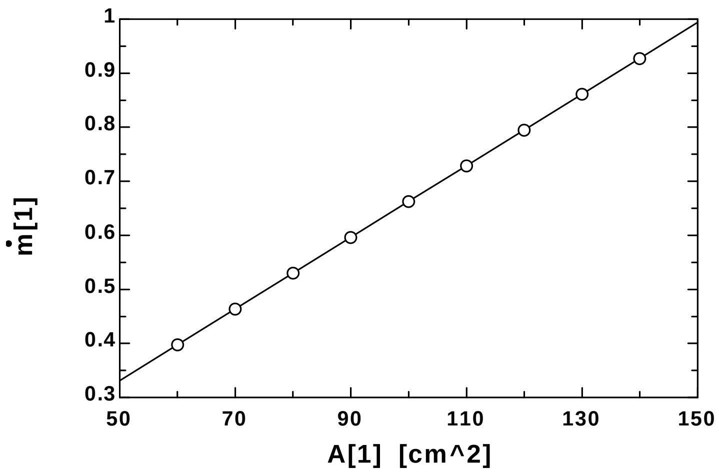
$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 \longrightarrow 0.5304 \text{ kg/s} = \frac{1}{1.313 \text{ m}^3/\text{kg}} A_2 (180 \text{ m/s}) \rightarrow A_2 = 0.00387 \text{ m}^2 = \mathbf{38.7 \text{ cm}^2}$$

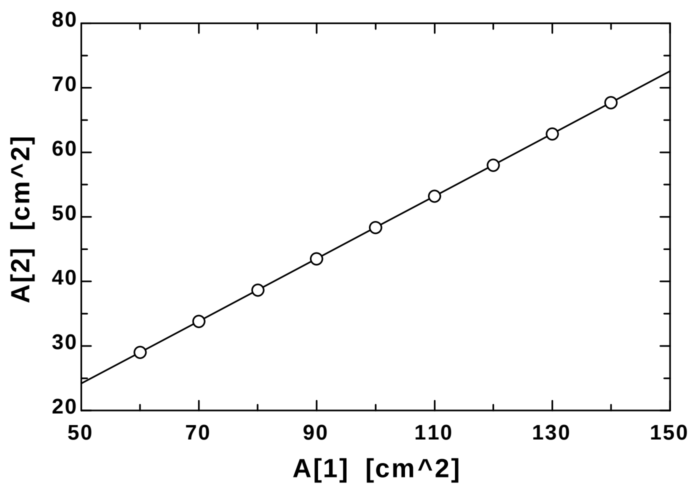
5-32 EES Problem 5-31 is reconsidered. The effect of the inlet area on the mass flow rate, exit velocity, and the exit area as the inlet area varies from 50 cm^2 to 150 cm^2 is to be investigated, and the final results are to be plotted against the inlet area.

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

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'Air' = WorkFluid$  then
        HCal:=ENTHALPY('Air',T=Tx) "Ideal gas equ."
   else
       HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"
"Property relation: Air is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"
"Knowns - obtain from the input diagram"
WorkFluid$ = 'Air'
T[1] = 200  [C]
P[1] = 300   [kPa]
Vel[1] = 30  [m/s]
P[2] = 100  [kPa]
Vel[2] = 180  [m/s]
A[1]=80 [cm^2]
Am[1]=A[1]*convert(cm^2,m^2)

"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

"Conservation of mass: "
m_dot[1]= m_dot[2]
"Mass flow rate"
m_dot[1]=Am[1]*Vel[1]/v[1]
m_dot[2]= Am[2]*Vel[2]/v[2]

"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)

"Definition"
A_ratio=A[1]/A[2]
A[2]=Am[2]*convert(m^2,cm^2)
```

A_1 [cm^2]	A_2 [cm^2]	m_1	T_2
50	24.19	0.3314	184.6
60	29.02	0.3976	184.6
70	33.86	0.4639	184.6
80	38.7	0.5302	184.6
90	43.53	0.5964	184.6
100	48.37	0.6627	184.6
110	53.21	0.729	184.6
120	58.04	0.7952	184.6
130	62.88	0.8615	184.6
140	67.72	0.9278	184.6
150	72.56	0.9941	184.6





5-33E Air is accelerated in an adiabatic nozzle. The velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The nozzle is adiabatic.

Properties The specific heat of air at the average temperature of (700+645)/2=672.5°F is c_p = 0.253 Btu/lbm·R (Table A-2E*b*).

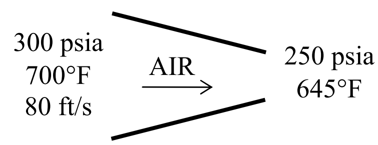
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$



Solving for exit velocity,

$$V_2 = \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} = \left[V_1^2 + 2c_p(T_1 - T_2)\right]^{0.5}$$

$$= \left[(80 \text{ ft/s})^2 + 2(0.253 \text{ Btu/lbm}\cdot\text{R})(700-645)\text{R}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)\right]^{0.5}$$

$$= \mathbf{838.6 \text{ ft/s}}$$

5-34 Air is decelerated in an adiabatic diffuser. The velocity at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The diffuser is adiabatic.

Properties The specific heat of air at the average temperature of (20+90)/2=55°C =328 K is c_p = 1.007 kJ/kg·K (Table A-2*b*).

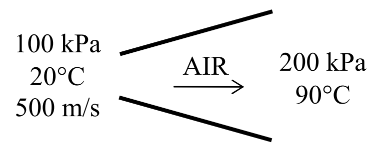
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$



Solving for exit velocity,

$$V_2 = \left[V_1^2 + 2(h_1 - h_2)\right]^{0.5} = \left[V_1^2 + 2c_p(T_1 - T_2)\right]^{0.5}$$

$$= \left[(500\text{ m/s})^2 + 2(1.007\text{ kJ/kg}\cdot\text{K})(20-90)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)\right]^{0.5}$$

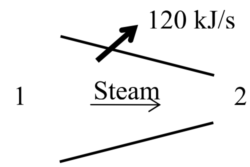
$$= \mathbf{330.2\ m/s}$$

5-35 Steam is accelerated in a nozzle from a velocity of 80 m/s. The mass flow rate, the exit velocity, and the exit area of the nozzle are to be determined.

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

Properties From the steam tables (Table A-6)

$$\left.\begin{array}{l} P_1 = 5 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.057838 \text{ m}^3/\text{kg} \\ h_1 = 3196.7 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 2 \text{ MPa} \\ T_2 = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.12551 \text{ m}^3/\text{kg} \\ h_2 = 3024.2 \text{ kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The mass flow rate of steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} V_1 A_1 = \frac{1}{0.057838 \text{ m}^3/\text{kg}} (80 \text{ m/s})(50 \times 10^{-4} \text{ m}^2) = \mathbf{6.92 \ kg/s}$$

(*b*) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Substituting, the exit velocity of the steam is determined to be

$$-120 \text{ kJ/s} = (6.916 \text{ kg/s})\left(3024.2 - 3196.7 + \frac{V_2^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

It yields $\qquad V_2 = \mathbf{562.7 \ m/s}$

(*c*) The exit area of the nozzle is determined from

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} V_2 A_2 \longrightarrow A_2 = \frac{\dot{m}\boldsymbol{v}_2}{V_2} = \frac{(6.916 \text{ kg/s})(0.12551 \text{ m}^3/\text{kg})}{562.7 \text{ m/s}} = \mathbf{15.42 \times 10^{-4} \ m^2}$$

5-36 [*Also solved by EES on enclosed CD***]** Steam is accelerated in a nozzle from a velocity of 40 m/s to 300 m/s. The exit temperature and the ratio of the inlet-to-exit area of the nozzle are to be determined.

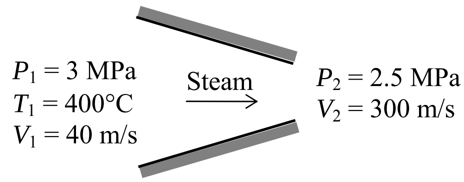
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-6),

$$\left.\begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.09938 \text{ m}^3/\text{kg} \\ h_1 = 3231.7 \text{ kJ/kg} \end{array}$$

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

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



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

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

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 3231.7 \text{ kJ/kg} - \frac{(300 \text{ m/s})^2 - (40 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3187.5 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_2 = 2.5 \text{ MPa} \\ h_2 = 3187.5 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{376.6°C} \\ \boldsymbol{v}_2 = 0.11533 \text{ m}^3/\text{kg} \end{array}$$

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\boldsymbol{v}_1}{\boldsymbol{v}_2}\frac{V_2}{V_1} = \frac{(0.09938 \text{ m}^3/\text{kg})(300 \text{ m/s})}{(0.11533 \text{ m}^3/\text{kg})(40 \text{ m/s})} = \mathbf{6.46}$$

5-37 Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m³/kg.K (Table A-1). The enthalpy of air at the inlet temperature of 400 K is h_1 = 400.98 kJ/kg (Table A-17).

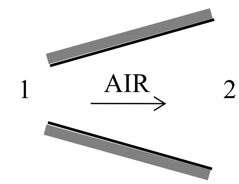
Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$,



or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 400.98 \text{ kJ/kg} - \frac{(30 \text{ m/s})^2 - (230 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 426.98 \text{ kJ/kg}$$

From Table A-17, T_2 = **425.6 K**

(b) The specific volume of air at the diffuser exit is

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(425.6 \text{ K})}{(100 \text{ kPa})} = 1.221 \text{ m}^3/\text{kg}$$

From conservation of mass,

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}\boldsymbol{v}_2}{V_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = \mathbf{0.0678 \ m^2}$$

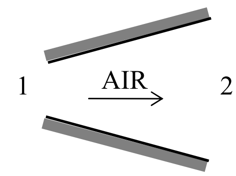
5-38E Air is decelerated in a diffuser from 600 ft/s to a low velocity. The exit temperature and the exit velocity of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The enthalpy of air at the inlet temperature of 20°F is h_1 = 114.69 Btu/lbm (Table A-17E).

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

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



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

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

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2},$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 114.69 \text{ Btu/lbm} - \frac{0 - (600 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 121.88 \text{ Btu/lbm}$$

From Table A-17E, T_2 = **510.0 R**

(*b*) The exit velocity of air is determined from the conservation of mass relation,

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow \frac{1}{RT_2/P_2} A_2 V_2 = \frac{1}{RT_1/P_1} A_1 V_1$$

Thus,

$$V_2 = \frac{A_1 T_2 P_1}{A_2 T_1 P_2} V_1 = \frac{1}{5}\frac{(510 \text{ R})(13 \text{ psia})}{(480 \text{ R})(14.5 \text{ psia})}(600 \text{ ft/s}) = \mathbf{114.3 \text{ ft/s}}$$

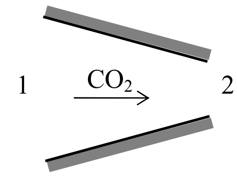
5-39 CO_2 gas is accelerated in a nozzle to 450 m/s. The inlet velocity and the exit temperature are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** CO_2 is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant and molar mass of CO_2 are 0.1889 kPa.m^3/kg.K and 44 kg/kmol (Table A-1). The enthalpy of CO_2 at 500°C is $\bar{h}_1 = 30{,}797$ kJ/kmol (Table A-20).

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume is determined to be

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(773 \text{ K})}{1000 \text{ kPa}} = 0.146 \text{ m}^3/\text{kg}$$



Thus,

$$\dot{m} = \frac{1}{v_1} A_1 V_1 \longrightarrow V_1 = \frac{\dot{m} v_1}{A_1} = \frac{(6000/3600 \text{ kg/s})(0.146 \text{ m}^3/\text{kg})}{40\times10^{-4} \text{m}^2} = \mathbf{60.8 \text{ m/s}}$$

(b) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$\bar{h}_2 = \bar{h}_1 - \frac{V_2^2 - V_1^2}{2} M$$

$$= 30{,}797 \text{ kJ/kmol} - \frac{(450 \text{ m/s})^2 - (60.8 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)(44 \text{ kg/kmol})$$

$$= 26{,}423 \text{ kJ/kmol}$$

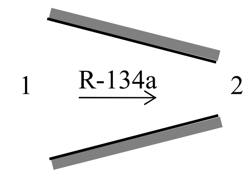
Then the exit temperature of CO_2 from Table A-20 is obtained to be $T_2 = \mathbf{685.8 \text{ K}}$

5-40 R-134a is accelerated in a nozzle from a velocity of 20 m/s. The exit velocity of the refrigerant and the ratio of the inlet-to-exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the refrigerant tables (Table A-13)

$$\left.\begin{array}{l} P_1 = 700 \text{ kPa} \\ T_1 = 120°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.043358 \text{ m}^3/\text{kg} \\ h_1 = 358.90 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 400 \text{ kPa} \\ T_2 = 30°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.056796 \text{ m}^3/\text{kg} \\ h_2 = 275.07 \text{ kJ/kg} \end{array}$$

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

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

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

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

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$0 = (275.07 - 358.90)\text{kJ/kg} + \frac{V_2^2 - (20 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

It yields V_2 **= 409.9 m/s**

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\boldsymbol{v}_1}{\boldsymbol{v}_2}\frac{V_2}{V_1} = \frac{(0.043358 \text{ m}^3/\text{kg})(409.9 \text{ m/s})}{(0.056796 \text{ m}^3/\text{kg})(20 \text{ m/s})} = \mathbf{15.65}$$

5-41 Nitrogen is decelerated in a diffuser from 200 m/s to a lower velocity. The exit velocity of nitrogen and the ratio of the inlet-to-exit area are to be determined.

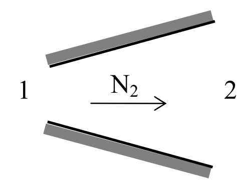
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Nitrogen is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The molar mass of nitrogen is M = 28 kg/kmol (Table A-1). The enthalpies are (Table A-18)

$$T_1 = 7°\text{C} = 280\text{ K} \rightarrow \bar{h}_1 = 8141\text{ kJ/kmol}$$
$$T_2 = 22°\text{C} = 295\text{ K} \rightarrow \bar{h}_2 = 8580\text{ kJ/kmol}$$

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

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



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

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

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = \frac{\bar{h}_2 - \bar{h}_1}{M} + \frac{V_2^2 - V_1^2}{2},$$

Substituting,

$$0 = \frac{(8580 - 8141)\text{ kJ/kmol}}{28\text{ kg/kmol}} + \frac{V_2^2 - (200\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

It yields $\mathbf{V_2 = 93.0\ m/s}$

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2V_2 = \frac{1}{v_1}A_1V_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_1}{v_2}\frac{V_2}{V_1} = \left(\frac{RT_1/P_1}{RT_2/P_2}\right)\frac{V_2}{V_1}$$

or,

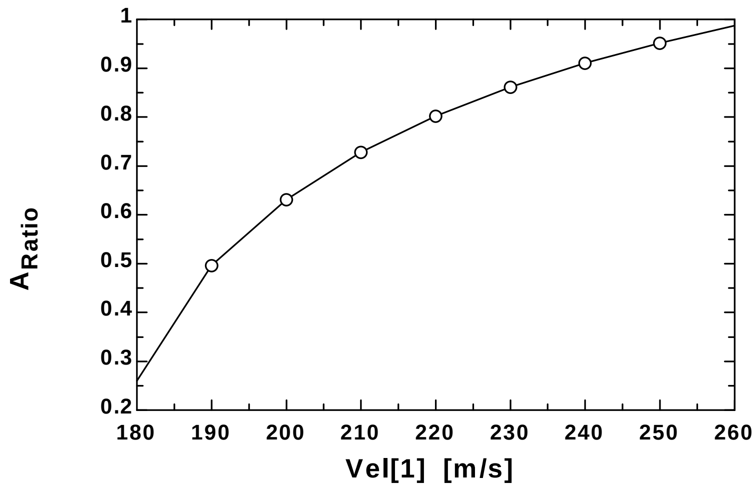
$$\frac{A_1}{A_2} = \left(\frac{T_1/P_1}{T_2/P_2}\right)\frac{V_2}{V_1} = \frac{(280\text{ K}/60\text{ kPa})(93.0\text{ m/s})}{(295\text{ K}/85\text{ kPa})(200\text{ m/s})} = \mathbf{0.625}$$

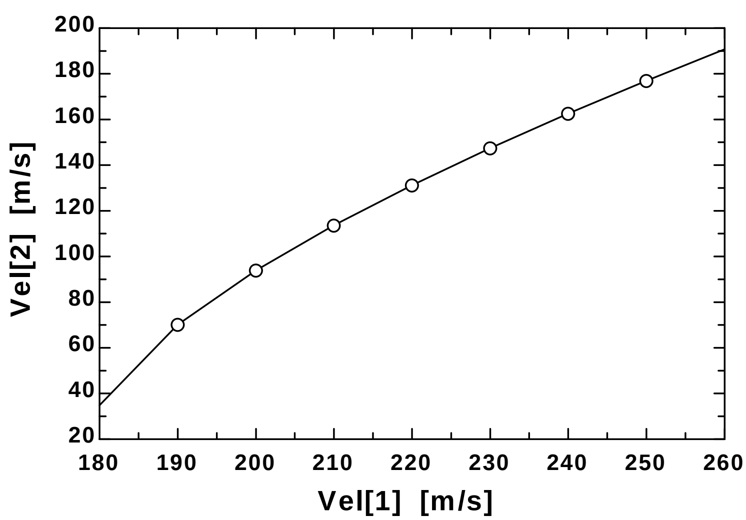
5-42 EES Problem 5-41 is reconsidered. The effect of the inlet velocity on the exit velocity and the ratio of the inlet-to-exit area as the inlet velocity varies from 180 m/s to 260 m/s is to be investigated. The final results are to be plotted against the inlet velocity.

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

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'N2' = WorkFluid$  then
      HCal:=ENTHALPY(WorkFluid$,T=Tx) "Ideal gas equ."
   else
     HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"
"Property relation: Nitrogen is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"

"Knowns"
WorkFluid$ = 'N2'
T[1] = 7  [C]
P[1] = 60   [kPa]
{Vel[1] = 200  [m/s]}
P[2] = 85  [kPa]
T[2] = 22 [C]

"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

"From the definition of mass flow rate, m_dot = A*Vel/v and conservation of mass the area ratio
A_Ratio = A_1/A_2 is:"
A_Ratio*Vel[1]/v[1] =Vel[2]/v[2]
"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)
```

A_{Ratio}	Vel_1 [m/s]	Vel_2 [m/s]
0.2603	180	34.84
0.4961	190	70.1
0.6312	200	93.88
0.7276	210	113.6
0.8019	220	131.2
0.8615	230	147.4
0.9106	240	162.5
0.9518	250	177
0.9869	260	190.8



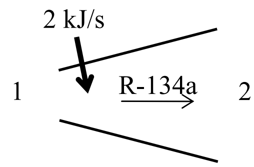


5-43 R-134a is decelerated in a diffuser from a velocity of 120 m/s. The exit velocity of R-134a and the mass flow rate of the R-134a are to be determined.

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

Properties From the R-134a tables (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = 0.025621 \text{ m}^3\text{/kg} \\ h_1 = 267.29 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ T_2 = 40°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.023375 \text{ m}^3\text{/kg} \\ h_2 = 274.17 \text{ kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity of R-134a is determined from the steady-flow mass balance to be

$$\frac{1}{v_2}A_2V_2 = \frac{1}{v_1}A_1V_1 \longrightarrow V_2 = \frac{v_2}{v_1}\frac{A_1}{A_2}V_1 = \frac{1}{1.8}\frac{(0.023375 \text{ m}^3\text{/kg})}{(0.025621 \text{ m}^3\text{/kg})}(120 \text{ m/s}) = \mathbf{60.8 \text{ m/s}}$$

(*b*) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

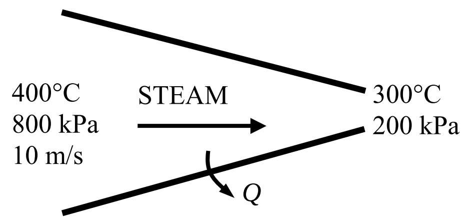
Substituting, the mass flow rate of the refrigerant is determined to be

$$2 \text{ kJ/s} = \dot{m}\left((274.17 - 267.29)\text{kJ/kg} + \frac{(60.8 \text{ m/s})^2 - (120 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

It yields $\dot{m} = \mathbf{1.308 \text{ kg/s}}$

5-44 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:

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

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

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

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

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

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

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

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

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

Turbines and Compressors

5-45C Yes.

5-46C The volume flow rate at the compressor inlet will be greater than that at the compressor exit.

5-47C Yes. Because energy (in the form of shaft work) is being added to the air.

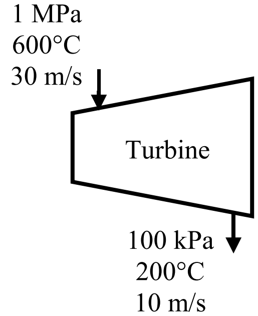
5-48C No.

5-49 Air is expanded in a turbine. The mass flow rate and outlet area are to be determined.

Assumptions **1** Air is an ideal gas. **2** The flow is steady.

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

Analysis The specific volumes of air at the inlet and outlet are



$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(600+273\ \text{K})}{1000\ \text{kPa}} = 0.2506\ \text{m}^3/\text{kg}$$

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(200+273\ \text{K})}{100\ \text{kPa}} = 1.3575\ \text{m}^3/\text{kg}$$

The mass flow rate is

$$\dot{m} = \frac{A_1 V_1}{v_1} = \frac{(0.1\ \text{m}^2)(30\ \text{m/s})}{0.2506\ \text{m}^3/\text{kg}} = \mathbf{11.97\ kg/s}$$

The outlet area is

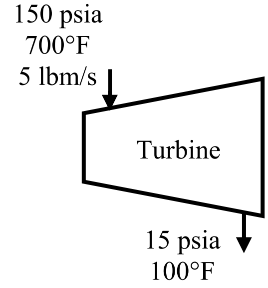
$$A_2 = \frac{\dot{m} v_2}{V_2} = \frac{(11.97\ \text{kg/s})(1.3575\ \text{m}^3/\text{kg})}{10\ \text{m/s}} = \mathbf{1.605\ m^2}$$

5-50E Air is expanded in a gas turbine. The inlet and outlet mass flow rates are to be determined.

Assumptions **1** Air is an ideal gas. **2** The flow is steady.

Properties The gas constant of air is R = 0.3704 psia·ft^3/lbm·R (Table A-1E).

Analysis The specific volumes of air at the inlet and outlet are



$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(700+460\text{ R})}{150\text{ psia}} = 2.864\text{ ft}^3/\text{lbm}$$

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(100+460\text{ R})}{15\text{ psia}} = 13.83\text{ ft}^3/\text{lbm}$$

The volume flow rates at the inlet and exit are then

$$\dot{V}_1 = \dot{m}\boldsymbol{v}_1 = (5\text{ lbm/s})(2.864\text{ ft}^3/\text{lbm}) = \mathbf{14.32\ ft^3/s}$$

$$\dot{V}_2 = \dot{m}\boldsymbol{v}_2 = (5\text{ lbm/s})(13.83\text{ ft}^3/\text{lbm}) = \mathbf{69.15\ ft^3/s}$$

5-51 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).

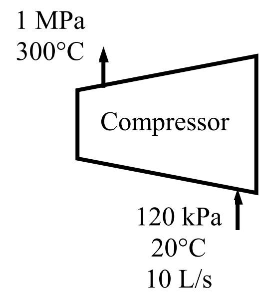
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



Thus,

$$w_{in} = c_p(T_2 - T_1) = (1.018 \text{ kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{285.0 \text{ kJ/kg}}$$

(*b*) The specific volume of air at the inlet and the mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273 \text{ K})}{120 \text{ kPa}} = 0.7008 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.010 \text{ m}^3/\text{s}}{0.7008 \text{ m}^3/\text{kg}} = 0.01427 \text{ kg/s}$$

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

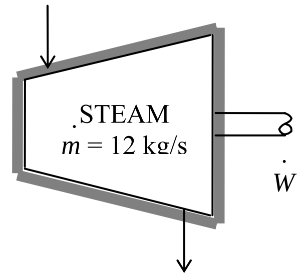
$$\dot{W}_{in} = \dot{m}c_p(T_2 - T_1) = (0.01427 \text{ kg/s})(1.018 \text{ kJ/kg}\cdot\text{K})(300 - 20)\text{K} = \mathbf{4.068 \text{ kW}}$$

5-52 Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 10\text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.029782\text{ m}^3/\text{kg} \\ h_1 = 3242.4\text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 10\text{ kPa} \\ x_2 = 0.92 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.92 \times 2392.1 = 2392.5\text{ kJ/kg}$$

Analysis (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(50\text{ m/s})^2 - (80\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = -1.95\text{ kJ/kg}$$

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

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

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

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

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

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{out} = -(12\text{ kg/s})(2392.5 - 3242.4 - 1.95)\text{kJ/kg} = \mathbf{10.2\ MW}$$

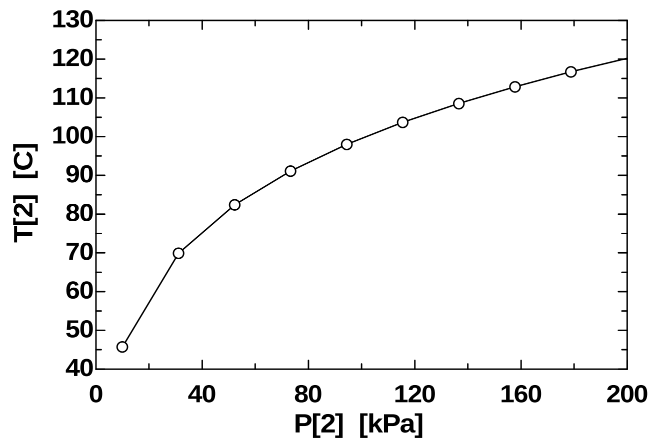
(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{v_1}A_1V_1 \longrightarrow A_1 = \frac{\dot{m}v_1}{V_1} = \frac{(12\text{ kg/s})(0.029782\text{ m}^3/\text{kg})}{80\text{ m/s}} = \mathbf{0.00447\ m^2}$$

5-53 EES Problem 5-52 is reconsidered. The effect of the turbine exit pressure on the power output of the turbine as the exit pressure varies from 10 kPa to 200 kPa is to be investigated. The power output is to be plotted against the exit pressure.

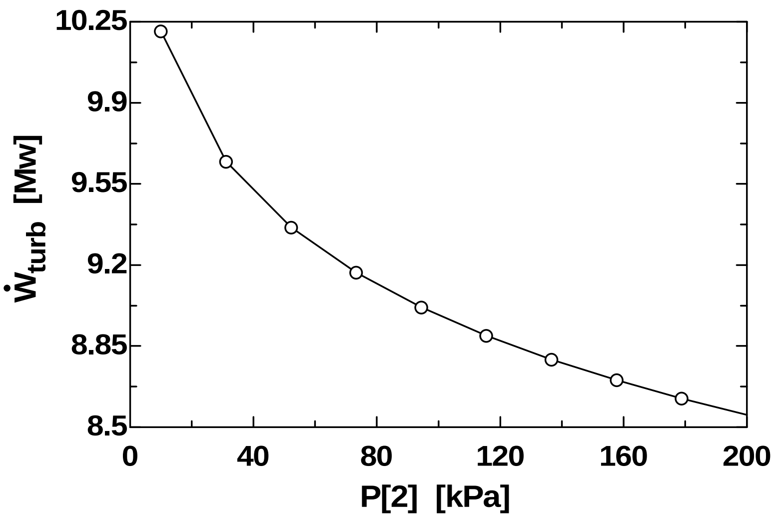
Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns "
T[1] = 450 [C]
P[1] = 10000  [kPa]
Vel[1] = 80  [m/s]
P[2] = 10  [kPa]
X_2=0.92
Vel[2] = 50  [m/s]
m_dot[1]=12 [kg/s]
Fluid$='Steam_IAPWS'

"Property Data"
h[1]=enthalpy(Fluid$,T=T[1],P=P[1])
h[2]=enthalpy(Fluid$,P=P[2],x=x_2)
T[2]=temperature(Fluid$,P=P[2],x=x_2)
v[1]=volume(Fluid$,T=T[1],p=P[1])
v[2]=volume(Fluid$,P=P[2],x=x_2)

"Conservation of mass: "
m_dot[1]= m_dot[2]

"Mass flow rate"
m_dot[1]=A[1]*Vel[1]/v[1]
m_dot[2]= A[2]*Vel[2]/v[2]
```




```
"Conservation of Energy - Steady Flow energy balance"
m_dot[1]*(h[1]+Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)) =
m_dot[2]*(h[2]+Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+W_dot_turb*convert(MW,kJ/s)
DELTAke=Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)-Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)
```

P_2 [kPa]	W_{turb} [MW]	T_2 [C]
10	10.22	45.81
31.11	9.66	69.93
52.22	9.377	82.4
73.33	9.183	91.16
94.44	9.033	98.02
115.6	8.912	103.7
136.7	8.809	108.6
157.8	8.719	112.9
178.9	8.641	116.7
200	8.57	120.2



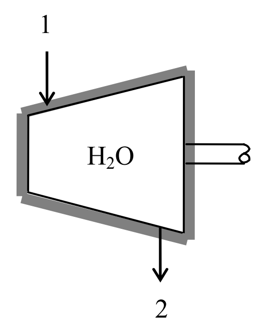
5-54 Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} h_1 = 3375.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 10 \text{ kPa} \\ x_2 = 0.90 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.90 \times 2392.1 = 2344.7 \text{ kJ/kg}$$



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

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

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

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

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

Substituting, the required mass flow rate of the steam is determined to be

$$5000 \text{ kJ/s} = -\dot{m}(2344.7 - 3375.1) \text{ kJ/kg} \longrightarrow \dot{m} = \mathbf{4.852 \text{ kg/s}}$$

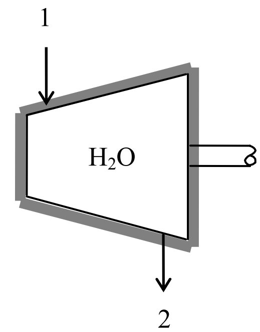
5-55E Steam expands in a turbine. The rate of heat loss from the steam for a power output of 4 MW is to be determined.

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

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

$$\left.\begin{array}{l} P_1 = 1000 \text{ psia} \\ T_1 = 900°\text{F} \end{array}\right\} h_1 = 1448.6 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 5 \text{ psia} \\ sat.vapor \end{array}\right\} h_2 = 1130.7 \text{ Btu/lbm}$$



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

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

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

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

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

Substituting,

$$\dot{Q}_{out} = -(45000/3600 \text{ lbm/s})(1130.7 - 1448.6)\text{Btu/lbm} - 4000 \text{ kJ/s}\left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}}\right) = \mathbf{182.0 \text{ Btu/s}}$$

5-56 Steam expands in a turbine. The exit temperature of the steam for a power output of 2 MW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 8\text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} h_1 = 3399.5\text{ kJ/kg}$$

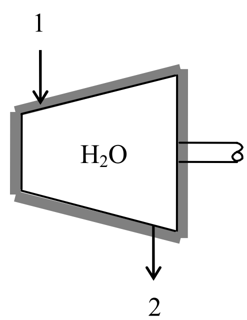
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Substituting,

$$2500\text{ kJ/s} = (3\text{ kg/s})(3399.5 - h_2)\text{kJ/kg}$$

$$h_2 = 2566.2\text{ kJ/kg}$$

Then the exit temperature becomes

$$\left.\begin{array}{l} P_2 = 20\text{ kPa} \\ h_2 = 2566.2\text{ kJ/kg} \end{array}\right\} T_2 = \mathbf{60.1°C}$$

5-57 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

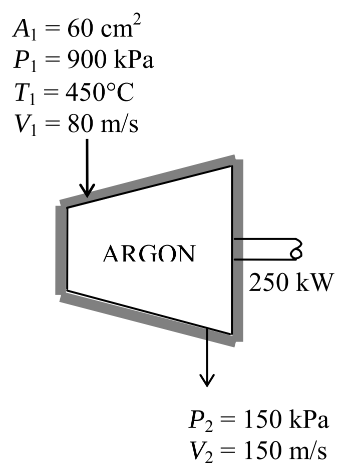
Properties The gas constant of Ar is R = 0.2081 kPa.m^3/kg.K. The constant pressure specific heat of Ar is c_p = 0.5203 kJ/kg·°C (Table A-2a)

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of argon and its mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.2081\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(723\ \text{K})}{900\ \text{kPa}} = 0.167\ \text{m}^3/\text{kg}$$

Thus,

$$\dot{m} = \frac{1}{v_1}A_1V_1 = \frac{1}{0.167\text{m}^3/\text{kg}}(0.006\ \text{m}^2)(80\ \text{m/s}) = 2.874\ \text{kg/s}$$



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

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

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

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

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

Substituting,

$$250\ \text{kJ/s} = -(2.874\ \text{kg/s})\left[(0.5203\ \text{kJ/kg}\cdot^\circ\text{C})(T_2 - 450^\circ\text{C}) + \frac{(150\ \text{m/s})^2 - (80\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

It yields

T_2 = **267.3°C**

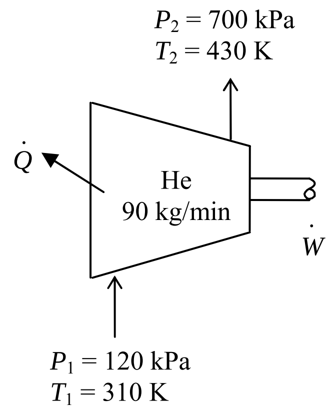
5-58 Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of helium is c_p = 5.1926 kJ/kg·K (Table A-2a).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

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



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

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

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

$$\dot{W}_{in} - \dot{Q}_{out} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

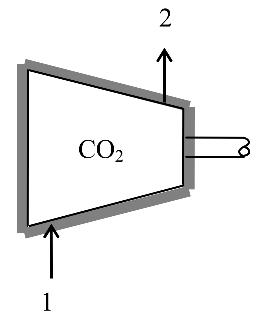
$$\dot{W}_{in} = \dot{Q}_{out} + \dot{m}c_p(T_2 - T_1)$$
$$= (90/60 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg}\cdot\text{K})(430 - 310)\text{K}$$
$$= \mathbf{965\ kW}$$

5-59 CO_2 is compressed by a compressor. The volume flow rate of CO_2 at the compressor inlet and the power input to the compressor are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with variable specific heats. **4** The device is adiabatic and thus heat transfer is negligible.

Properties The gas constant of CO_2 is $R = 0.1889$ kPa.m^3/kg.K, and its molar mass is $M = 44$ kg/kmol (Table A-1). The inlet and exit enthalpies of CO_2 are (Table A-20)

$$T_1 = 300\text{ K} \quad \rightarrow \quad \bar{h}_1 = 9{,}431\text{ kJ/kmol}$$
$$T_2 = 450\text{ K} \quad \rightarrow \quad \bar{h}_2 = 15{,}483\text{ kJ/kmol}$$



Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its volume flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{100\text{ kPa}} = 0.5667\text{ m}^3/\text{kg}$$

$$\dot{\boldsymbol{V}} = \dot{m}\boldsymbol{v}_1 = (0.5\text{ kg/s})(0.5667\text{ m}^3/\text{kg}) = \mathbf{0.283\ m^3/s}$$

(*b*) We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}(\bar{h}_2 - \bar{h}_1)/M$$

Substituting

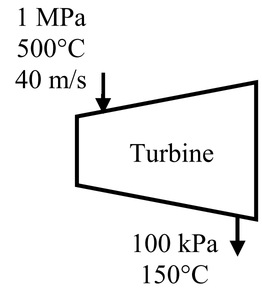
$$\dot{W}_{in} = \frac{(0.5\text{ kg/s})(15{,}483 - 9{,}431\text{ kJ/kmol})}{44\text{ kg/kmol}} = \mathbf{68.8\ kW}$$

5-60 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

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

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

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

5-61 Air is compressed in an adiabatic compressor. The mass flow rate of the air and the power input are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The compressor is adiabatic. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of (20+400)/2=210°C=483 K is c_p = 1.026 kJ/kg·K (Table A-2*b*). The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1).

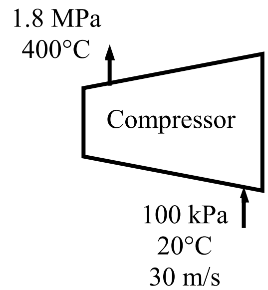
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



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

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\ \text{K})}{100\ \text{kPa}} = 0.8409\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\boldsymbol{v}_1} = \frac{(0.15\ \text{m}^2)(30\ \text{m/s})}{0.8409\ \text{m}^3/\text{kg}} = \mathbf{5.351\ kg/s}$$

Similarly at the outlet,

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400+273\ \text{K})}{1800\ \text{kPa}} = 0.1073\ \text{m}^3/\text{kg}$$

$$V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{(5.351\ \text{kg/s})(0.1073\ \text{m}^3/\text{kg})}{0.08\ \text{m}^2} = 7.177\ \text{m/s}$$

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

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

$$= (5.351\ \text{kg/s})\left[(1.026\ \text{kJ/kg}\cdot\text{K})(400-20)\text{K} + \frac{(7.177\ \text{m/s})^2 - (30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{2084\ kW}$$

5-62E Air is expanded in an adiabatic turbine. The mass flow rate of the air and the power produced are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The turbine is well-insulated, and thus there is no heat transfer. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of (800+250)/2=525°F is c_p = 0.2485 Btu/lbm·R (Table A-2E*b*). The gas constant of air is R = 0.3704 psia·ft^3/lbm·R (Table A-1E).

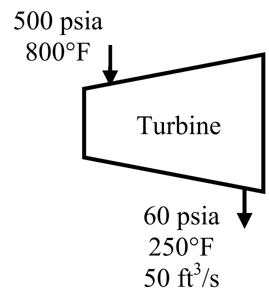
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



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

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(250 + 460 \text{ R})}{60 \text{ psia}} = 4.383 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_2}{\boldsymbol{v}_2} = \frac{50 \text{ ft}^3/\text{s}}{4.383 \text{ ft}^3/\text{lbm}} = 11.41 \text{ kg/s}$$

$$V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{(11.41 \text{ lbm/s})(4.383 \text{ ft}^3/\text{lbm})}{1.2 \text{ ft}^2} = 41.68 \text{ ft/s}$$

Similarly at the inlet,

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(800 + 460 \text{ R})}{500 \text{ psia}} = 0.9334 \text{ ft}^3/\text{lbm}$$

$$V_1 = \frac{\dot{m}\boldsymbol{v}_1}{A_1} = \frac{(11.41 \text{ lbm/s})(0.9334 \text{ ft}^3/\text{lbm})}{0.6 \text{ ft}^2} = 17.75 \text{ ft/s}$$

Substituting into the energy balance equation gives

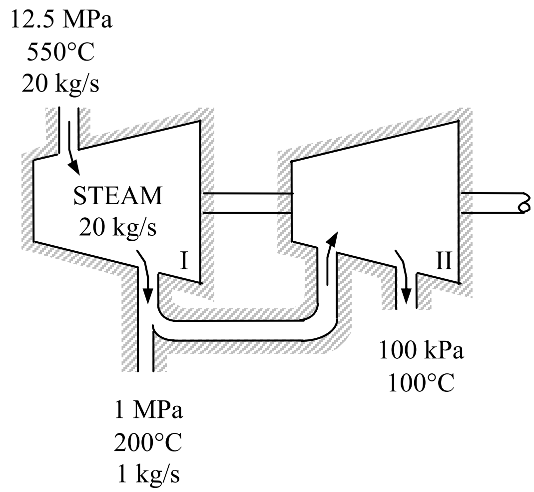
$$\dot{W}_{\text{out}} = \dot{m}\left(c_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right)$$

$$= (11.41 \text{ lbm/s})\left[(0.2485 \text{ Btu/lbm}\cdot\text{R})(800 - 250)\text{R} + \frac{(17.75 \text{ ft/s})^2 - (41.68 \text{ m/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{1559 \text{ kW}}$$

5-63 Steam expands in a two-stage adiabatic turbine from a specified state to another state. Some steam is extracted at the end of the first stage. The power output of the turbine is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible.



Properties From the steam tables (Table A-6)

$$\left.\begin{array}{l} P_1 = 12.5 \text{ MPa} \\ T_1 = 550°\text{C} \end{array}\right\} h_1 = 3476.5 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ T_2 = 200°\text{C} \end{array}\right\} h_2 = 2828.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 100 \text{ kPa} \\ T_3 = 100°\text{C} \end{array}\right\} h_3 = 2675.8 \text{ kJ/kg}$$

Analysis The mass flow rate through the second stage is

$$\dot{m}_3 = \dot{m}_1 - \dot{m}_2 = 20 - 1 = 19 \text{ kg/s}$$

We take the entire turbine, including the connection part between the two stages, as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters the turbine and two fluid streams leave, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Substituting, the power output of the turbine is

$$\dot{W}_{out} = (20 \text{ kg/s})(3476.5 \text{ kJ/kg}) - (1 \text{ kg/s})(2828.3 \text{ kJ/kg}) - (19 \text{ kg/s})(2675.8 \text{ kJ/kg})$$
$$= \mathbf{15{,}860 \text{ kW}}$$

Throttling Valves

5-64C Yes.

5-65C No. Because air is an ideal gas and $h = h(T)$ for ideal gases. Thus if h remains constant, so does the temperature.

5-66C If it remains in the liquid phase, no. But if some of the liquid vaporizes during throttling, then yes.

5-67C The temperature of a fluid can increase, decrease, or remain the same during a throttling process. Therefore, this claim is valid since no thermodynamic laws are violated.

5-68 Refrigerant-134a is throttled by a capillary tube. The quality of the refrigerant at the exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

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

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

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

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0$.

The inlet enthalpy of R-134a is, from the refrigerant tables (Table A-11),

$$\left.\begin{array}{l} T_1 = 50°\text{C} \\ \text{sat. liquid} \end{array}\right\} h_1 = h_f = 123.49 \text{ kJ/kg}$$

The exit quality is

$$\left.\begin{array}{l} T_2 = -12°\text{C} \\ h_2 = h_1 \end{array}\right\} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{123.49 - 35.92}{207.38} = \mathbf{0.422}$$

5-69 Steam is throttled from a specified pressure to a specified state. The quality at the inlet is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

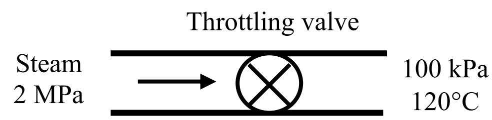
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0$.

The enthalpy of steam at the exit is (Table A-6),

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ T_2 = 120°\text{C} \end{array}\right\} h_2 = 2716.1 \text{ kJ/kg}$$

The quality of the steam at the inlet is (Table A-5)

$$\left.\begin{array}{l} P_2 = 2000 \text{ kPa} \\ h_1 = h_2 = 2716.1 \text{ kJ/kg} \end{array}\right\} x_1 = \frac{h_2 - h_f}{h_{fg}} = \frac{2716.1 - 908.47}{1889.8} = \mathbf{0.957}$$

5-70 [*Also solved by EES on enclosed CD***]** Refrigerant-134a is throttled by a valve. The pressure and internal energy after expansion are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of R-134a is, from the refrigerant tables (Tables A-11 through 13),

$$\left.\begin{array}{l} P_1 = 0.8 \text{ MPa} \\ T_1 = 25°\text{C} \end{array}\right\} h_1 \cong h_{f@25°\text{C}} = 86.41 \text{ kJ/kg}$$

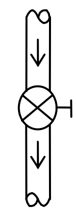
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left.\begin{array}{l} T_2 = -20°\text{C} \\ (h_2 = h_1) \end{array}\right\} \begin{array}{ll} h_f = 25.49 \text{ kJ/kg}, & u_f = 25.39 \text{ kJ/kg} \\ h_g = 238.41 \text{ kJ/kg} & u_g = 218.84 \text{ kJ/kg} \end{array}$$

Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state, and thus

$$P_2 = P_{sat\,@\,-20°\text{C}} = \mathbf{132.82\ kPa}$$

Also,

$$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{86.41 - 25.49}{212.91} = 0.2861$$

Thus,

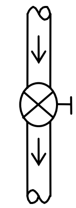
$$u_2 = u_f + x_2 u_{fg} = 25.39 + 0.2861 \times 193.45 = \mathbf{80.74\ kJ/kg}$$

5-71 Steam is throttled by a well-insulated valve. The temperature drop of the steam after the expansion is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of steam is (Tables A-6),

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500^\circ\text{C} \end{array}\right\} h_1 = 3399.5 \text{ kJ/kg}$$



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

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

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

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then the exit temperature of steam becomes

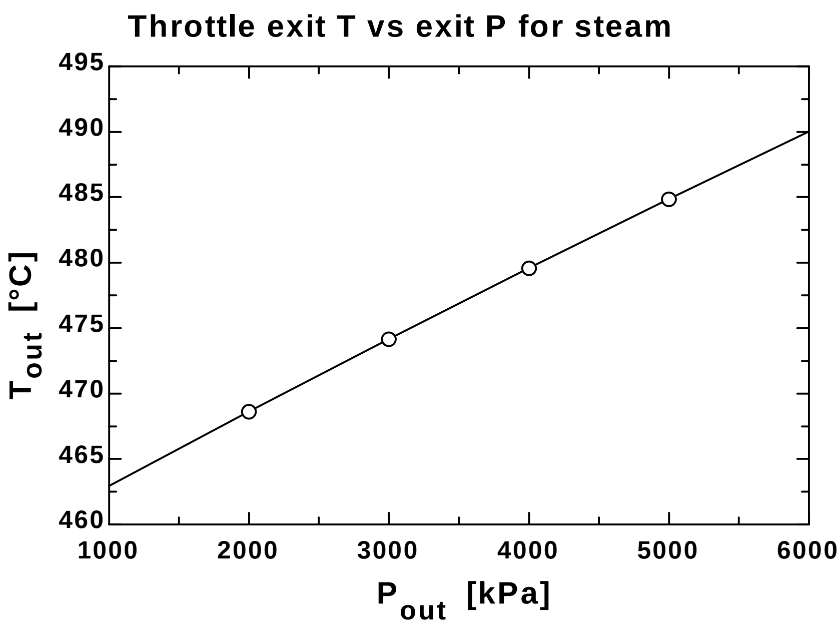
$$\left.\begin{array}{l} P_2 = 6 \text{ MPa} \\ (h_2 = h_1) \end{array}\right\} T_2 = \mathbf{490.1^\circ C}$$

5-72 EES Problem 5-71 is reconsidered. The effect of the exit pressure of steam on the exit temperature after throttling as the exit pressure varies from 6 MPa to 1 MPa is to be investigated. The exit temperature of steam is to be plotted against the exit pressure.

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

```
"Input information from Diagram Window"
{WorkingFluid$='Steam_iapws' "WorkingFluid: can be  changed to ammonia or other fluids"
P_in=8000 [kPa]
T_in=500  [C]
P_out=6000  [kPa]}
$Warning off

"Analysis"
m_dot_in=m_dot_out  "steady-state mass balance"
m_dot_in=1  "mass flow rate is arbitrary"
m_dot_in*h_in+Q_dot-W_dot-m_dot_out*h_out=0  "steady-state energy balance"
Q_dot=0  "assume the throttle to operate adiabatically"
W_dot=0  "throttles do not have any means of producing power"
h_in=enthalpy(WorkingFluid$,T=T_in,P=P_in)  "property table lookup"
T_out=temperature(WorkingFluid$,P=P_out,h=h_out)  "property table lookup"
x_out=quality(WorkingFluid$,P=P_out,h=h_out)  "x_out is the quality at the outlet"

P[1]=P_in;  P[2]=P_out;  h[1]=h_in;  h[2]=h_out  "use arrays to place points on property plot"
```

P_{out} [kPa]	T_{out} [C]
1000	463.1
2000	468.8
3000	474.3
4000	479.7
5000	484.9
6000	490.1



5-73E High-pressure air is throttled to atmospheric pressure. The temperature of air after the expansion is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved. **5** Air is an ideal gas.

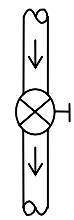
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$.

For an ideal gas,

$$h = h(T).$$

Therefore,

$$T_2 = T_1 = \mathbf{90°F}$$

5-74 Carbon dioxide flows through a throttling valve. The temperature change of CO_2 is to be determined if CO_2 is assumed an ideal gas and a real gas.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

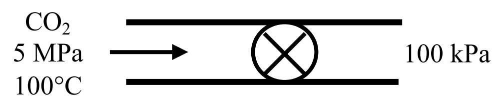
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$.

(*a*) For an ideal gas, $h = h(\text{T})$, and therefore,

$$T_2 = T_1 = 100°C \longrightarrow \Delta T = T_1 - T_2 = \mathbf{0°C}$$

(*b*) We obtain real gas properties of CO_2 from EES software as follows

$$\left.\begin{array}{l} P_1 = 5\text{ MPa} \\ T_1 = 100°\text{C} \end{array}\right\} h_1 = 34.77\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 100\text{ kPa} \\ h_2 = h_1 = 34.77\text{ kJ/kg} \end{array}\right\} T_2 = 66.0°\text{C}$$

Note that EES uses a different reference state from the textbook for CO_2 properties. The temperature difference in this case becomes

$$\Delta T = T_1 - T_2 = 100 - 66.0 = \mathbf{34.0°C}$$

That is, the temperature of CO_2 decreases by 34°C in a throttling process if its real gas properties are used.

Mixing Chambers and Heat Exchangers

5-75C Yes, if the mixing chamber is losing heat to the surrounding medium.

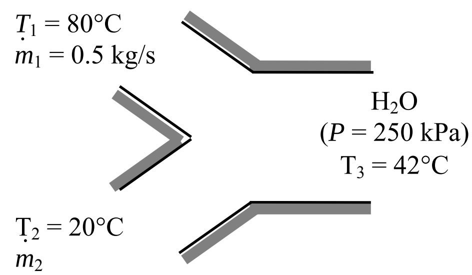
5-76C Under the conditions of no heat and work interactions between the mixing chamber and the surrounding medium.

5-77C Under the conditions of no heat and work interactions between the heat exchanger and the surrounding medium.

5-78 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There are no work interactions.

Properties Noting that $T < T_{sat\ @\ 250\ kPa} = 127.41°C$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,



$$h_1 \cong h_{f\,@\,80°C} = 335.02 \text{ kJ/kg}$$

$$h_2 \cong h_{f\,@\,20°C} = 83.915 \text{ kJ/kg}$$

$$h_3 \cong h_{f\,@\,42°C} = 175.90 \text{ kJ/kg}$$

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

Mass balance:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0\ (steady)} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

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

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

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

Combining the two relations and solving for $\dot{m}_2$ gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2}\dot{m}_1$$

Substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{(335.02 - 175.90)\text{ kJ/kg}}{(175.90 - 83.915)\text{ kJ/kg}}(0.5\text{ kg/s}) = \mathbf{0.865\ \ kg/s}$$

5-79E Liquid water is heated in a chamber by mixing it with saturated water vapor. If both streams enter at the same rate, the temperature and quality (if saturated) of the exit stream is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From steam tables (Tables A-5E through A-6E),

$$h_1 \cong h_{f\,@\,50°F} = 18.07 \text{ Btu/lbm}$$

$$h_2 = h_{g\,@\,50\text{ psia}} = 1174.2 \text{ Btu/lbm}$$

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

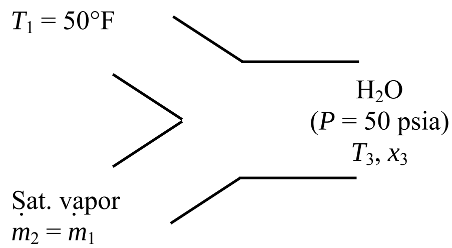
Mass balance:

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

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

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 2\dot{m}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$



Energy balance:

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

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

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

Combining the two gives

$$\dot{m}h_1 + \dot{m}h_2 = 2\dot{m}h_3 \text{ or } h_3 = (h_1 + h_2)/2$$

Substituting,

$$h_3 = (18.07 + 1174.2)/2 = 596.16 \text{ Btu/lbm}$$

At 50 psia, $h_f = 250.21$ Btu/lbm and $h_g = 1174.2$ Btu/lbm. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{sat\,@\,50\text{ psia}} = \mathbf{280.99°F}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{596.16 - 250.21}{924.03} = \mathbf{0.374}$$

5-80 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),

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

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

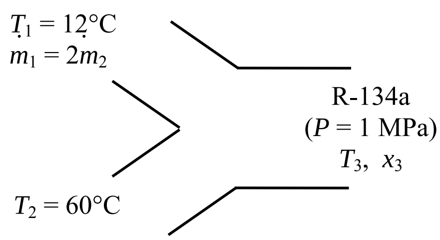
Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

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

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

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



Energy balance:

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

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

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

Combining the two gives $2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3$ *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,

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

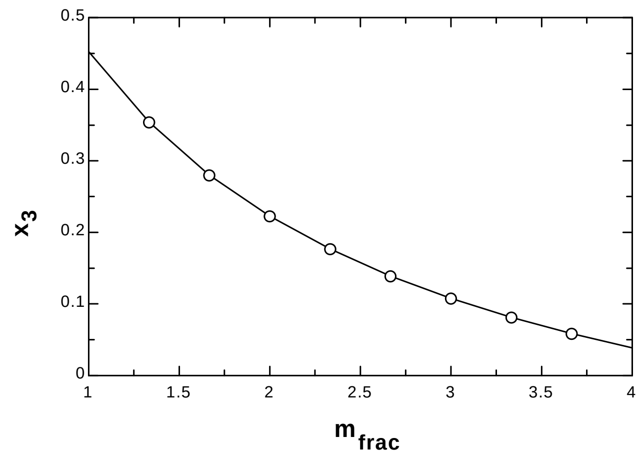
and

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

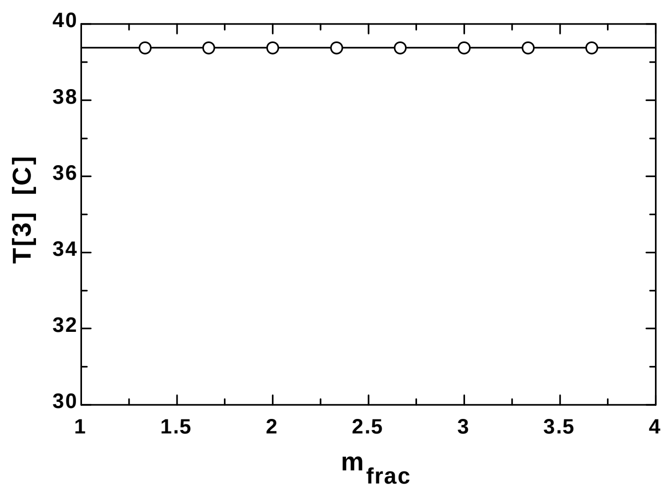
5-81 EES Problem 5-80 is reconsidered. The effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream as the ratio of the mass flow rate of the cold stream to that of the hot stream varies from 1 to 4 is to be investigated. The mixture temperature and quality are to be plotted against the cold-to-hot mass flow rate ratio.

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

```
"Input Data"
"m_frac = 2"  "m_frac =m_dot_cold/m_dot_hot= m_dot_1/m_dot_2"
T[1]=12 [C]
P[1]=1000 [kPa]
T[2]=60 [C]
P[2]=1000 [kPa]
m_dot_1=m_frac*m_dot_2
P[3]=1000 [kPa]
m_dot_1=1

"Conservation of mass for the R134a: Sum of m_dot_in=m_dot_out"
m_dot_1+ m_dot_2 =m_dot_3

"Conservation of Energy for steady-flow: neglect changes in KE and PE"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0   "Steady-flow requirement"
E_dot_in=m_dot_1*h[1] + m_dot_2*h[2]
E_dot_out=m_dot_3*h[3]

"Property data are given by:"
h[1] =enthalpy(R134a,T=T[1],P=P[1])
h[2] =enthalpy(R134a,T=T[2],P=P[2])
T[3] =temperature(R134a,P=P[3],h=h[3])
x_3=QUALITY(R134a,h=h[3],P=P[3])
```



m_{frac}	T_3 [C]	x_3
1	39.37	0.4491
1.333	39.37	0.3509
1.667	39.37	0.2772
2	39.37	0.2199
2.333	39.37	0.174
2.667	39.37	0.1365
3	39.37	0.1053
3.333	39.37	0.07881
3.667	39.37	0.05613
4	39.37	0.03649



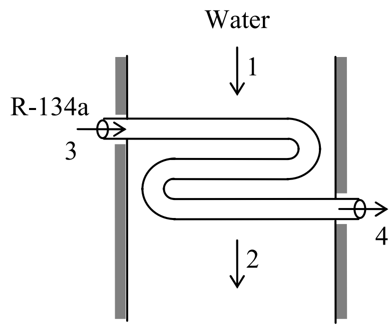
5-82 Refrigerant-134a is condensed in a water-cooled condenser. The mass flow rate of the cooling water required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

Properties The enthalpies of R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_3 = 700\text{ kPa} \\ T_3 = 70°\text{C} \end{array}\right\} h_3 = 308.33\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 700\text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_4 = h_{f@700\text{ kPa}} = 88.82\text{ kJ/kg}$$



Water exists as compressed liquid at both states, and thus (Table A-4)

$$h_1 \cong h_{f@15°\text{C}} = 62.98\text{ kJ/kg}$$

$$h_2 \cong h_{f@25°\text{C}} = 104.83\text{ kJ/kg}$$

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

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_w \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

Energy balance (for the heat exchanger):

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

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_w(h_2 - h_1) = \dot{m}_R(h_3 - h_4)$

Solving for $\dot{m}_w$: $\dot{m}_w = \dfrac{h_3 - h_4}{h_2 - h_1}\dot{m}_R$

Substituting,

$$\dot{m}_w = \frac{(308.33 - 88.82)\text{kJ/kg}}{(104.83 - 62.98)\text{kJ/kg}}(8\text{ kg/min}) = \mathbf{42.0\ kg/min}$$

5-83E [*Also solved by EES on enclosed CD***]** Air is heated in a steam heating system. For specified flow rates, the volume flow rate of air at the inlet is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

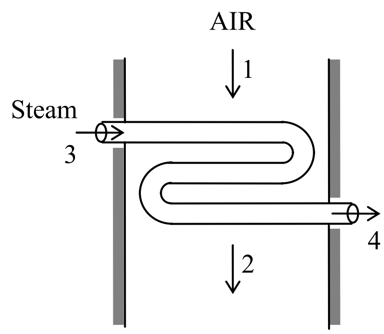
Properties The gas constant of air is 0.3704 psia.ft^3/lbm.R (Table A-1E). The constant pressure specific heat of air is c_p = 0.240 Btu/lbm·°F (Table A-2E). The enthalpies of steam at the inlet and the exit states are (Tables A-4E through A-6E)

$$\left.\begin{array}{l} P_3 = 30 \text{ psia} \\ T_3 = 400°\text{F} \end{array}\right\} h_3 = 1237.9 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 25 \text{ psia} \\ T_4 = 212°\text{F} \end{array}\right\} h_4 \cong h_{f@212°\text{F}} = 180.21 \text{ Btu/lbm}$$

Analysis We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (*for each fluid stream*):



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

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

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_a \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

Energy balance (for the entire heat exchanger):

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

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_a (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$

Solving for $\dot{m}_a$:

$$\dot{m}_a = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \frac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$$

Substituting,

$$\dot{m}_a = \frac{(1237.9 - 180.21)\text{Btu/lbm}}{(0.240 \text{ Btu/lbm} \cdot °\text{F})(130 - 80)°\text{F}} (15 \text{ lbm/min}) = 1322 \text{ lbm/min} = 22.04 \text{ lbm/s}$$

Also,

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(540 \text{ R})}{14.7 \text{ psia}} = 13.61 \text{ ft}^3/\text{lbm}$$

Then the volume flow rate of air at the inlet becomes

$$\dot{\boldsymbol{V}}_1 = \dot{m}_a \boldsymbol{v}_1 = (22.04 \text{ lbm/s})(13.61 \text{ ft}^3/\text{lbm}) = \mathbf{300 \text{ ft}^3/s}$$

5-84 Steam is condensed by cooling water in the condenser of a power plant. If the temperature rise of the cooling water is not to exceed 10°C, the minimum mass flow rate of the cooling water required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Liquid water is an incompressible substance with constant specific heats at room temperature.

Properties The cooling water exists as compressed liquid at both states, and its specific heat at room temperature is c = 4.18 kJ/kg·°C (Table A-3). The enthalpies of the steam at the inlet and the exit states are (Tables A-5 and A-6)

$$\left.\begin{array}{l} P_3 = 20 \text{ kPa} \\ x_3 = 0.95 \end{array}\right\} h_3 = h_f + x_3 h_{fg} = 251.42 + 0.95 \times 2357.5 = 2491.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_4 \cong h_{f@20 \text{ kPa}} = 251.42 \text{ kJ/kg}$$

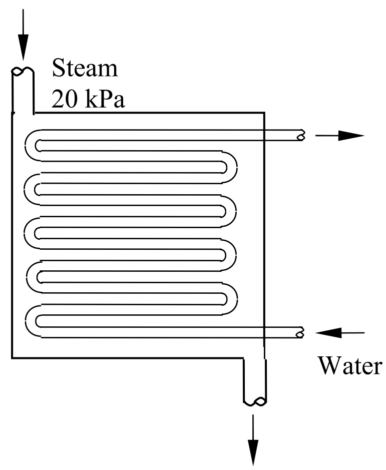
Analysis We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (for each fluid stream):

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

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

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_w \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$



Energy balance (for the heat exchanger):

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

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two,

$$\dot{m}_w (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$$

Solving for $\dot{m}_w$:

$$\dot{m}_w = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \frac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$$

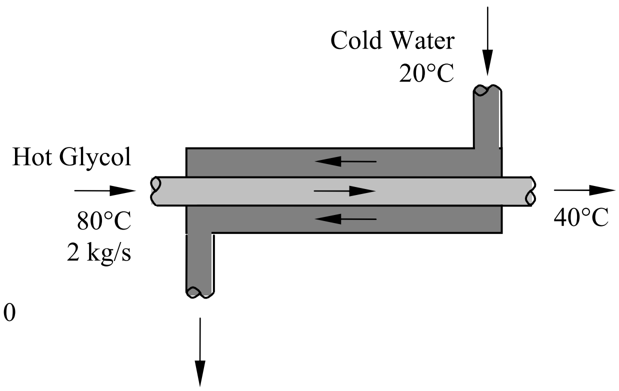
Substituting,

$$\dot{m}_w = \frac{(2491.1 - 251.42) \text{kJ/kg}}{(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(10^\circ\text{C})} (20{,}000/3600 \text{ kg/s}) = \mathbf{297.7 \text{ kg/s}}$$

5-85 Ethylene glycol is cooled by water in a heat exchanger. The rate of heat transfer in the heat exchanger and the mass flow rate of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg.°C, respectively.



Analysis (*a*) We take the ethylene glycol tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Then the rate of heat transfer becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{glycol} = (2\text{ kg/s})(2.56\text{ kJ/kg.°C})(80°\text{C} - 40°\text{C}) = \mathbf{204.8\ kW}$$

(*b*) The rate of heat transfer from glycol must be equal to the rate of heat transfer to the water. Then,

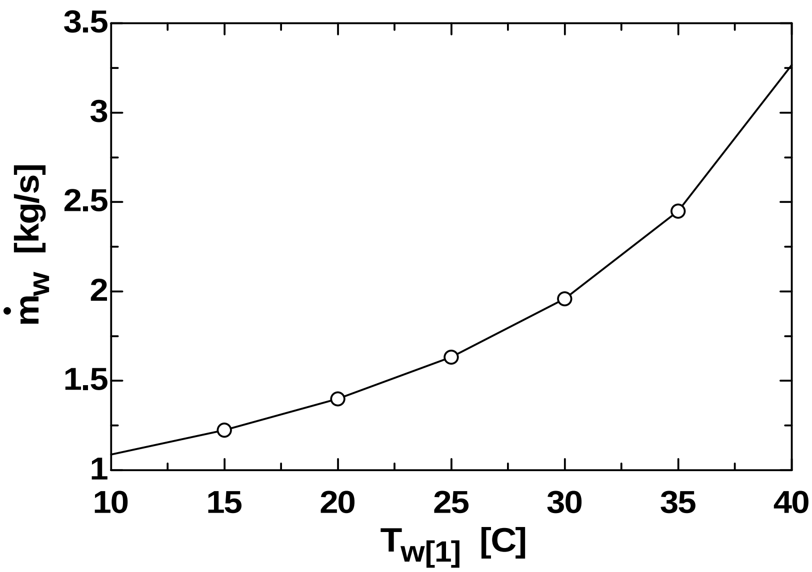
$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} \longrightarrow \dot{m}_{water} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{204.8\text{ kJ/s}}{(4.18\text{ kJ/kg.°C})(55°\text{C} - 20°\text{C})} = \mathbf{1.4\ kg/s}$$

5-86 EES Problem 5-85 is reconsidered. The effect of the inlet temperature of cooling water on the mass flow rate of water as the inlet temperature varies from 10°C to 40°C at constant exit temperature) is to be investigated. The mass flow rate of water is to be plotted against the inlet temperature.

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

```
"Input Data"

{T_w[1]=20 [C]}
T_w[2]=55 [C]  "w: water"
m_dot_eg=2 [kg/s] "eg: ethylene glycol"
T_eg[1]=80 [C]
T_eg[2]=40 [C]
C_p_w=4.18 [kJ/kg-K]
C_p_eg=2.56 [kJ/kg-K]

"Conservation of mass for the water: m_dot_w_in=m_dot_w_out=m_dot_w"
"Conservation of mass for the ethylene glycol: m_dot_eg_in=m_dot_eg_out=m_dot_eg"

"Conservation of Energy for steady-flow: neglect changes in KE and PE in each mass steam"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0 "Steady-flow requirement"
E_dot_in=m_dot_w*h_w[1] + m_dot_eg*h_eg[1]
E_dot_out=m_dot_w*h_w[2] + m_dot_eg*h_eg[2]
Q_exchanged =m_dot_eg*h_eg[1] - m_dot_eg*h_eg[2]

"Property data are given by:"
h_w[1] =C_p_w*T_w[1]  "liquid approximation applied for water and ethylene glycol"
h_w[2] =C_p_w*T_w[2]
h_eg[1] =C_p_eg*T_eg[1]
h_eg[2] =C_p_eg*T_eg[2]
```

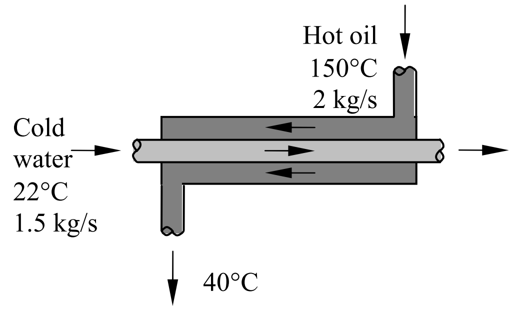
m_w [kg/s]	$T_{w,1}$ [C]
1.089	10
1.225	15
1.4	20
1.633	25
1.96	30
2.45	35
3.266	40



5-87 Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer in the heat exchanger and the exit temperature of water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg.°C, respectively.



Analysis We take the oil tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Then the rate of heat transfer from the oil becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} = (2 \text{ kg/s})(2.2 \text{ kJ/kg.°C})(150°\text{C} - 40°\text{C}) = \mathbf{484\ kW}$$

Noting that the heat lost by the oil is gained by the water, the outlet temperature of the water is determined from

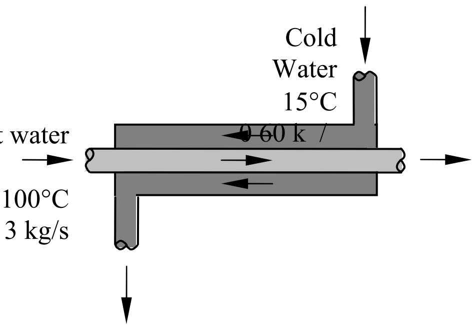
$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}_{water}c_p} = 22°\text{C} + \frac{484 \text{ kJ/s}}{(1.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})} = \mathbf{99.2°C}$$

5-88 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the exit temperature of hot water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

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

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

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{cold water}} = (0.60 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) = \mathbf{75.24 \text{ kW}}$$

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

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 100°\text{C} - \frac{75.24 \text{ kW}}{(3 \text{ kg/s})(4.19 \text{ kJ/kg.°C})} = \mathbf{94.0°C}$$

5-89 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the outlet temperature of the air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively.

Analysis We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

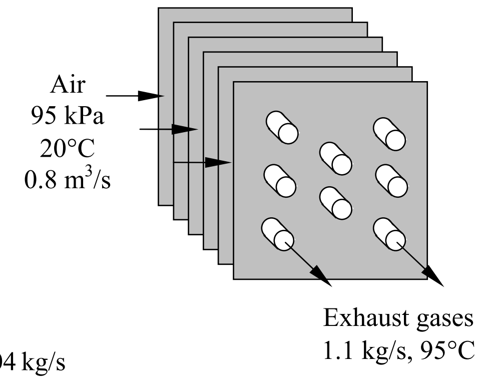
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

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

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

Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{gas}$$
$$= (1.1 \text{ kg/s})(1.1 \text{ kJ/kg.°C})(180°\text{C} - 95°\text{C})$$
$$= \mathbf{102.85\ kW}$$



The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95 \text{ kPa})(0.8 \text{ m}^3/\text{s})}{(0.287 \text{ kPa.m}^3/\text{kg.K}) \times 293 \text{ K}} = 0.904 \text{ kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

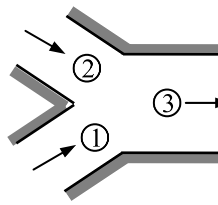
$$\dot{Q} = \dot{m}c_p(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{102.85 \text{ kW}}{(0.904 \text{ kg/s})(1.005 \text{ kJ/kg.°C})} = \mathbf{133.2°C}$$

5-90 An adiabatic open feedwater heater mixes steam with feedwater. The outlet mass flow rate and the outlet velocity are to be determined.

Assumptions Steady operating conditions exist.

Properties From a mass balance

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.2 + 10 = \mathbf{10.2\ kg/s}$$



The specific volume at the exit is (Table A-4)

$$\left.\begin{array}{l} P_3 = 100\ \text{kPa} \\ T_3 = 60°\text{C} \end{array}\right\} \boldsymbol{v}_3 \cong \boldsymbol{v}_{f\,@\,60°\text{C}} = 0.001017\ \text{m}^3/\text{kg}$$

The exit velocity is then

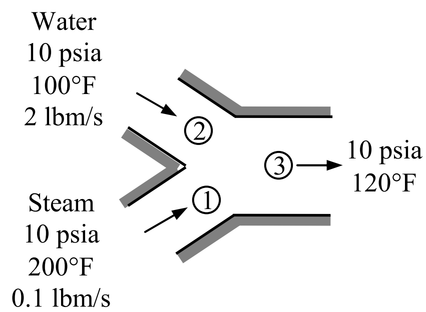
$$V_3 = \frac{\dot{m}_3 \boldsymbol{v}_3}{A_3} = \frac{4\dot{m}_3 \boldsymbol{v}_3}{\pi D^2}$$
$$= \frac{4(10.2\ \text{kg/s})(0.001017\ \text{m}^3/\text{kg})}{\pi (0.03\ \text{m})^2}$$
$$= \mathbf{14.68\ m/s}$$

5-91E An adiabatic open feedwater heater mixes steam with feedwater. The outlet mass flow rate and the outlet velocity are to be determined for two exit temperatures.

Assumptions Steady operating conditions exist.

Properties From a mass balance

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.1 + 2 = \mathbf{2.1\ lbm/s}$$



The specific volume at the exit is (Table A-4E)

$$\left.\begin{array}{l} P_3 = 10\text{ psia} \\ T_3 = 120°\text{F} \end{array}\right\} \boldsymbol{v}_3 \cong \boldsymbol{v}_{f\,@\,120°\text{F}} = 0.01620\text{ ft}^3/\text{lbm}$$

The exit velocity is then

$$V_3 = \frac{\dot{m}_3 \boldsymbol{v}_3}{A_3} = \frac{4\dot{m}_3 \boldsymbol{v}_3}{\pi D^2}$$
$$= \frac{4(2.1\text{ lbm/s})(0.01620\text{ ft}^3/\text{lbm})}{\pi (0.5\text{ ft})^2} = \mathbf{0.1733\ ft/s}$$

When the temperature at the exit is 180°F, we have

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.1 + 2 = \mathbf{2.1\ lbm/s}$$

$$\left.\begin{array}{l} P_3 = 10\text{ psia} \\ T_3 = 180°\text{F} \end{array}\right\} \boldsymbol{v}_3 \cong \boldsymbol{v}_{f\,@\,180°\text{F}} = 0.01651\text{ ft}^3/\text{lbm}$$

$$V_3 = \frac{\dot{m}_3 \boldsymbol{v}_3}{A_3} = \frac{4\dot{m}_3 \boldsymbol{v}_3}{\pi D^2} = \frac{4(2.1\text{ lbm/s})(0.01651\text{ ft}^3/\text{lbm})}{\pi (0.5\text{ ft})^2} = \mathbf{0.1766\ ft/s}$$

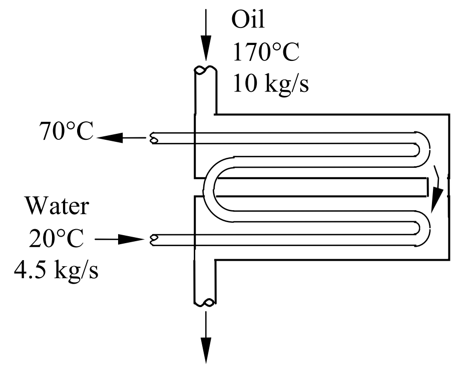
The mass flow rate at the exit is same while the exit velocity slightly increases when the exit temperature is 180°F instead of 120°F.

5-92 Water is heated by hot oil in a heat exchanger. The rate of heat transfer in the heat exchanger and the outlet temperature of oil are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

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

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

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = \mathbf{940.5 \text{ kW}}$$

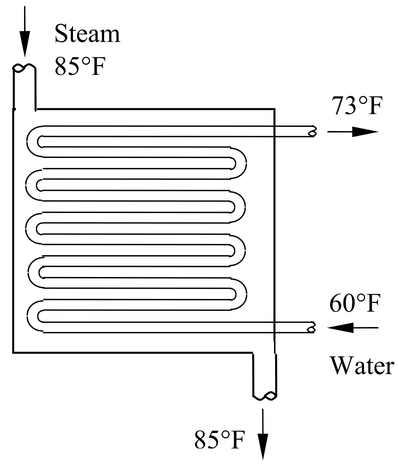
Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

5-93E Steam is condensed by cooling water in a condenser. The rate of heat transfer in the heat exchanger and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy of vaporization of water at 85°F is 1045.2 Btu/lbm (Table A-4E).



Analysis We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

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

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} = (138 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73°\text{F} - 60°\text{F}) = \textbf{1794 Btu/s}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

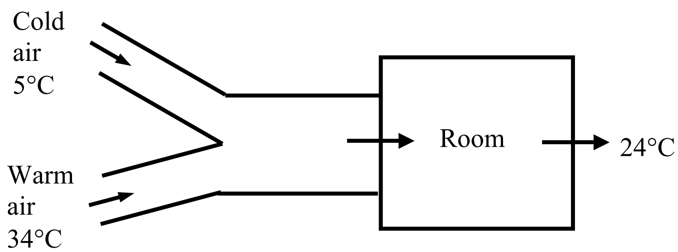
$$\dot{Q} = (\dot{m}h_{fg})_{steam} = \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{1794 \text{ Btu/s}}{1045.2 \text{ Btu/lbm}} = \textbf{1.72 lbm/s}$$

5-94 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-17) 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 $\quad \dot{m}_1 h_1 + 1.6\dot{m}_1 h_2 = 2.6\dot{m}_1 h_3 \;\; or \;\; h_3 = (h_1 + 1.6h_2)/2.6$

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

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

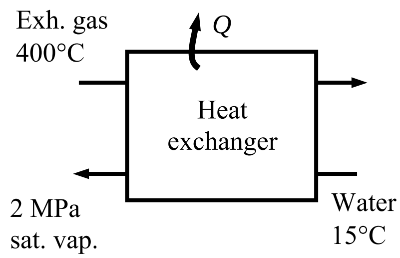
5-95 A heat exchanger that is not insulated is used to produce steam from the heat given up by the exhaust gases of an internal combustion engine. The temperature of exhaust gases at the heat exchanger exit and the rate of heat transfer to the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Exhaust gases are assumed to have air properties with constant specific heats.

Properties The constant pressure specific heat of the exhaust gases is taken to be c_p = 1.045 kJ/kg·°C (Table A-2). The inlet and exit enthalpies of water are (Tables A-4 and A-5)

$$\left.\begin{array}{l} T_{\text{w,in}} = 15°\text{C} \\ x = 0\ (\text{sat. liq.}) \end{array}\right\} h_{\text{w,in}} = 62.98\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_{\text{w,out}} = 2\ \text{MPa} \\ x = 1\ (\text{sat. vap.}) \end{array}\right\} h_{\text{w,out}} = 2798.3\ \text{kJ/kg}$$



Analysis We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (*for each fluid stream*):

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

Energy balance (for the entire heat exchanger):

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

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

$$\dot{m}_{\text{exh}} h_{\text{exh,in}} + \dot{m}_{\text{w}} h_{\text{w,in}} = \dot{m}_{\text{exh}} h_{\text{exh,out}} + \dot{m}_{\text{w}} h_{\text{w,out}} + \dot{Q}_{\text{out}} \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

or

$$\dot{m}_{\text{exh}} c_p T_{\text{exh,in}} + \dot{m}_w h_{\text{w,in}} = \dot{m}_{\text{exh}} c_p T_{\text{exh,out}} + \dot{m}_w h_{\text{w,out}} + \dot{Q}_{\text{out}}$$

Noting that the mass flow rate of exhaust gases is 15 times that of the water, substituting gives

$$\begin{aligned} &15\dot{m}_{\text{w}}(1.045\ \text{kJ/kg.°C})(400°\text{C}) + \dot{m}_{\text{w}}(62.98\ \text{kJ/kg}) \\ &\quad = 15\dot{m}_{\text{w}}(1.045\ \text{kJ/kg.°C})T_{\text{exh,out}} + \dot{m}_{\text{w}}(2798.3\ \text{kJ/kg}) + \dot{Q}_{out} \end{aligned} \qquad (1)$$

The heat given up by the exhaust gases and heat picked up by the water are

$$\dot{Q}_{exh} = \dot{m}_{\text{exh}} c_p (T_{\text{exh,in}} - T_{\text{exh,out}}) = 15\dot{m}_{\text{w}}(1.045\ \text{kJ/kg.°C})(400 - T_{\text{exh,out}})°\text{C} \qquad (2)$$

$$\dot{Q}_w = \dot{m}_{\text{w}}(h_{\text{w,out}} - h_{\text{w,in}}) = \dot{m}_{\text{w}}(2798.3 - 62.98)\text{kJ/kg} \qquad (3)$$

The heat loss is

$$\dot{Q}_{out} = f_{\text{heat loss}} \dot{Q}_{exh} = 0.1\dot{Q}_{exh} \qquad (4)$$

The solution may be obtained by a trial-error approach. Or, solving the above equations simultaneously using EES software, we obtain

$$T_{\text{exh,out}} = \mathbf{206.1°C},\ \dot{Q}_{\text{w}} = \mathbf{97.26\ kW},\ \dot{m}_{\text{w}} = 0.03556\ \text{kg/s},\ \dot{m}_{\text{exh}} = 0.5333\ \text{kg/s}$$

5-96 Two streams of water are mixed in an insulated chamber. The temperature of the exit stream is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-4),

$$h_1 \cong h_{f\,@\,90°C} = 377.04 \text{ kJ/kg}$$

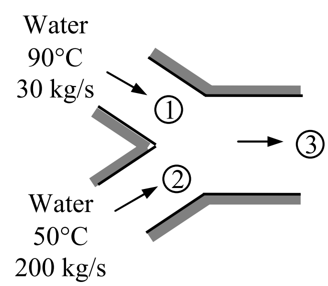
$$h_2 \cong h_{f\,@\,50°C} = 209.34 \text{ kJ/kg}$$

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

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

Mass balance: $$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$



Energy balance:

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

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

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

Solving for the exit enthalpy,

$$h_3 = \frac{\dot{m}_1 h_1 + \dot{m}_2 h_2}{\dot{m}_1 + \dot{m}_2} = \frac{(30)(377.04) + (200)(209.34)}{30 + 200} = 231.21 \text{ kJ/kg}$$

The temperature corresponding to this enthalpy is

T_3 =**55.2°C** (Table A-4)

5-97 A chilled-water heat-exchange unit is designed to cool air by water. The maximum water outlet temperature is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2*a*). The specific heat of water is 4.18 kJ/kg·K (Table A-3).

Analysis The water temperature at the heat exchanger exit will be maximum when all the heat released by the air is picked up by the water. First, the inlet specific volume and the mass flow rate of air are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303\ \text{K})}{100\ \text{kPa}} = 0.8696\ \text{m}^3/\text{kg}$$

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{5\ \text{m}^3/\text{s}}{0.8696\ \text{m}^3/\text{kg}} = 5.750\ \text{kg/s}$$

We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_3 = \dot{m}_a \text{ and } \dot{m}_2 = \dot{m}_4 = \dot{m}_w$$

Energy balance (for the entire heat exchanger):

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

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

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two,

$$\dot{m}_a(h_1 - h_3) = \dot{m}_w(h_4 - h_2)$$

$$\dot{m}_a c_{p,a}(T_1 - T_3) = \dot{m}_w c_{p,w}(T_4 - T_2)$$

Solving for the exit temperature of water,

$$T_4 = T_2 + \frac{\dot{m}_a c_{p,a}(T_1 - T_3)}{\dot{m}_w c_{p,w}} = 8°\text{C} + \frac{(5.750\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot°\text{C})(30-18)°\text{C}}{(2\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})} = \mathbf{16.3°C}$$

Pipe and duct Flow

5-98 Saturated liquid water is heated in a steam boiler at a specified rate. The rate of heat transfer in the boiler is to be determined.

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

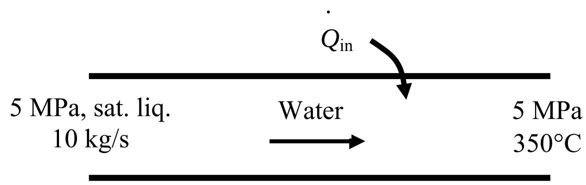
Analysis We take the pipe in which the water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



The enthalpies of water at the inlet and exit of the boiler are (Table A-5, A-6).

$$\left.\begin{array}{l} P_1 = 5\text{ MPa} \\ x = 0 \end{array}\right\} h_1 \cong h_{f\ @\ 5\text{ MPa}} = 1154.5\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 5\text{ MPa} \\ T_2 = 350°\text{C} \end{array}\right\} h_2 = 3069.3\text{ kJ/kg}$$

Substituting,

$$\dot{Q}_{in} = (10\text{ kg/s})(3069.3 - 1154.5)\text{kJ/kg} = \mathbf{19{,}150\ kW}$$

5-99 Air at a specified rate is heated by an electrical heater. The current is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The heat losses from the air is negligible.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2*a*).

Analysis We take the pipe in which the air is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

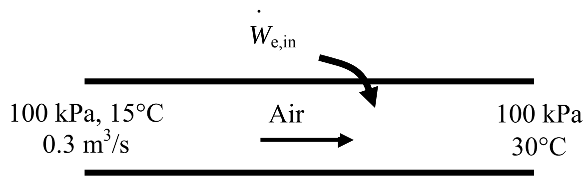
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}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

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

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

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

$$\mathbf{V}I = \dot{m}c_p(T_2 - T_1)$$



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

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{100\ \text{kPa}} = 0.8266\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.3\ \text{m}^3/\text{s}}{0.8266\ \text{m}^3/\text{kg}} = 0.3629\ \text{kg/s}$$

Substituting into the energy balance equation and solving for the current gives

$$I = \frac{\dot{m}c_p(T_2 - T_1)}{\mathbf{V}} = \frac{(0.3629\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})(30-15)\text{K}}{110\ \text{V}}\left(\frac{1000\ \text{VI}}{1\ \text{kJ/s}}\right) = \mathbf{49.7\ Amperes}$$

5-100E The cooling fan of a computer draws air, which is heated in the computer by absorbing the heat of PC circuits. The electrical power dissipated by the circuits is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** All the heat dissipated by the circuits are picked up by the air drawn by the fan.

Properties The gas constant of air is 0.3704 psia·ft^3/lbm·R (Table A-1E). The constant pressure specific heat of air at room temperature is c_p = 0.240 Btu/lbm·°F (Table A-2E*a*).

Analysis We take the pipe in which the air is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

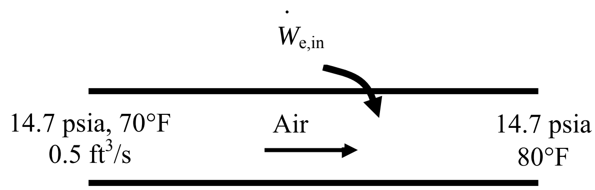
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

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

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

$$\dot{W}_{e,in} = \dot{m}c_p(T_2 - T_1)$$



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

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(530 \text{ R})}{14.7 \text{ psia}} = 13.35 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{\boldsymbol{v}_1} = \frac{0.5 \text{ ft}^3/\text{s}}{13.35 \text{ ft}^3/\text{lbm}} = 0.03745 \text{ lbm/s}$$

Substituting,

$$\dot{W}_{e,out} = (0.03745 \text{ lbm/s})(0.240 \text{ Btu/lbm} \cdot \text{R})(80-70)\text{Btu/lbm}\left(\frac{1 \text{ kW}}{0.94782 \text{ Btu}}\right) = \mathbf{0.0948\ kW}$$

5-101 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of T_{avg} = (45+60)/2 = 52.5°C = 325.5 K is c_p = 1.0065 kJ/kg.°C. The gas constant for air is R = 0.287 kJ/kg.K (Table A-2).

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Then the required mass flow rate of air to absorb heat at a rate of 60 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{60 \text{ W}}{(1006.5 \text{ J/kg.°C})(60 - 45)\text{°C}}$$
$$= 0.00397 \text{ kg/s} = 0.238 \text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.238 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.341 \text{ m}^3\text{/min}}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \rightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.341 \text{ m}^3\text{/min})}{\pi (110 \text{ m/min})}} = 0.063 \text{ m} = \mathbf{6.3 \text{ cm}}$$

5-102 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of T_{ave} = (45+60)/2 = 52.5°C is c_p = 1.0065 kJ/kg.°C The gas constant for air is R = 0.287 kJ/kg.K (Table A-2).

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Then the required mass flow rate of air to absorb heat at a rate of 100 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in})$$

$$\dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{100 \text{ W}}{(1006.5 \text{ J/kg.°C})(60 - 45)\text{°C}} = 0.006624 \text{ kg/s} = 0.397 \text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(60+273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.57\ m^3/min}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.57 \text{ m}^3/\text{min})}{\pi(110 \text{ m/min})}} = 0.081 \text{ m} = \mathbf{8.1\ cm}$$

5-103E Electronic devices mounted on a cold plate are cooled by water. The amount of heat generated by the electronic devices is to be determined.

Assumptions **1** Steady operating conditions exist. **2** About 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation. **3** Kinetic and potential energy changes are negligible.

Properties The properties of water at room temperature are ρ = 62.1 lbm/ft^3 and c_p = 1.00 Btu/lbm.°F (Table A-3E).

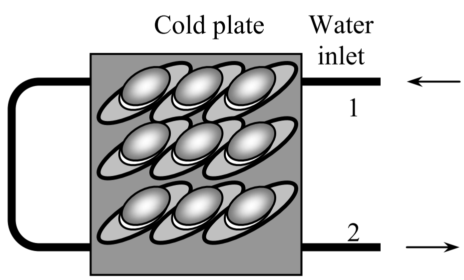
Analysis We take the tubes of the cold plate to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then mass flow rate of water and the rate of heat removal by the water are determined to be

$$\dot{m} = \rho AV = \rho \frac{\pi D^2}{4} V = (62.1\ \text{lbm/ft}^3)\frac{\pi(0.25/12\ \text{ft})^2}{4}(60\ \text{ft/min}) = 1.270\ \text{lbm/min} = 76.2\ \text{lbm/h}$$

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (76.2\ \text{lbm/h})(1.00\ \text{Btu/lbm.°F})(105 - 95)\text{°F} = 762\ \text{Btu/h}$$

which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

$$\dot{Q} = \frac{762\ \text{Btu/h}}{0.85} = \mathbf{896\ Btu/h} = \mathbf{263\ W}$$

5-104 [*Also solved by EES on enclosed CD***]** The components of an electronic device located in a horizontal duct of rectangular cross section are cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

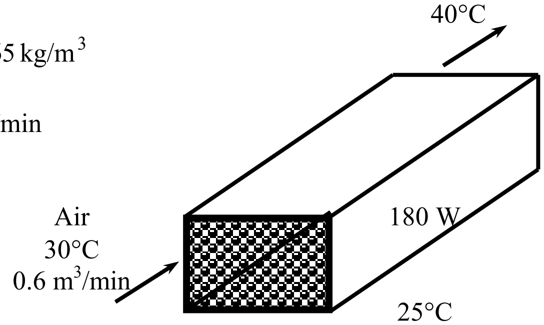
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible

Properties The gas constant of air is R = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.165\text{ kg/m}^3)(0.6\text{ m}^3/\text{min}) = 0.700\text{ kg/min}$$



We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{\text{air}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} = (0.700/60\text{ kg/s})(1.005\text{ kJ/kg.°C})(40-30)\text{°C} = 0.117\text{ kW} = 117\text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63\text{ W}}$$

5-105 The components of an electronic device located in a horizontal duct of circular cross section is cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible

Properties The gas constant of air is R = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.165\text{ kg/m}^3)(0.6\text{ m}^3/\text{min}) = 0.700\text{ kg/min}$$

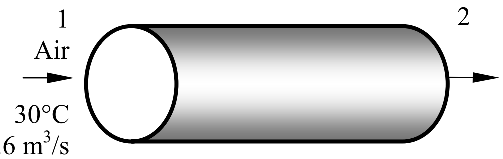
We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{air} = [\dot{m}c_p(T_{out} - T_{in})]_{air} = (0.700/60\text{ kg/s})(1.005\text{ kJ/kg.°C})(40-30)\text{°C} = 0.117\text{ kW} = 117\text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{external} = \dot{Q}_{total} - \dot{Q}_{internal} = 180 - 117 = \mathbf{63\ W}$$

5-106E Water is heated in a parabolic solar collector. The required length of parabolic collector is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat loss from the tube is negligible so that the entire solar energy incident on the tube is transferred to the water. **3** Kinetic and potential energy changes are negligible

Properties The specific heat of water at room temperature is c_p = 1.00 Btu/lbm.°F (Table A-2E).

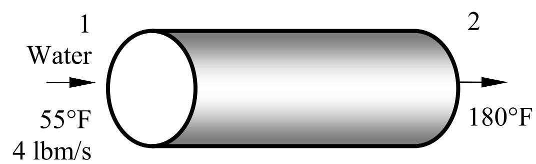
Analysis We take the thin aluminum tube to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the total rate of heat transfer to the water flowing through the tube becomes

$$\dot{Q}_{total} = \dot{m}c_p(T_e - T_i) = (4 \text{ lbm/s})(1.00 \text{ Btu/lbm.°F})(180 - 55)\text{°F} = 500 \text{ Btu/s} = 1{,}800{,}000 \text{ Btu/h}$$

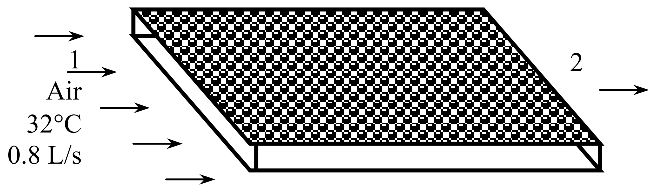
The length of the tube required is

$$L = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{1{,}800{,}000 \text{ Btu/h}}{400 \text{ Btu/h.ft}} = \mathbf{4500 \text{ ft}}$$

5-107 Air enters a hollow-core printed circuit board. The exit temperature of the air is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** The local atmospheric pressure is 1 atm. **4** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is R = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).



Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(32+273)\text{K}} = 1.16 \text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.16 \text{ kg/m}^3)(0.0008 \text{ m}^3/\text{s}) = 0.000928 \text{ kg/s}$$

We take the hollow core to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Then the exit temperature of air leaving the hollow core becomes

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}c_p} = 32\,°\text{C} + \frac{20 \text{ J/s}}{(0.000928 \text{ kg/s})(1005 \text{ J/kg.°C})} = \mathbf{53.4°C}$$

5-108 A computer is cooled by a fan blowing air through the case of the computer. The required flow rate of the air and the fraction of the temperature rise of air that is due to heat generated by the fan are to be determined.

Assumptions **1** Steady flow conditions exist. **2** Air is an ideal gas with constant specific heats. **3** The pressure of air is 1 atm. **4** Kinetic and potential energy changes are negligible

Properties The specific heat of air at room temperature is $c_p = 1.005$ kJ/kg.°C (Table A-2).

Analysis (*a*) We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Noting that the fan power is 25 W and the 8 PCBs transfer a total of 80 W of heat to air, the mass flow rate of air is determined to be

$$\dot{Q}_{in} + \dot{W}_{in} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q}_{in} + \dot{W}_{in}}{c_p(T_e - T_i)} = \frac{(8 \times 10)\text{ W} + 25\text{ W}}{(1005\text{ J/kg.°C})(10\text{°C})} = \mathbf{0.0104\ kg/s}$$

(*b*) The fraction of temperature rise of air that is due to the heat generated by the fan and its motor can be determined from

$$\dot{Q} = \dot{m}c_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{25\text{ W}}{(0.0104\text{ kg/s})(1005\text{ J/kg.°C})} = 2.4\text{°C}$$

$$f = \frac{2.4\text{°C}}{10\text{°C}} = 0.24 = \mathbf{24\%}$$

5-109 Hot water enters a pipe whose outer surface is exposed to cold air in a basement. The rate of heat loss from the water is to be determined.

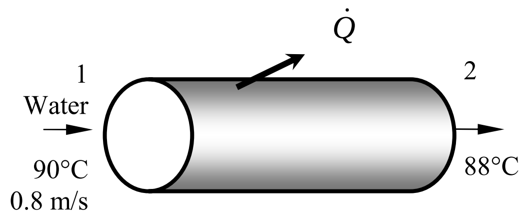
Assumptions **1** Steady flow conditions exist. **2** Water is an incompressible substance with constant specific heats. **3** The changes in kinetic and potential energies are negligible.

Properties The properties of water at the average temperature of (90+88)/2 = 89°C are $\rho = 965 \text{ kg/m}^3$ and c_p = 4.21 kJ/kg.°C (Table A-3).

Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c V = (965 \text{ kg/m}^3)\frac{\pi(0.04 \text{ m})^2}{4}(0.8 \text{ m/s}) = 0.970 \text{ kg/s}$$

We take the section of the pipe in the basement to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

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

Then the rate of heat transfer from the hot water to the surrounding air becomes

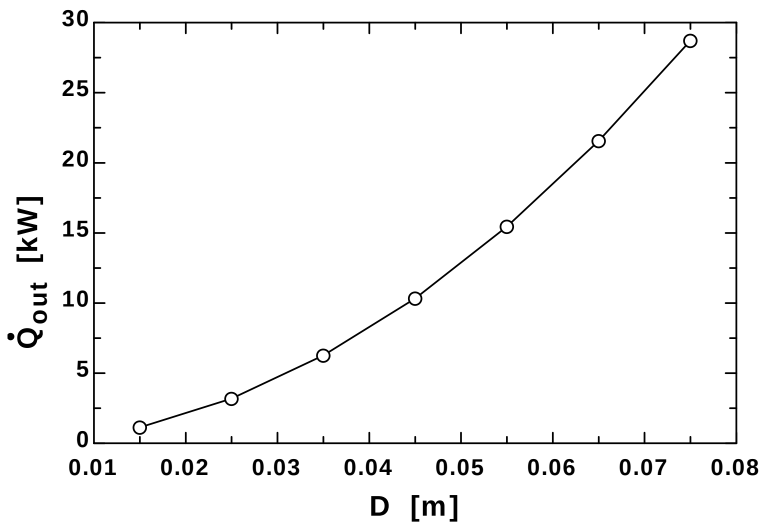
$$\dot{Q}_{out} = \dot{m}c_p[T_{in} - T_{out}]_{water} = (0.970 \text{ kg/s})(4.21 \text{ kJ/kg.°C})(90 - 88)\text{°C} = \mathbf{8.17 \text{ kW}}$$

5-110 EES Problem 5-109 is reconsidered. The effect of the inner pipe diameter on the rate of heat loss as the pipe diameter varies from 1.5 cm to 7.5 cm is to be investigated. The rate of heat loss is to be plotted against the diameter.

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

```
"Knowns:"

{D = 0.04 [m]}
rho = 965 [kg/m^3]
Vel = 0.8 [m/s]
T_1 = 90 [C]
T_2 = 88 [C]
C_P = 4.21[kJ/kg-C]

"Analysis:"

"The mass flow rate of water is:"
Area = pi*D^2/4
m_dot = rho*Area*Vel

"We take the section of the pipe in the basement to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as"

E_dot_in - E_dot_out = DELTAE_dot_sys

DELTAE_dot_sys = 0 "Steady-flow assumption"

E_dot_in = m_dot*h_in
E_dot_out =Q_dot_out+m_dot*h_out

h_in = C_P * T_1
h_out = C_P * T_2
```

D [m]	Q_{out} [kW]
0.015	1.149
0.025	3.191
0.035	6.254
0.045	10.34
0.055	15.44
0.065	21.57
0.075	28.72



5-111 A room is to be heated by an electric resistance heater placed in a duct in the room. The power rating of the electric heater and the temperature rise of air as it passes through the heater are to be determined.

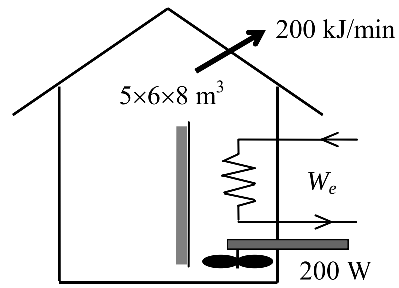
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the room.

Properties The gas constant of air is 0.287 kPa.m³/kg.K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 and c_v = 0.718 kJ/kg·K (Table A-2).

Analysis (*a*) The total mass of air in the room is

$$V = 5\times 6\times 8\ \text{m}^3 = 240\ \text{m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(98\ \text{kPa})(240\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})} = 284.6\ \text{kg}$$



We first take the *entire room* as our system, which is a closed system since no mass leaks in or out. The power rating of the electric heater is determined by applying the conservation of energy relation to this constant volume closed system:

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

$$W_{\text{e,in}} + W_{\text{fan,in}} - Q_{out} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}\right) = mc_{v,\text{avg}}(T_2 - T_1)$$

Solving for the electrical work input gives

$$\begin{aligned}\dot{W}_{\text{e,in}} &= \dot{Q}_{\text{out}} - \dot{W}_{\text{fan,in}} + mc_v(T_2 - T_1)/\Delta t \\ &= (200/60\ \text{kJ/s}) - (0.2\ \text{kJ/s}) + (284.6\ \text{kg})(0.718\ \text{kJ/kg}\cdot{}^\circ\text{C})(25-15)^\circ\text{C}/(15\times 60\ \text{s}) \\ &= \mathbf{5.40\ \ kW}\end{aligned}$$

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

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

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

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

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

$$\Delta T = T_2 - T_1 = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{\dot{m}c_p} = \frac{(5.40 + 0.2)\ \text{kJ/s}}{(50/60\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})} = \mathbf{6.7^\circ C}$$

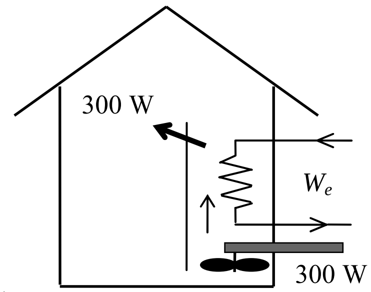
5-112 A house is heated by an electric resistance heater placed in a duct. The power rating of the electric heater is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·K (Table A-2)

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

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



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

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

$$\dot{W}_{e,in} + \dot{W}_{fan,in} = \dot{Q}_{out} + \dot{m}(h_2 - h_1) = \dot{Q}_{out} + \dot{m}c_p(T_2 - T_1)$$

Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{e,in} = \dot{Q}_{out} + \dot{m}c_p\Delta T - \dot{W}_{fan,in} = (0.3 \text{ kJ/s}) + (0.6 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot°\text{C})(7°\text{C}) - 0.3 \text{ kW} = \mathbf{4.22 \ kW}$$

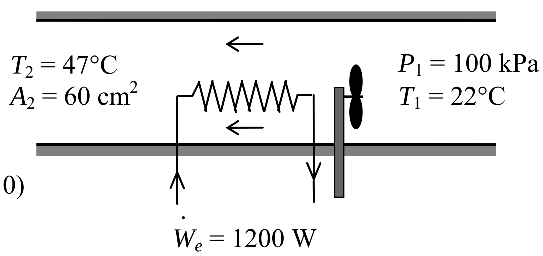
5-113 A hair dryer consumes 1200 W of electric power when running. The inlet volume flow rate and the exit velocity of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The power consumed by the fan and the heat losses are negligible.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·K (Table A-2)

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

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



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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{e,in}}{c_p(T_2 - T_1)} = \frac{1.2\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot^\circ\text{C})(47-22)^\circ\text{C}} = 0.04776\text{ kg/s}$$

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295\text{ K})}{(100\text{ kPa})} = 0.8467\text{ m}^3/\text{kg}$$

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (0.04776\text{ kg/s})(0.8467\text{ m}^3/\text{kg}) = \mathbf{0.0404\ m^3/s}$$

(*b*) The exit velocity of air is determined from the mass balance $\dot{m}_1 = \dot{m}_2 = \dot{m}$ to be

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(320\text{K})}{(100\text{ kPa})} = 0.9184\text{ m}^3/\text{kg}$$

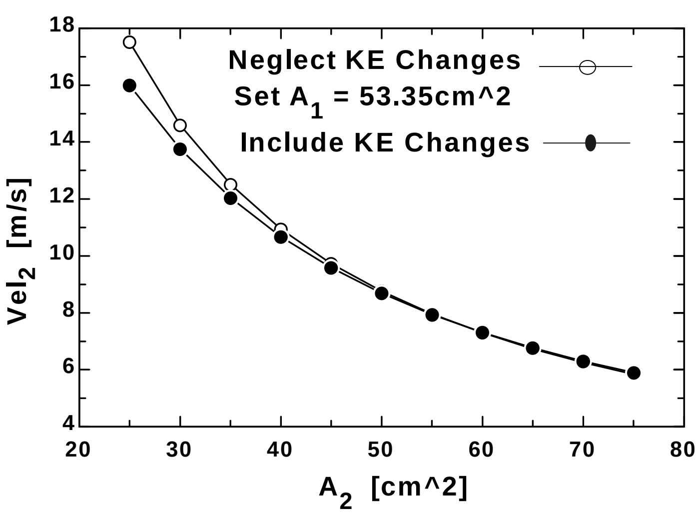
$$\dot{m} = \frac{1}{\boldsymbol{v}_2}A_2V_2 \longrightarrow V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{(0.04776\text{ kg/s})(0.9184\text{ m}^3/\text{kg})}{60\times10^{-4}\text{ m}^2} = \mathbf{7.31\ m/s}$$

5-114 EES Problem 5-113 is reconsidered. The effect of the exit cross-sectional area of the hair drier on the exit velocity as the exit area varies from 25 cm^2 to 75 cm^2 is to be investigated. The exit velocity is to be plotted against the exit cross-sectional area.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
R=0.287 [kPa-m^3/kg-K]
P= 100 [kPa]
T_1 = 22 [C]
T_2= 47 [C]
{A_2 = 60 [cm^2]}
A_1 = 53.35 [cm^2]
W_dot_ele=1200 [W]

"Analysis:

 We take the hair dryer as the system, which is a control volume since mass
crosses the boundary. There is only one inlet and one exit.  Thus, the
energy balance for this steady-flow system can be expressed in the rate form
as:"
E_dot_in = E_dot_out
E_dot_in = W_dot_ele*convert(W,kW) + m_dot_1*(h_1+Vel_1^2/2*convert(m^2/s^2,kJ/kg))
E_dot_out = m_dot_2*(h_2+Vel_2^2/2*convert(m^2/s^2,kJ/kg))
h_2  = enthalpy(air, T = T_2)
h_1= enthalpy(air, T = T_1)

"The volume flow rates of air are determined to be:"
V_dot_1 = m_dot_1*v_1
P*v_1=R*(T_1+273)
V_dot_2 = m_dot_2*v_2
P*v_2=R*(T_2+273)
m_dot_1 = m_dot_2
Vel_1=V_dot_1/(A_1*convert(cm^2,m^2))

"(b) The exit velocity of air is determined from the mass balance  to be"
Vel_2=V_dot_2/(A_2*convert(cm^2,m^2))
```

A_2 [cm^2]	Vel_2 [m/s]
25	16
30	13.75
35	12.03
40	10.68
45	9.583
50	8.688
55	7.941
60	7.31
65	6.77
70	6.303
75	5.896



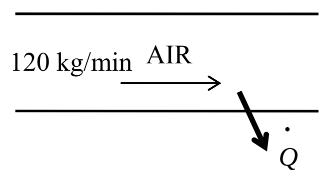
5-115 The ducts of a heating system pass through an unheated area. The rate of heat loss from the air in the ducts is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

Properties The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·K (Table A-2)

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

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



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

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

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$

Substituting,

$$\dot{Q}_{out} = (120 \text{ kg/min})(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(4^\circ\text{C}) = \mathbf{482\ kJ/min}$$

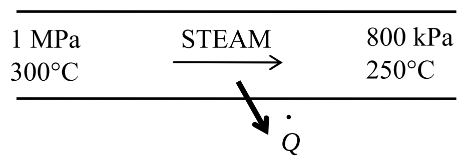
5-116 Steam pipes pass through an unheated area, and the temperature of steam drops as a result of heat losses. The mass flow rate of steam and the rate of heat loss from are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

Properties From the steam tables (Table A-6),

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.25799 \text{ m}^3/\text{kg} \\ h_1 = 3051.6 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 250°\text{C} \end{array}\right\} h_2 = 2950.4 \text{ kJ/kg}$$



Analysis (*a*) The mass flow rate of steam is determined directly from

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 = \frac{1}{0.25799 \text{ m}^3/\text{kg}} \left[\pi (0.06 \text{ m})^2\right] (2 \text{ m/s}) = \mathbf{0.0877\ kg/s}$$

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

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

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

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

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

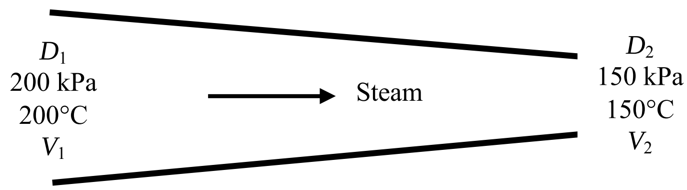
Substituting, the rate of heat loss is determined to be

$$\dot{Q}_{loss} = (0.0877 \text{ kg/s})(3051.6 - 2950.4) \text{ kJ/kg} = \mathbf{8.87\ kJ/s}$$

5-117 Steam flows through a non-constant cross-section pipe. The inlet and exit velocities of the steam are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

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



Mass balance:

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

$$\dot{m}_{in} = \dot{m}_{out} \longrightarrow A_1 \frac{V_1}{\boldsymbol{v}_1} = A_1 \frac{V_1}{\boldsymbol{v}_1} \longrightarrow \frac{\pi D_1^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi D_2^2}{4}\frac{V_2}{\boldsymbol{v}_2}$$

Energy balance:

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

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

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

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

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 1.0805 \text{ m}^3/\text{kg} \\ h_1 = 2870.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 1.2855 \text{ m}^3/\text{kg} \\ h_2 = 2772.9 \text{ kJ/kg} \end{array}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting,

$$\frac{\pi(1.8\text{ m})^2}{4}\frac{V_1}{(1.0805\text{ m}^3/\text{kg})} = \frac{\pi(1.0\text{ m})^2}{4}\frac{V_2}{(1.2855\text{ m}^3/\text{kg})} \qquad (1)$$

$$2870.7\text{ kJ/kg} + \frac{V_1^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 2772.9 \text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) \quad (2)$$

There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

$V_1 =$ **118.8 m/s**

$V_2 =$ **458.0 m/s**

5-118E Saturated liquid water is heated in a steam boiler. The heat transfer per unit mass is to be determined.

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

Analysis We take the pipe in which the water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

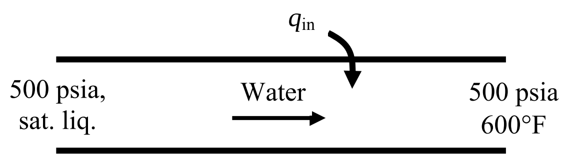
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

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

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

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

$$q_{in} = h_2 - h_1$$



The enthalpies of water at the inlet and exit of the boiler are (Table A-5E, A-6E).

$$\left.\begin{array}{l} P_1 = 500 \text{ psia} \\ x = 0 \end{array}\right\} h_1 \cong h_{f\,@\,500\text{ psia}} = 449.51 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ psia} \\ T_2 = 600°\text{F} \end{array}\right\} h_2 = 1298.6 \text{ Btu/lbm}$$

Substituting,

$$q_{in} = 1298.6 - 449.51 = \mathbf{849.1 \text{ Btu/lbm}}$$

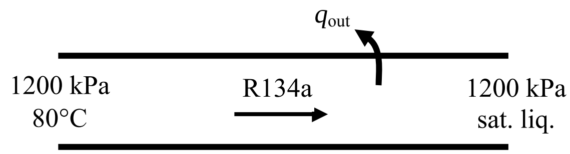
5-119 R-134a is condensed in a condenser. The heat transfer per unit mass is to be determined.

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

Analysis We take the pipe in which R-134a is condensed as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}h_1 = \dot{m}h_2 + \dot{Q}_{out}$$
$$\dot{Q}_{out} = \dot{m}(h_1 - h_2)$$
$$q_{out} = h_1 - h_2$$



The enthalpies of R-134a at the inlet and exit of the condenser are (Table A-12, A-13).

$$\left.\begin{array}{l} P_1 = 1200 \text{ kPa} \\ T_1 = 80°\text{C} \end{array}\right\} h_1 = 311.39 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1200 \text{ kPa} \\ x = 0 \end{array}\right\} h_2 \cong h_{f\,@1200\text{ kPa}} = 117.77 \text{ kJ/kg}$$

Substituting,

$$q_{out} = 311.39 - 117.77 = \mathbf{193.6\ kJ/kg}$$

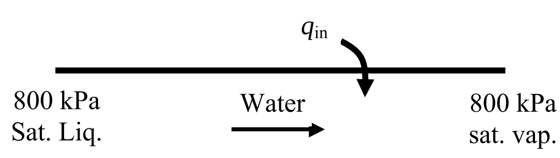
5-120 Water is heated at constant pressure so that it changes a state from saturated liquid to saturated vapor. The heat transfer per unit mass is to be determined.

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

Analysis We take the pipe in which water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}h_1 + \dot{Q}_{in} = \dot{m}h_2$$
$$\dot{Q}_{in} = \dot{m}(h_2 - h_1)$$
$$q_{in} = h_2 - h_1 = h_{fg}$$



where $h_{fg\,@800\text{ kPa}} = 2047.5 \text{ kJ/kg}$ (Table A-5)

Thus,

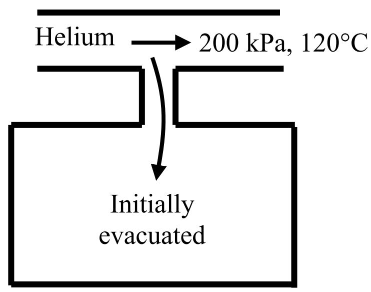
$$q_{in} = \mathbf{2047.5\ kJ/kg}$$

Charging and Discharging Processes

5-121 Helium flows from a supply line to an initially evacuated tank. The flow work of the helium in the supply line and the final temperature of the helium in the tank are to be determined.

Properties The properties of helium are R = 2.0769 kJ/kg.K, c_p = 5.1926 kJ/kg.K, c_v= 3.1156 kJ/kg.K (Table A-2a).

Analysis The flow work is determined from its definition but we first determine the specific volume



$$v = \frac{RT_{\text{line}}}{P} = \frac{(2.0769\ \text{kJ/kg.K})(120 + 273\ \text{K})}{(200\ \text{kPa})} = 4.0811\ \text{m}^3/\text{kg}$$

$$w_{\text{flow}} = Pv = (200\ \text{kPa})(4.0811\ \ \text{m}^3/\text{kg}) = \mathbf{816.2\ kJ/kg}$$

Noting that the flow work in the supply line is converted to sensible internal energy in the tank, the final helium temperature in the tank is determined as follows

$$u_{\text{tank}} = h_{\text{line}}$$

$$h_{\text{line}} = c_p T_{\text{line}} = (5.1926\ \text{kJ/kg.K})(120 + 273\ \text{K}) = 2040.7\ \text{kJ/kg}$$

$$u_{\text{-tank}} = c_v T_{\text{tank}} \longrightarrow 2040.7\ \text{kJ/kg} = (3.1156\ \text{kJ/kg.K})T_{\text{tank}} \longrightarrow T_{\text{tank}} = \mathbf{655.0\ K}$$

Alternative Solution: Noting the definition of specific heat ratio, the final temperature in the tank can also be determined from

$$T_{\text{tank}} = kT_{\text{line}} = 1.667(120 + 273\ \text{K}) = \mathbf{655.1 K}$$

which is practically the same result.

5-122 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

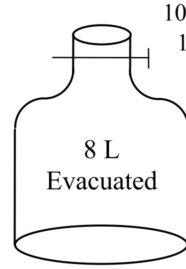
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

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

Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{in} - m_{out} = \Delta m_{system} \quad \rightarrow \quad m_i = m_2 \quad (\text{since } m_{out} = m_{initial} = 0)$$



Energy balance:

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

$$Q_{in} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{out} = E_{initial} = ke \cong pe \cong 0)$$

Combining the two balances:

$$Q_{in} = m_2 (u_2 - h_i)$$

where

$$m_2 = \frac{P_2 \mathbf{V}}{RT_2} = \frac{(100 \text{ kPa})(0.008 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 0.0096 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{array}{l} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{array}$$

Substituting,

$$Q_{in} = (0.0096 \text{ kg})(206.91 - 290.16) \text{ kJ/kg} = -0.8 \text{ kJ}$$

or

$$Q_{out} = \mathbf{0.8 \text{ kJ}}$$

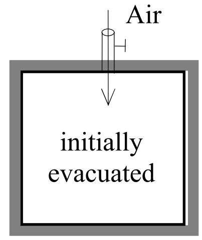
Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

5-123 An insulated rigid tank is evacuated. A valve is opened, and air is allowed to fill the tank until mechanical equilibrium is established. The final temperature in the tank is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The device is adiabatic and thus heat transfer is negligible.

Properties The specific heat ratio for air at room temperature is $k = 1.4$ (Table A-2).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as



Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$$

Energy balance:

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

$$m_i h_i = m_2 u_2 \quad (\text{since } Q \cong W \cong E_{\text{out}} = E_{\text{initial}} = ke \cong pe \cong 0)$$

Combining the two balances:

$$u_2 = h_i \rightarrow c_v T_2 = c_p T_i \rightarrow T_2 = (c_p / c_v) T_i = k T_i$$

Substituting,

$$T_2 = 1.4 \times 290 \text{ K} = 406 \text{ K} = \mathbf{133^\circ C}$$

5-124 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).

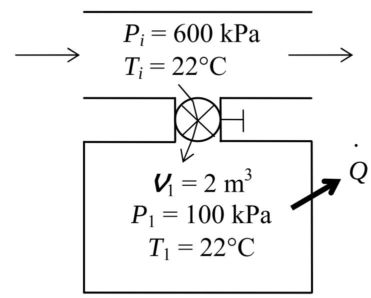
Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The properties of air are (Table A-17)

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

Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

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



Energy balance:

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

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

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

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

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

5-125 A rigid tank initially contains saturated R-134a liquid-vapor mixture. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The final temperature in the tank, the mass of R-134a that entered, and the heat transfer are to be determined.

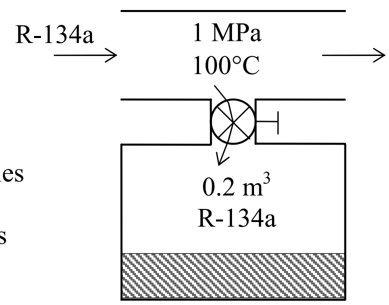
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} T_1 = 8°\text{C} \\ x_1 = 0.7 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.0007887 + 0.7\times(0.052762 - 0.0007887) = 0.03717\ \text{m}^3/\text{kg} \\ u_1 = u_f + x_1 u_{fg} = 62.39 + 0.7\times 172.19 = 182.92\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@800\ \text{kPa}} = 0.02562\ \text{m}^3/\text{kg} \\ u_2 = u_{g@800\ \text{kPa}} = 246.79\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.0\ \text{MPa} \\ T_i = 100°C \end{array}\right\} h_i = 335.06\ \text{kJ/kg}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

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

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(*a*) The tank contains saturated vapor at the final state at 800 kPa, and thus the final temperature is the saturation temperature at this pressure,

$$T_2 = T_{\text{sat @ 800 kPa}} = \mathbf{31.31°C}$$

(*b*) The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{0.2\ \text{m}^3}{0.03717\ \text{m}^3/\text{kg}} = 5.38\ \text{kg}$$

$$m_2 = \frac{V}{v_2} = \frac{0.2\ \text{m}^3}{0.02562\ \text{m}^3/\text{kg}} = 7.81\ \text{kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 7.81 - 5.38 = \mathbf{2.43\ kg}$$

(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(2.43\ \text{kg})(335.06\ \text{kJ/kg}) + (7.81\ \text{kg})(246.79\ \text{kJ/kg}) - (5.38\ \text{kg})(182.92\ \text{kJ/kg}) \\ &= \mathbf{130\ kJ} \end{aligned}$$

5-126 A cylinder initially contains saturated liquid-vapor mixture of water. The cylinder is connected to a supply line, and the steam is allowed to enter the cylinder until all the liquid is vaporized. The final temperature in the cylinder and the mass of the steam that entered are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The device is insulated and thus heat transfer is negligible.

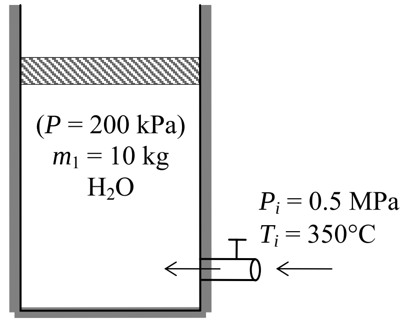
Properties The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.6 \end{array}\right\} h_1 = h_f + x_1 h_{fg}$$

$$= 504.71 + 0.6 \times 2201.6 = 1825.6 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} h_2 = h_{g@200 \text{ kPa}} = 2706.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_i = 0.5 \text{ MPa} \\ T_i = 350°\text{C} \end{array}\right\} h_i = 3168.1 \text{ kJ/kg}$$



Analysis (*a*) The cylinder contains saturated vapor at the final state at a pressure of 200 kPa, thus the final temperature in the cylinder must be

$$T_2 = T_{\text{sat @ 200 kPa}} = \mathbf{120.2°C}$$

(*b*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

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

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

Combining the two relations gives $0 = W_{\text{b,out}} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$

or, $0 = -(m_2 - m_1)h_i + m_2 h_2 - m_1 h_1$

since the boundary work and ΔU combine into ΔH for constant pressure expansion and compression processes. Solving for m_2 and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3168.1 - 1825.6) \text{ kJ/kg}}{(3168.1 - 2706.3) \text{ kJ/kg}} (10 \text{ kg}) = 29.07 \text{ kg}$$

Thus,

$$m_i = m_2 - m_1 = 29.07 - 10 = \mathbf{19.07 \text{ kg}}$$

5-127E A scuba diver's air tank is to be filled with air from a compressed air line. The temperature and mass in the tank at the final state are to be determined.

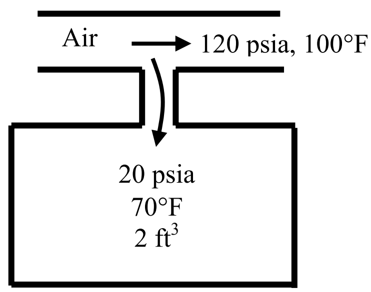
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The gas constant of air is 0.3704 psia·ft^3/lbm·R (Table A-1E). The specific heats of air at room temperature are c_p = 0.240 Btu/lbm·R and c_v= 0.171 Btu/lbm·R (Table A-2E*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

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



Energy balance:

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

$$m_i h_i = m_2 u_2 - m_1 u_1$$

$$m_i c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

Combining the two balances:

$$(m_2 - m_1) c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(20 \text{ psia})(2 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(70+460 \text{ R})} = 0.2038 \text{ lbm}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(120 \text{ psia})(2 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})T_2} = \frac{647.9}{T_2}$$

Substituting,

$$\left(\frac{647.9}{T_2} - 0.2038\right)(0.24)(560) = \frac{647.9}{T_2}(0.171)T_2 - (0.2038)(0.171)(530)$$

whose solution by trial-error or an equation solver such as EES is

$$T_2 = \mathbf{727.4\ R} = \mathbf{267.4°F}$$

The final mass is then

$$m_2 = \frac{647.9}{T_2} = \frac{647.9}{727.4} = \mathbf{0.890\ lbm}$$

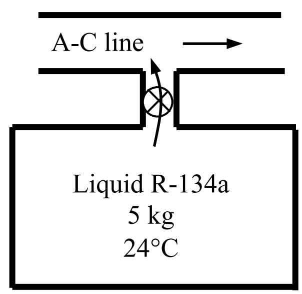
5-128 R-134a from a tank is discharged to an air-conditioning line in an isothermal process. The final quality of the R-134a in the tank and the total heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the exit remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$
$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$
$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + m_e h_e$$

Combining the two balances:

$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$

The initial state properties of R-134a in the tank are

$$\left.\begin{array}{l} T_1 = 24°\text{C} \\ x = 0 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.0008261\ \text{m}^3/\text{kg} \\ u_1 = 84.44\ \text{kJ/kg} \\ h_e = 84.98\ \text{kJ/kg} \end{array} \qquad \text{(Table A-11)}$$

Note that we assumed that the refrigerant leaving the tank is at saturated liquid state, and found the exiting enthalpy accordingly. The volume of the tank is

$$\boldsymbol{V} = m_1 \boldsymbol{v}_1 = (5\ \text{kg})(0.0008261\ \text{m}^3/\text{kg}) = 0.004131\ \text{m}^3$$

The final specific volume in the container is

$$\boldsymbol{v}_2 = \frac{\boldsymbol{V}}{m_2} = \frac{0.004131\ \text{m}^3}{0.25\ \text{kg}} = 0.01652\ \text{m}^3/\text{kg}$$

The final state is now fixed. The properties at this state are (Table A-11)

$$\left.\begin{array}{l} T_2 = 24°\text{C} \\ \boldsymbol{v}_2 = 0.01652\ \text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \dfrac{0.01652 - 0.0008261}{0.031834 - 0.0008261} = \mathbf{0.5061} \\ u_2 = u_f + x_2 u_{fg} = 84.44\ \text{kJ/kg} + (0.5061)(158.65\ \text{kJ/kg}) = 164.73\ \text{kJ/kg} \end{array}$$

Substituting into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e \\ &= (0.25\ \text{kg})(164.73\ \text{kJ/kg}) - (5\ \text{kg})(84.44\ \text{kJ/kg}) + (4.75\ \text{kg})(84.98\ \text{kJ/kg}) \\ &= \mathbf{22.64\ kJ} \end{aligned}$$

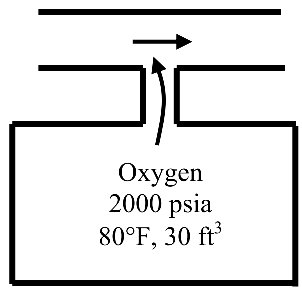
5-129E Oxygen is supplied to a medical facility from 10 compressed oxygen tanks in an isothermal process. The mass of oxygen used and the total heat transfer to the tanks are to be determined.

Assumptions **1** This is an unsteady process but it can be analyzed as a uniform-flow process. **2** Oxygen is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved.

Properties The gas constant of oxygen is 0.3353 psia·ft^3/lbm·R (Table A-1E). The specific heats of oxygen at room temperature are c_p = 0.219 Btu/lbm·R and c_v= 0.157 Btu/lbm·R (Table A-2E*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:



$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$
$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$
$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + m_e h_e$$
$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$

Combining the two balances:

$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses, and the mass used are

$$m_1 = \frac{P_1 \mathbf{V}}{RT_1} = \frac{(2000 \text{ psia})(30 \text{ ft}^3)}{(0.3353 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(80+460 \text{ R})} = 331.4 \text{ lbm}$$

$$m_2 = \frac{P_2 \mathbf{V}}{RT_2} = \frac{(100 \text{ psia})(30 \text{ ft}^3)}{(0.3353 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(80+460 \text{ R})} = 16.57 \text{ lbm}$$

$$m_e = m_1 - m_2 = 331.4 - 16.57 = \mathbf{314.8\ lbm}$$

Substituting into the energy balance equation,

$$Q_{\text{in}} = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$
$$= (16.57)(0.157)(540) - (331.4)(0.157)(540) + (314.8)(0.219)(540)$$
$$= \mathbf{10{,}540\ Btu}$$

5-130 The air in an insulated, rigid compressed-air tank is released until the pressure in the tank reduces to a specified value. The final temperature of the air in the tank is to be determined.

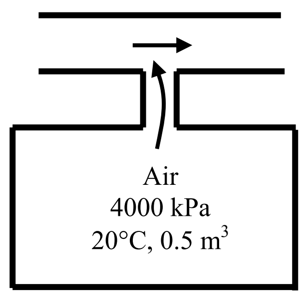
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The tank is well-insulated, and thus there is no heat transfer.

Properties The gas constant of air is 0.287 kPa·m³/kg·K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 kJ/kg·K and c_v= 0.718 kJ/kg·K (Table A-2*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$



Energy balance:

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

$$-m_e h_e = m_2 u_2 - m_1 u_1$$
$$0 = m_2 u_2 - m_1 u_1 + m_e h_e$$
$$0 = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$

Combining the two balances:

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(4000\text{ kPa})(0.5\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})} = 23.78\text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(2000\text{ kPa})(0.5\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{3484}{T_2}$$

The temperature of air leaving the tank changes from the initial temperature in the tank to the final temperature during the discharging process. We assume that the temperature of the air leaving the tank is the average of initial and final temperatures in the tank. Substituting into the energy balance equation gives

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

$$0 = \frac{3484}{T_2}(0.718)T_2 - (23.78)(0.718)(293) + \left(23.78 - \frac{3484}{T_2}\right)(1.005)\left(\frac{293 + T_2}{2}\right)$$

whose solution by trial-error or by an equation solver such as EES is

$$T_2 = \mathbf{241\,K} = \mathbf{-32°C}$$

5-131E Steam is supplied from a line to a weighted piston-cylinder device. The final temperature (and quality if appropriate) of the steam in the piston cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

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

Energy balance:

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

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$
$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

The boundary work is determined from

$$W_{b,\text{out}} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = P(m_2 \boldsymbol{v}_2 - m_1 \boldsymbol{v}_1) = P m_2 \boldsymbol{v}_2$$

Substituting, the energy balance equation simplifies into

$$P m_2 \boldsymbol{v}_2 = m_2 (h_i - u_2)$$
$$P \boldsymbol{v}_2 = h_i - u_2$$

The enthalpy of steam at the inlet is

$$\left. \begin{aligned} P_1 &= 300 \text{ psia} \\ T_1 &= 450°\text{F} \end{aligned} \right\} h_i = 1226.4 \text{ Btu/lbm} \quad \text{(Table A-6E)}$$

Substituting this value into the energy balance equation and using an iterative solution of this equation gives (or better yet using EES software)

$$T_2 = \mathbf{425.1°F}$$
$$u_2 = 1135.5 \text{ Btu/lbm}$$
$$\boldsymbol{v}_2 = 2.4575 \text{ Btu/lbm}$$

The final mass is

$$m_2 = \frac{\boldsymbol{V}_2}{\boldsymbol{v}_2} = \frac{10 \text{ ft}^3}{2.4575 \text{ ft}^3\text{/lbm}} = 4.069 \text{ kg}$$

and the work produced is

$$W_{b,\text{out}} = P\boldsymbol{V}_2 = (200 \text{ psia})(10 \text{ ft}^3)\left(\frac{1 \text{ Btu}}{5.404 \text{ psia} \cdot \text{ft}^3}\right) = \mathbf{370.1 Btu}$$

5-132E Oxygen is supplied from a line to a weighted piston-cylinder device. The final temperature of the oxygen in the piston cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic. **4** Oxygen is an ideal gas with constant specific heats.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

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

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$
$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$
$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

The boundary work is determined from

$$W_{b,\text{out}} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = P(m_2 \boldsymbol{v}_2 - m_1 \boldsymbol{v}_1) = P m_2 \boldsymbol{v}_2$$

Substituting, the energy balance equation simplifies into

$$P m_2 \boldsymbol{v}_2 = m_2 (h_i - u_2)$$
$$P \boldsymbol{v}_2 = h_i - u_2$$
$$RT_2 = c_p T_i - c_v T_2$$

Solving for the final temperature,

$$RT_2 = c_p T_i - c_v T_2 \longrightarrow T_2 = \frac{c_p}{R + c_v} T_i = \frac{c_p}{c_p} T_i = T_i = \textbf{450°F}$$

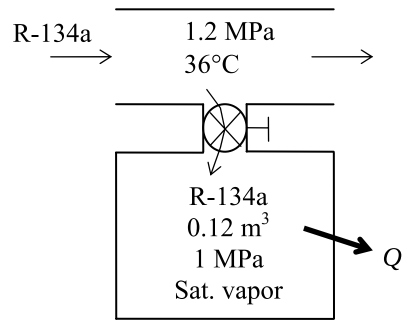
The work produced is

$$W_{b,\text{out}} = P\boldsymbol{V}_2 = (200\ \text{psia})(10\ \text{ft}^3)\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) = \textbf{370.1 Btu}$$

5-133 A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered and the heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)



$$\left.\begin{array}{l}P_1 = 1 \text{ MPa} \\ \text{sat.vapor}\end{array}\right\} \begin{array}{l}\boldsymbol{v}_1 = \boldsymbol{v}_{g@1 \text{ MPa}} = 0.02031 \text{ m}^3/\text{kg} \\ u_1 = u_{g@1 \text{ MPa}} = 250.68 \text{ kJ/kg}\end{array}$$

$$\left.\begin{array}{l}P_2 = 1.2 \text{ MPa} \\ \text{sat. liquid}\end{array}\right\} \begin{array}{l}\boldsymbol{v}_2 = \boldsymbol{v}_{f@1.2 \text{ MPa}} = 0.0008934 \text{ m}^3/\text{kg} \\ u_2 = u_{f@1.2 \text{ MPa}} = 116.70 \text{ kJ/kg}\end{array}$$

$$\left.\begin{array}{l}P_i = 1.2 \text{ MPa} \\ T_i = 36°\text{C}\end{array}\right\} h_i = h_{f@36°\text{C}} = 102.30 \text{ kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

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

Energy balance:

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

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(*a*) The initial and the final masses in the tank are

$$m_1 = \frac{\mathcal{V}_1}{\boldsymbol{v}_1} = \frac{0.12 \text{ m}^3}{0.02031 \text{ m}^3/\text{kg}} = 5.91 \text{ kg}$$

$$m_2 = \frac{\mathcal{V}_2}{\boldsymbol{v}_2} = \frac{0.12 \text{ m}^3}{0.0008934 \text{ m}^3/\text{kg}} = 134.31 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 134.31 - 5.91 = \mathbf{128.4 \text{ kg}}$$

(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(128.4 \text{ kg})(102.30 \text{ kJ/kg}) + (134.31 \text{ kg})(116.70 \text{ kJ/kg}) - (5.91 \text{ kg})(250.68 \text{ kJ/kg}) \\ &= \mathbf{1057 \text{ kJ}} \end{aligned}$$

5-134 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of the mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer is to be determined.

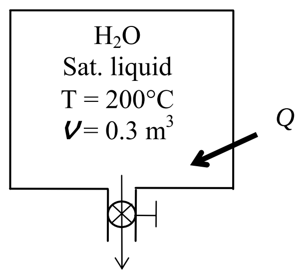
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

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

$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@200°\text{C}} = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = u_{f@200°\text{C}} = 850.46 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_e = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@200°\text{C}} = 852.26 \text{ kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as



Mass balance: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_e = m_1 - m_2$

Energy balance:

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

$$Q_{in} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.3 \text{ m}^3}{0.001157 \text{m}^3/\text{kg}} = 259.4 \text{ kg}$$

$$m_2 = \tfrac{1}{2} m_1 = \tfrac{1}{2}(259.4 \text{ kg}) = 129.7 \text{ kg}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 259.4 - 129.7 = 129.7 \text{ kg}$$

Now we determine the final internal energy,

$$v_2 = \frac{V}{m_2} = \frac{0.3 \text{ m}^3}{129.7 \text{ kg}} = 0.002313 \text{ m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002313 - 0.001157}{0.12721 - 0.001157} = 0.009171$$

$$\left.\begin{array}{l} T_2 = 200°\text{C} \\ x_2 = 0.009171 \end{array}\right\} u_2 = u_f + x_2 u_{fg} = 850.46 + (0.009171)(1743.7) = 866.46 \text{ kJ/kg}$$

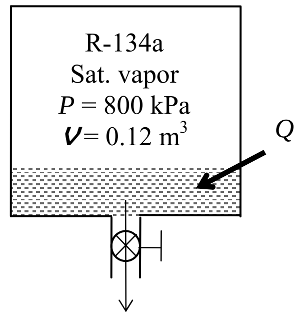
Then the heat transfer during this process is determined from the energy balance by substitution to be

$$\begin{aligned} Q &= (129.7 \text{ kg})(852.26 \text{ kJ/kg}) + (129.7 \text{ kg})(866.46 \text{ kJ/kg}) - (259.4 \text{ kg})(850.46 \text{ kJ/kg}) \\ &= \mathbf{2308 \text{ kJ}} \end{aligned}$$

5-135 A rigid tank initially contains saturated liquid-vapor mixture of refrigerant-134a. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank at constant pressure until no liquid remains inside. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of R-134a are (Tables A-11 through A-13)



$$P_1 = 800 \text{ kPa} \rightarrow \boldsymbol{v}_f = 0.0008458 \text{ m}^3/\text{kg}, \boldsymbol{v}_g = 0.025621 \text{ m}^3/\text{kg}$$
$$u_f = 94.79 \text{ kJ/kg}, \quad u_g = 246.79 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{g@800 \text{ kPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 800 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@800 \text{ kPa}} = 95.47 \text{ kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

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

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{\boldsymbol{V}_f}{\boldsymbol{v}_f} + \frac{\boldsymbol{V}_g}{\boldsymbol{v}_g} = \frac{0.12 \times 0.25 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} + \frac{0.12 \times 0.75 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 35.47 + 3.513 = 38.98 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (35.47)(94.79) + (3.513)(246.79) = 4229.2 \text{ kJ}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{0.12 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 4.684 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 38.98 - 4.684 = 34.30 \text{ kg}$$

$$Q_{\text{in}} = (34.30 \text{ kg})(95.47 \text{ kJ/kg}) + (4.684 \text{ kg})(246.79 \text{ kJ/kg}) - 4229 \text{ kJ} = \mathbf{201.2 \text{ kJ}}$$

5-136E A rigid tank initially contains saturated liquid-vapor mixture of R-134a. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until all the liquid in the tank disappears. The amount of heat transfer is to be determined.

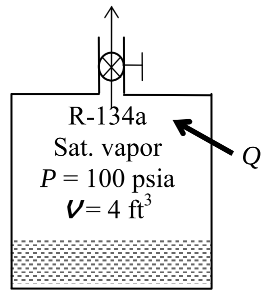
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Properties The properties of R-134a are (Tables A-11E through A-13E)

$$P_1 = 100 \text{ psia} \rightarrow v_f = 0.01332 \text{ ft}^3/\text{lbm},\ v_g = 0.4776 \text{ ft}^3/\text{lbm}$$
$$u_f = 37.623 \text{ Btu/lbm},\ u_g = 104.99 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@100 \text{ psia}} = 0.4776 \text{ ft}^3/\text{lbm} \\ u_2 = u_{g@100 \text{ psia}} = 104.99 \text{ Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_e = 100 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} h_e = h_{g@100 \text{ psia}} = 113.83 \text{ Btu/lbm}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

Energy balance:

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

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{4 \times 0.2 \text{ ft}^3}{0.01332 \text{ ft}^3/\text{lbm}} + \frac{4 \times 0.8 \text{ ft}^3}{0.4776 \text{ ft}^3/\text{lbm}} = 60.04 + 6.70 = 66.74 \text{ lbm}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (60.04)(37.623) + (6.70)(104.99) = 2962 \text{ Btu}$$

$$m_2 = \frac{V}{v_2} = \frac{4 \text{ ft}^3}{0.4776 \text{ ft}^3/\text{lbm}} = 8.375 \text{ lbm}$$

Then from the mass and energy balances,

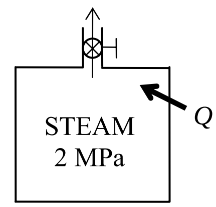
$$m_e = m_1 - m_2 = 66.74 - 8.375 = 58.37 \text{ lbm}$$

$$\begin{aligned} Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (58.37 \text{ lbm})(113.83 \text{ Btu/lbm}) + (8.375 \text{ lbm})(104.99 \text{ Btu/lbm}) - 2962 \text{ Btu} \\ &= \mathbf{4561\ Btu} \end{aligned}$$

5-137 A rigid tank initially contains superheated steam. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until the temperature rises to 500°C. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the steam leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

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



$$\left.\begin{array}{l} P_1 = 2 \text{ MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.12551 \text{ m}^3/\text{kg} \\ u_1 = 2773.2 \text{ kJ/kg}, \quad h_1 = 3024.2 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2 \text{ MPa} \\ T_2 = 500°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.17568 \text{ m}^3/\text{kg} \\ u_2 = 3116.9 \text{ kJ/kg}, \quad h_2 = 3468.3 \text{ kJ/kg} \end{array}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

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

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The state and thus the enthalpy of the steam leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values. Thus,

$$h_e \cong \frac{h_1 + h_2}{2} = \frac{3024.2 + 3468.3 \text{ kJ/kg}}{2} = 3246.2 \text{ kJ/kg}$$

The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.2 \text{ m}^3}{0.12551 \text{ m}^3/\text{kg}} = 1.594 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.2 \text{ m}^3}{0.17568 \text{ m}^3/\text{kg}} = 1.138 \text{kg}$$

Then from the mass and energy balance relations,

$$m_e = m_1 - m_2 = 1.594 - 1.138 = 0.456 \text{ kg}$$

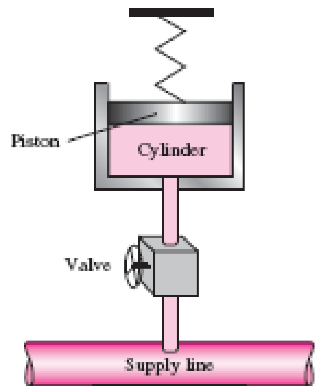
$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (0.456 \text{ kg})(3246.2 \text{ kJ/kg}) + (1.138 \text{ kg})(3116.9 \text{ kJ/kg}) - (1.594 \text{ kg})(2773.2 \text{ kJ/kg}) \\ &= \mathbf{606.8 \text{ kJ}} \end{aligned}$$

5-138 Steam is supplied from a line to a piston-cylinder device equipped with a spring. The final temperature (and quality if appropriate) of the steam in the cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \longrightarrow m_i = m_2$



Energy balance:

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

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$

$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$

Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

Because of the spring, the relation between the pressure and volume is a linear relation. According to the data in the problem statement,

$$P - 300 = \frac{2700}{5} \mathbf{V}$$

The final vapor volume is then

$$\mathbf{V}_2 = \frac{5}{2700}(1500 - 300) = 2.222 \text{ m}^3$$

The work needed to compress the spring is

$$W_{b,\text{out}} = \int P d\mathbf{V} = \int_0^{\mathbf{V}_2} \left(\frac{2700}{5} \mathbf{V} + 300 \right) d\mathbf{V} = \frac{2700}{2 \times 5} \mathbf{V}_2^2 + 300 \mathbf{V}_2 = 270 \times 2.222^2 + 300 \times 2.222 = \mathbf{2000 \text{ kJ}}$$

The enthalpy of steam at the inlet is

$$\left. \begin{array}{l} P_1 = 1500 \text{ kPa} \\ T_1 = 200°\text{C} \end{array} \right\} h_i = 2796.0 \text{ kJ/kg} \quad (\text{Table A-6})$$

Substituting the information found into the energy balance equation gives

$$W_{b,\text{out}} = m_2 (h_i - u_2) \longrightarrow W_{b,\text{out}} = \frac{\mathbf{V}_2}{\boldsymbol{v}_2}(h_i - u_2) \longrightarrow 2000 = \frac{2.222}{\boldsymbol{v}_2}(2796.0 - u_2)$$

Using an iterative solution of this equation with steam tables gives (or better yet using EES software)

$$T_2 = \mathbf{233.2°C}$$

$$u_2 = 2664.8 \text{ kJ/kg}$$

$$\boldsymbol{v}_2 = 0.1458 \text{ m}^3/\text{kg}$$

5-139 Air is supplied from a line to a piston-cylinder device equipped with a spring. The final temperature of the steam in the cylinder and the total work produced as the device is filled are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** The process is adiabatic. **4** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is 0.287 kPa·m^3/kg·K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 kJ/kg·K and c_v= 0.718 kJ/kg·K (Table A-2*a*).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, and also noting that the initial mass in the system is zero, the mass and energy balances for this uniform-flow system can be expressed as

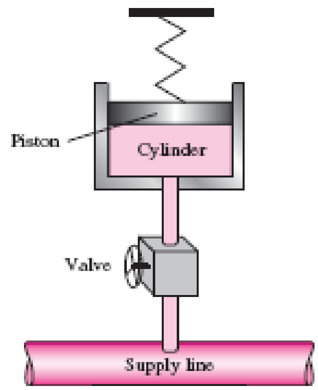
Mass balance: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \longrightarrow m_i = m_2$$

Energy balance:

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

$$m_i h_i - W_{b,\text{out}} = m_2 u_2$$

$$W_{b,\text{out}} = m_i h_i - m_2 u_2$$



Combining the two balances:

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

Because of the spring, the relation between the pressure and volume is a linear relation. According to the data in the problem statement,

$$P - 300 = \frac{2700}{5} V$$

The final air volume is then

$$V_2 = \frac{5}{2700}(2000 - 300) = 3.148 \text{ m}^3$$

The work needed to compress the spring is

$$W_{b,\text{out}} = \int P dV = \int_0^{V_2} \left(\frac{2700}{5} V + 300 \right) dV = \frac{2700}{2 \times 5} V_2^2 + 300 V_2 = 270 \times 3.148^2 + 300 \times 3.148 = \mathbf{3620 \text{ kJ}}$$

Substituting the information found into the energy balance equation gives

$$W_{b,\text{out}} = m_2 (h_i - u_2)$$

$$W_{b,\text{out}} = \frac{P_2 V_2}{RT_2} (c_p T_i - c_v T_2)$$

$$3620 = \frac{2000 \times 3.148}{(0.287) T_2} (1.005 \times 523 - 0.718 \times T_2)$$

The final temperature is then

$$T_2 = \mathbf{595.2 \text{ K}} = \mathbf{322.2°C}$$

5-140 A hot-air balloon is considered. The final volume of the balloon and work produced by the air inside the balloon as it expands the balloon skin are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There is no heat transfer.

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

Analysis The specific volume of the air at the entrance and exit, and in the balloon is

$$\boldsymbol{v} = \frac{RT}{P} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(35+273\ \text{K})}{100\ \text{kPa}} = 0.8840\ \text{m}^3/\text{kg}$$

The mass flow rate at the entrance is then

$$\dot{m}_i = \frac{A_i V_i}{\boldsymbol{v}} = \frac{(1\ \text{m}^2)(2\ \text{m/s})}{0.8840\ \text{m}^3/\text{kg}} = 2.262\ \text{kg/s}$$

while that at the outlet is

$$\dot{m}_e = \frac{A_e V_e}{\boldsymbol{v}} = \frac{(0.5\ \text{m}^2)(1\ \text{m/s})}{0.8840\ \text{m}^3/\text{kg}} = 0.5656\ \text{kg/s}$$

Applying a mass balance to the balloon,

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$m_i - m_e = m_2 - m_1$$
$$m_2 - m_1 = (\dot{m}_i - \dot{m}_e)\Delta t = [(2.262 - 0.5656)\ \text{kg/s}](2\times 60\ \text{s}) = 203.6\ \text{kg}$$

The volume in the balloon then changes by the amount

$$\Delta\boldsymbol{V} = (m_2 - m_1)\boldsymbol{v} = (203.6\ \text{kg})(0.8840\ \text{m}^3/\text{kg}) = 180\ \text{m}^3$$

and the final volume of the balloon is

$$\boldsymbol{V}_2 = \boldsymbol{V}_1 + \Delta\boldsymbol{V} = 75 + 180 = \mathbf{255\ m^3}$$

In order to push back the boundary of the balloon against the surrounding atmosphere, the amount of work that must be done is

$$W_{b,\text{out}} = P\Delta\boldsymbol{V} = (100\ \text{kPa})(180\ \text{m}^3)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = \mathbf{18{,}000\ kJ}$$

5-141 An insulated rigid tank initially contains helium gas at high pressure. A valve is opened, and half of the mass of helium is allowed to escape. The final temperature and pressure in the tank are to be determined.

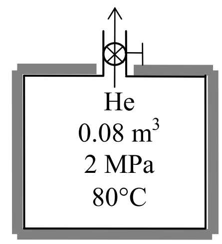
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The tank is insulated and thus heat transfer is negligible. **5** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is k =1.667 (Table A-2).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

$$m_2 = \tfrac{1}{2} m_1 \text{ (given)} \longrightarrow m_e = m_2 = \tfrac{1}{2} m_1$$



Energy balance:

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

$$-m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong Q \cong ke \cong pe \cong 0)$$

Note that the state and thus the enthalpy of helium leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values.

Combining the mass and energy balances: $0 = \tfrac{1}{2} m_1 h_e + \tfrac{1}{2} m_1 u_2 - m_1 u_1$

Dividing by $m_1/2$ $\quad 0 = h_e + u_2 - 2u_1 \;\; or \;\; 0 = c_p \dfrac{T_1 + T_2}{2} + c_v T_2 - 2c_v T_1$

Dividing by c_v: $\quad 0 = k(T_1 + T_2) + 2T_2 - 4T_1 \quad$ since $k = c_p / c_v$

Solving for T_2: $\quad T_2 = \dfrac{(4-k)}{(2+k)} T_1 = \dfrac{(4-1.667)}{(2+1.667)}(353 \text{ K}) = \mathbf{225 \text{ K}}$

The final pressure in the tank is

$$\frac{P_1 \boldsymbol{V}}{P_2 \boldsymbol{V}} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow P_2 = \frac{m_2 T_2}{m_1 T_2} P_1 = \frac{1}{2}\frac{225}{353}(2000 \text{ kPa}) = \mathbf{637 \text{ kPa}}$$

5-142E An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to 30 psia. The amount of electrical work transferred is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

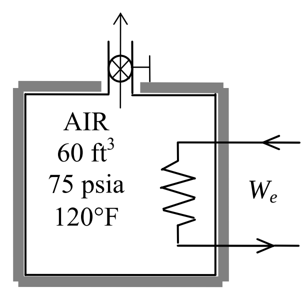
Properties The gas constant of air is R =0.3704 psia.ft^3/lbm.R (Table A-1E). The properties of air are (Table A-17E)

$$T_i = 580 \text{ R} \longrightarrow h_i = 138.66 \text{ Btu/lbm}$$
$$T_1 = 580 \text{ R} \longrightarrow u_1 = 98.90 \text{ Btu/lbm}$$
$$T_2 = 580 \text{ R} \longrightarrow u_2 = 98.90 \text{ Btu/lbm}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$



Energy balance:

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

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(75 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(580 \text{ R})} = 20.95 \text{ lbm}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(30 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(580 \text{ R})} = 8.38 \text{ lbm}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 20.95 - 8.38 = 12.57 \text{ lbm}$$

$$\begin{aligned} W_{\text{e,in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (12.57 \text{ lbm})(138.66 \text{ Btu/lbm}) + (8.38 \text{ lbm})(98.90 \text{ Btu/lbm}) - (20.95 \text{ lbm})(98.90 \text{ Btu/lbm}) \\ &= \mathbf{500 \ Btu} \end{aligned}$$

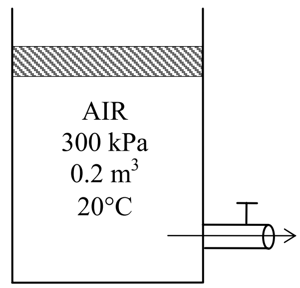
5-143 A vertical cylinder initially contains air at room temperature. Now a valve is opened, and air is allowed to escape at constant pressure and temperature until the volume of the cylinder goes down by half. The amount air that left the cylinder and the amount of heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions. **4** Air is an ideal gas with constant specific heats. **5** The direction of heat transfer is to the cylinder (will be verified).

Properties The gas constant of air is $R = 0.287$ kPa.m³/kg.K (Table A-1).

Analysis (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:



$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \;\rightarrow\; m_e = m_1 - m_2$$

Energy balance:

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

$$Q_{\text{in}} + W_{\text{b,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

The initial and the final masses of air in the cylinder are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(300\text{ kPa})(0.2\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 0.714\text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(300\text{ kPa})(0.1\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 0.357\text{ kg} = \tfrac{1}{2}m_1$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 0.714 - 0.357 = \mathbf{0.357\text{ kg}}$$

(*b*) This is a constant pressure process, and thus the W_b and the ΔU terms can be combined into ΔH to yield

$$Q = m_e h_e + m_2 h_2 - m_1 h_1$$

Noting that the temperature of the air remains constant during this process, we have

$$h_i = h_1 = h_2 = h.$$

Also,

$$m_e = m_2 = \tfrac{1}{2}m_1 .$$

Thus,

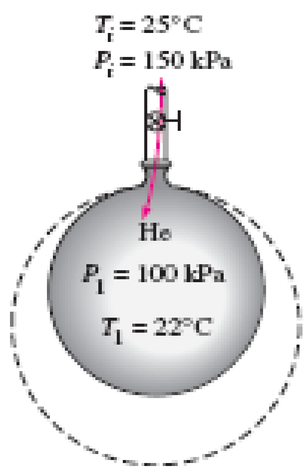
$$Q = \left(\tfrac{1}{2}m_1 + \tfrac{1}{2}m_1 - m_1\right)h = \mathbf{0}$$

5-144 A balloon is initially filled with helium gas at atmospheric conditions. The tank is connected to a supply line, and helium is allowed to enter the balloon until the pressure rises from 100 to 150 kPa. The final temperature in the balloon is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Helium is an ideal gas with constant specific heats. **3** The expansion process is quasi-equilibrium. **4** Kinetic and potential energies are negligible. **5** There are no work interactions involved other than boundary work. **6** Heat transfer is negligible.

Properties The gas constant of helium is R = 2.0769 kJ/kg·K (Table A-1). The specific heats of helium are c_p = 5.1926 and c_v = 3.1156 kJ/kg·K (Table A-2a).

Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as



Mass balance:

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

Energy balance:

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

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(65\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295\text{ K})} = 10.61\text{ kg}$$

$$\frac{P_1}{P_2} = \frac{V_1}{V_2} \longrightarrow V_2 = \frac{P_2}{P_1}V_1 = \frac{150\text{ kPa}}{100\text{ kPa}}(65\text{ m}^3) = 97.5\text{ m}^3$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(150\text{ kPa})(97.5\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(T_2\text{ K})} = \frac{7041.74}{T_2}\text{ kg}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = \frac{7041.74}{T_2} - 10.61\text{ kg}$$

Noting that P varies linearly with V, the boundary work done during this process is

$$W_b = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100+150)\text{kPa}}{2}(97.5 - 65)\text{m}^3 = 4062.5\text{ kJ}$$

Using specific heats, the energy balance relation reduces to

$$W_{\text{b,out}} = m_i c_p T_i - m_2 c_v T_2 + m_1 c_v T_1$$

Substituting,

$$4062.5 = \left(\frac{7041.74}{T_2} - 10.61\right)(5.1926)(298) - \frac{7041.74}{T_2}(3.1156)T_2 + (10.61)(3.1156)(295)$$

It yields $\quad T_2 =$ **333.6 K**

5-145 An insulated piston-cylinder device with a linear spring is applying force to the piston. A valve at the bottom of the cylinder is opened, and refrigerant is allowed to escape. The amount of refrigerant that escapes and the final temperature of the refrigerant are to be determined.

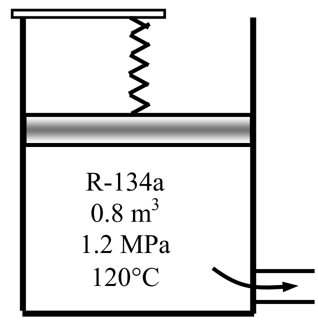
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process assuming that the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible.

Properties The initial properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1.2 \text{ MPa} \\ T_1 = 120°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.02423 \text{ m}^3/\text{kg} \\ u_1 = 325.03 \text{ kJ/kg} \\ h_1 = 354.11 \text{ kJ/kg} \end{array}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$



Energy balance:

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

$$W_{\text{b,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial mass and the relations for the final and exiting masses are

$$m_1 = \frac{V_1}{v_1} = \frac{0.8 \text{ m}^3}{0.02423 \text{ m}^3/\text{kg}} = 33.02 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.5 \text{ m}^3}{v_2}$$

$$m_e = m_1 - m_2 = 33.02 - \frac{0.5 \text{ m}^3}{v_2}$$

Noting that the spring is linear, the boundary work can be determined from

$$W_{\text{b,in}} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(1200 + 600) \text{ kPa}}{2}(0.8 - 0.5)\text{m}^3 = 270 \text{ kJ}$$

Substituting the energy balance,

$$270 - \left(33.02 - \frac{0.5 \text{ m}^3}{v_2}\right)h_e = \left(\frac{0.5 \text{ m}^3}{v_2}\right)u_2 - (33.02 \text{ kg})(325.03 \text{ kJ/kg}) \qquad \text{(Eq. 1)}$$

where the enthalpy of exiting fluid is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder. That is,

$$h_e = \frac{h_1 + h_2}{2} = \frac{(354.11 \text{ kJ/kg}) + h_2}{2}$$

Final state properties of the refrigerant (h_2, u_2, and v_2) are all functions of final pressure (known) and temperature (unknown). The solution may be obtained by a trial-error approach by trying different final state temperatures until Eq. (1) is satisfied. Or solving the above equations simultaneously using an equation solver with built-in thermodynamic functions such as EES, we obtain

T_2 = **96.8°C**, m_e = **22.47 kg**, h_2 = 336.20 kJ/kg,
u_2 = 307.77 kJ/kg, v_2 = 0.04739 m^3/kg, m_2 = 10.55 kg

5-146 Steam at a specified state is allowed to enter a piston-cylinder device in which steam undergoes a constant pressure expansion process. The amount of mass that enters and the amount of heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid entering the device remains constant. **2** Kinetic and potential energies are negligible.

Properties The properties of steam at various states are (Tables A-4 through A-6)

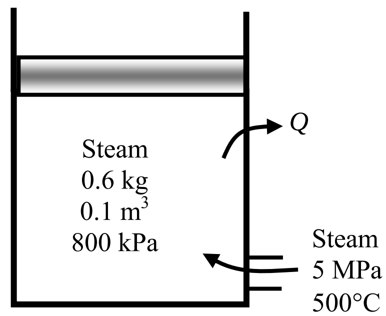
$$\boldsymbol{v}_1 = \frac{\boldsymbol{V}_1}{m_1} = \frac{0.1\text{ m}^3}{0.6\text{ kg}} = 0.16667\text{ m}^3/\text{kg}$$

$$P_2 = P_1$$

$$\left.\begin{array}{l} P_1 = 800\text{ kPa} \\ \boldsymbol{v}_1 = 0.16667\text{ m}^3/\text{kg} \end{array}\right\} u_1 = 2004.4\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800\text{ kPa} \\ T_2 = 250°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.29321\text{ m}^3/\text{kg} \\ u_2 = 2715.9\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 5\text{ MPa} \\ T_i = 500°\text{C} \end{array}\right\} h_i = 3434.7\text{ kJ/kg}$$



Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

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

Energy balance:

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

$$Q_{\text{in}} - W_{\text{b,out}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

Noting that the pressure remains constant, the boundary work is determined from

$$W_{\text{b,out}} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = (800\text{ kPa})(2\times 0.1 - 0.1)\text{m}^3 = 80\text{ kJ}$$

The final mass and the mass that has entered are

$$m_2 = \frac{\boldsymbol{V}_2}{\boldsymbol{v}_2} = \frac{0.2\text{ m}^3}{0.29321\text{ m}^3/\text{kg}} = 0.682\text{ kg}$$

$$m_i = m_2 - m_1 = 0.682 - 0.6 = \mathbf{0.082\ kg}$$

(*b*) Finally, substituting into energy balance equation

$$Q_{\text{in}} - 80\text{ kJ} + (0.082\text{ kg})(3434.7\text{ kJ/kg}) = (0.682\text{ kg})(2715.9\text{ kJ/kg}) - (0.6\text{ kg})(2004.4\text{ kJ/kg})$$

$$Q_{\text{in}} = \mathbf{447.9\ kJ}$$

Review Problems

5-147 A water tank open to the atmosphere is initially filled with water. The tank discharges to the atmosphere through a long pipe connected to a valve. The initial discharge velocity from the tank and the time required to empty the tank are to be determined.

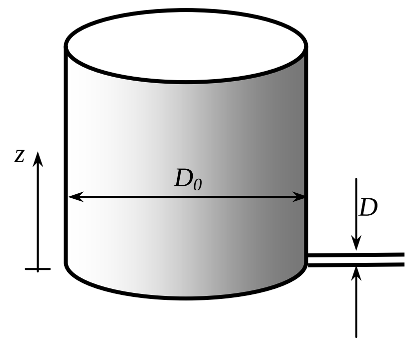
Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The tank is considered to be empty when the water level drops to the center of the valve.

Analysis (*a*) Substituting the known quantities, the discharge velocity can be expressed as

$$V=\sqrt{\frac{2gz}{1.5+fL/D}}=\sqrt{\frac{2gz}{1.5+0.015(100\text{ m})/(0.10\text{ m})}}=\sqrt{0.1212gz}$$

Then the initial discharge velocity becomes

$$V_1=\sqrt{0.1212gz_1}=\sqrt{0.1212(9.81\text{ m/s}^2)(2\text{ m})}=\mathbf{1.54\ m/s}$$



where z is the water height relative to the center of the orifice at that time.

(*b*) The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{V}=A_{\text{pipe}}V_2=\frac{\pi D^2}{4}\sqrt{0.1212gz}$$

Then the amount of water that flows through the pipe during a differential time interval dt is

$$dV=\dot{V}dt=\frac{\pi D^2}{4}\sqrt{0.1212gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV=A_{\text{tank}}(-dz)=-\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{0.1212gz}\,dt=-\frac{\pi D_0^2}{4}dz \;\rightarrow\; dt=-\frac{D_0^2}{D^2}\frac{dz}{\sqrt{0.1212gz}}=-\frac{D_0^2}{D^2\sqrt{0.1212g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t=0$ when $z=z_1$ to $t=t_f$ when $z=0$ (completely drained tank) gives

$$\int_{t=0}^{t_f}dt=-\frac{D_0^2}{D^2\sqrt{0.1212g}}\int_{z=z_1}^{0}z^{-1/2}dz \;\rightarrow\; t_f=-\frac{D_0^2}{D^2\sqrt{0.1212g}}\left|\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0}=\frac{2D_0^2}{D^2\sqrt{0.1212g}}z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f=\frac{2D_0^2}{D^2}\sqrt{\frac{z_1}{0.1212g}}=\frac{2(10\text{ m})^2}{(0.1\text{ m})^2}\sqrt{\frac{2\text{ m}}{0.1212(9.81\text{ m/s}^2)}}=25{,}940\text{ s}=\mathbf{7.21\,h}$$

Discussion The draining time can be shortened considerably by installing a pump in the pipe.

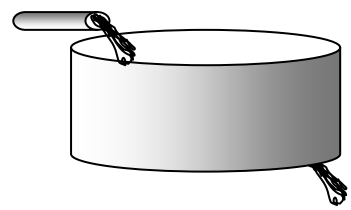
5-148 The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.

Assumptions **1** Water is supplied and discharged steadily. **2** The rate of evaporation of water is negligible. **3** No water is supplied or removed through other means.

Analysis The conservation of mass principle applied to the pool requires that the rate of increase in the amount of water in the pool be equal to the difference between the rate of supply of water and the rate of discharge. That is,

$$\frac{dm_{\text{pool}}}{dt} = \dot{m}_i - \dot{m}_e \quad \rightarrow \quad \dot{m}_i = \frac{dm_{\text{pool}}}{dt} + \dot{m}_e \quad \rightarrow \quad \dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e$$

since the density of water is constant and thus the conservation of mass is equivalent to conservation of volume. The rate of discharge of water is



$$\dot{V}_e = A_e V_e = (\pi D^2/4)V_e = [\pi(0.05\text{ m})^2/4](5\text{ m/s}) = 0.00982\text{ m}^3/\text{s}$$

The rate of accumulation of water in the pool is equal to the cross-section of the pool times the rate at which the water level rises,

$$\frac{dV_{\text{pool}}}{dt} = A_{\text{cross-section}} V_{\text{level}} = (3\text{ m}\times 4\text{ m})(0.015\text{ m/min}) = 0.18\text{ m}^3/\text{min} = 0.00300\text{ m}^3/\text{s}$$

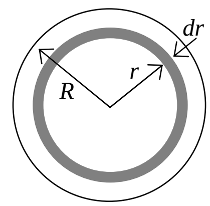
Substituting, the rate at which water is supplied to the pool is determined to be

$$\dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e = 0.003 + 0.00982 = \mathbf{0.01282\ m^3/s}$$

Therefore, water is supplied at a rate of 0.01282 m^3/s = 12.82 L/s.

5-149 A fluid is flowing in a circular pipe. A relation is to be obtained for the average fluid velocity in therms of $V(r)$, R, and r.

Analysis Choosing a circular ring of area $dA = 2\pi r dr$ as our differential area, the mass flow rate through a cross-sectional area can be expressed as



$$\dot{m} = \int_A \rho V(r) dA = \int_0^R \rho V(r) 2\pi r\, dr$$

Solving for V_{avg},

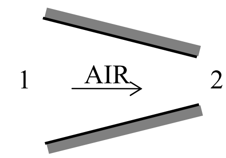
$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R V(r) r\, dr$$

5-150 Air is accelerated in a nozzle. The density of air at the nozzle exit is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 4.18 kg/m^3 at the inlet.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,



$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

$$\rho_2 = \frac{A_1}{A_2}\frac{V_1}{V_2}\rho_1 = 2\frac{120\text{ m/s}}{380\text{ m/s}}(4.18\text{ kg/m}^3) = \mathbf{2.64\ kg/m^3}$$

Discussion Note that the density of air decreases considerably despite a decrease in the cross-sectional area of the nozzle.

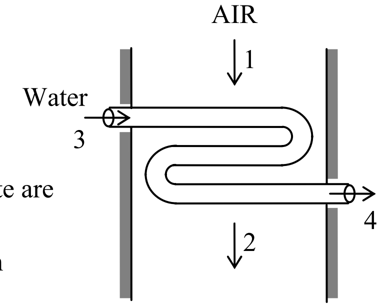
5-151E A heat exchanger that uses hot air to heat cold water is considered. The total flow power and the flow works for both the air and water streams are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm.R =0.06855 Btu/lbm.R (Table A-1E). The specific volumes of water at the inlet and exit are (Table A-4E)

$$\left.\begin{array}{l} P_3 = 20 \text{ psia} \\ T_3 = 50°\text{F} \end{array}\right\} \boldsymbol{v}_3 \cong \boldsymbol{v}_{f\,@\,50°\text{F}} = 0.01602 \text{ ft}^3/\text{lbm}$$

$$\left.\begin{array}{l} P_4 = 17 \text{ psia} \\ T_4 = 90°\text{F} \end{array}\right\} \boldsymbol{v}_4 \cong \boldsymbol{v}_{f\,@\,90°\text{F}} = 0.01610 \text{ ft}^3/\text{lbm}$$



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

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(200+460 \text{ R})}{20 \text{ psia}} = 12.22 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{(100/60) \text{ ft}^3/\text{s}}{12.22 \text{ ft}^3/\text{lbm}} = 0.1364 \text{ lbm/s}$$

Combining the flow work expression with the ideal gas equation of state gives

$$w_{\text{flow}} = P_2\boldsymbol{v}_2 - P_1\boldsymbol{v}_1 = R(T_2 - T_1) = (0.06855 \text{ Btu/lbm}\cdot\text{R})(100-200)\text{R} = \mathbf{-6.855\ Btu/lbm}$$

The flow work of water is

$$\begin{aligned} w_{\text{flow}} &= P_4\boldsymbol{v}_4 - P_3\boldsymbol{v}_3 \\ &= \left[(17 \text{ psia})(0.01610 \text{ ft}^3/\text{lbm}) - (20 \text{ psia})(0.01602 \text{ ft}^3/\text{lbm})\right]\left(\frac{1 \text{ Btu}}{5.404 \text{ psia}\cdot\text{ft}^3}\right) \\ &= \mathbf{-0.00864\ Btu/lbm} \end{aligned}$$

The net flow power for the heat exchanger is

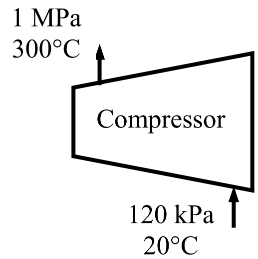
$$\begin{aligned} \dot{W}_{\text{flow}} &= \dot{m}_{\text{air}} w_{\text{flow}} + \dot{m}_{\text{air}} w_{\text{flow}} \\ &= (0.1364 \text{ lbm/s})(-6.855 \text{ Btu/lbm}) + (0.5 \text{ lbm/s})(-0.00864 \text{ Btu/lbm}) \\ &= -0.9393 \text{ Btu/s}\left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}}\right) = \mathbf{-1.329\ hp} \end{aligned}$$

5-152 An air compressor consumes 4.5 kW of power to compress a specified rate of air. The flow work required by the compressor is to be compared to the power used to increase the pressure of the air.

Assumptions **1** Flow through the compressor is steady. **2** Air is an ideal gas.

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

Analysis The specific volume of the air at the inlet is



$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\ \text{K})}{120\ \text{kPa}} = 0.7008\ \text{m}^3/\text{kg}$$

The mass flow rate of the air is

$$\dot{m} = \frac{V_1}{v_1} = \frac{0.010\ \text{m}^3/\text{s}}{0.7008\ \text{m}^3/\text{kg}} = 0.01427\ \text{kg/s}$$

Combining the flow work expression with the ideal gas equation of state gives the flow work as

$$w_{\text{flow}} = P_2 v_2 - P_1 v_1 = R(T_2 - T_1) = (0.287\ \text{kJ/kg}\cdot\text{K})(300-20)\text{K} = 80.36\ \text{kJ/kg}$$

The flow power is

$$\dot{W}_{\text{flow}} = \dot{m} w_{\text{flow}} = (0.01427\ \text{kg/s})(80.36\ \text{kJ/kg}) = \mathbf{1.147\ kW}$$

The remainder of compressor power input is used to increase the pressure of the air:

$$\dot{W} = \dot{W}_{\text{total,in}} - \dot{W}_{\text{flow}} = 4.5 - 1.147 = \mathbf{3.353\ kW}$$

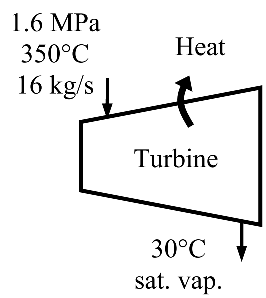
5-153 Steam expands in a turbine whose power production is 9000 kW. The rate of heat lost from the turbine is to be determined.

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

Properties From the steam tables (Tables A-6 and A-4)

$$\left.\begin{array}{l} P_1 = 1.6 \text{ MPa} \\ T_1 = 350°\text{C} \end{array}\right\} h_1 = 3146.0 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_2 = 30°\text{C} \\ x_2 = 1 \end{array}\right\} h_2 = 2555.6 \text{ kJ / kg}$$



Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that there is one inlet and one exiti the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Substituting,

$$\dot{Q}_{out} = (16 \text{ kg/s})(3146.0 - 2555.6) \text{ kJ/kg} - 9000 \text{ kW} = \mathbf{446.4 \text{ kW}}$$

5-154E Nitrogen gas flows through a long, constant-diameter adiabatic pipe. The velocities at the inlet and exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Nitrogen is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions. **5** The re is no heat transfer from the nitrogen.

Properties The specific heat of nitrogen at the room temperature iss c_p = 0.248 Btu/lbm·R (Table A-2E*a*).

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

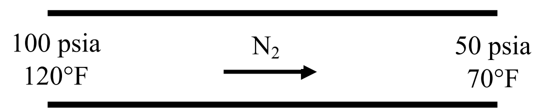
$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}}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

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

$$\frac{V_1^2 - V_2^2}{2} = c_p(T_2 - T_1)$$



Combining the mass balance and ideal gas equation of state yields

$$\dot{m}_1 = \dot{m}_2$$

$$\frac{A_1 V_1}{\boldsymbol{v}_1} = \frac{A_2 V_2}{\boldsymbol{v}_2}$$

$$V_2 = \frac{A_1}{A_2}\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}V_1 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}V_1 = \frac{T_2}{T_1}\frac{P_1}{P_2}V_1$$

Substituting this expression for V_2 into the energy balance equation gives

$$V_1 = \left[\frac{2c_p(T_2 - T_1)}{1 - \left(\dfrac{T_2}{T_1}\dfrac{P_1}{P_2}\right)^2}\right]^{0.5} = \left[\frac{2(0.248)(70 - 120)}{1 - \left(\dfrac{530}{580}\dfrac{100}{50}\right)^2}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)\right]^{0.5} = \mathbf{515\ ft/s}$$

The velocity at the exit is

$$V_2 = \frac{T_2}{T_1}\frac{P_1}{P_2}V_1 = \frac{530}{580}\frac{100}{50}515 = \mathbf{941\ ft/s}$$

5-155 Water at a specified rate is heated by an electrical heater. The current is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The heat losses from the water is negligible.

Properties The specific heat and the density of water are taken to be c_p = 4.18 kJ/kg·°C and ρ = 1 kg/L (Table A-3).

Analysis We take the pipe in which water is heated as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

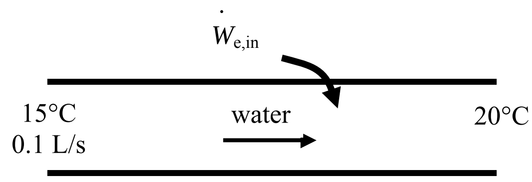
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0$$

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

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

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

$$\mathbf{V}I = \dot{m}c_p(T_2 - T_1)$$



The mass flow rate of the water is

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1\,\text{kg/L})(0.1\,\text{L/s}) = 0.1\,\text{kg/s}$$

Substituting into the energy balance equation and solving for the current gives

$$I = \frac{\dot{m}c_p(T_2 - T_1)}{\mathbf{V}} = \frac{(0.1\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot\text{K})(20-15)\text{K}}{110\,\text{V}}\left(\frac{1000\,\text{VI}}{1\,\text{kJ/s}}\right) = \mathbf{19.0\ A}$$

5-156 Steam flows in an insulated pipe. The mass flow rate of the steam and the speed of the steam at the pipe outlet are to be determined.

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

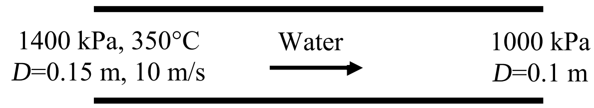
Analysis We take the pipe in which steam flows as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}h_1 = \dot{m}h_2$$

$$h_1 = h_2$$



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

$$\left.\begin{array}{l} P = 1400 \text{ kPa} \\ T = 350°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.20029 \text{ m}^3/\text{kg} \\ h_1 = 3150.1 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1000 \text{ kPa} \\ h_2 = h_1 = 3150.1 \text{ kJ/kg} \end{array}\right\} \boldsymbol{v}_2 = 0.28064 \text{ m}^3/\text{kg}$$

The mass flow rate is

$$\dot{m} = \frac{A_1 V_1}{\boldsymbol{v}_1} = \frac{\pi D_1^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi (0.15 \text{ m})^2}{4}\frac{10 \text{ m/s}}{0.20029 \text{ m}^3/\text{kg}} = \mathbf{0.8823 \text{ kg/s}}$$

The outlet velocity will then be

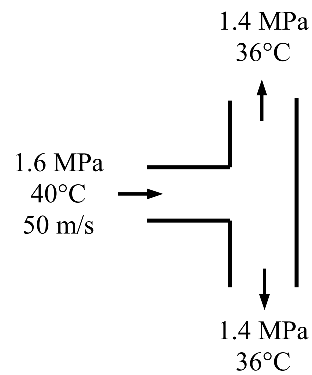
$$V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{4\dot{m}\boldsymbol{v}_2}{\pi D_2^2} = \frac{4(0.8823 \text{ kg/s})(0.28064 \text{ m}^3/\text{kg})}{\pi (0.10 \text{ m})^2} = \mathbf{31.53 \text{ m/s}}$$

5-157 The mass flow rate of a compressed air line is divided into two equal streams by a T-fitting in the line. The velocity of the air at the outlets and the rate of change of flow energy (flow power) across the T-fitting are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** The flow is steady. **3** Since the outlets are identical, it is presumed that the flow divides evenly between the two.

Properties The gas constant of air is R = 0.287 kPa·m³/kg·K (Table A-1).

Analysis The specific volumes of air at the inlet and outlets are



$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(40+273\,\text{K})}{1600\,\text{kPa}} = 0.05614\,\text{m}^3/\text{kg}$$

$$\boldsymbol{v}_2 = \boldsymbol{v}_3 = \frac{RT_2}{P_2} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(36+273\,\text{K})}{1400\,\text{kPa}} = 0.06335\,\text{m}^3/\text{kg}$$

Assuming an even division of the inlet flow rate, the energy balance can be written as

$$\frac{A_1V_1}{\boldsymbol{v}_1} = 2\frac{A_2V_2}{\boldsymbol{v}_2} \longrightarrow V_2 = V_3 = \frac{A_1}{A_2}\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}\frac{V_1}{2} = \frac{0.06335}{0.05614}\frac{50}{2} = \mathbf{28.21\,m/s}$$

The mass flow rate at the inlet is

$$\dot{m}_1 = \frac{A_1V_1}{\boldsymbol{v}_1} = \frac{\pi D_1^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi(0.025\,\text{m})^2}{4}\frac{50\,\text{m/s}}{0.05614\,\text{m}^3/\text{kg}} = 0.4372\,\text{kg/s}$$

while that at the outlets is

$$\dot{m}_2 = \dot{m}_3 = \frac{\dot{m}_1}{2} = \frac{0.4372\,\text{kg/s}}{2} = 0.2186\,\text{kg/s}$$

Substituting the above results into the flow power expression produces

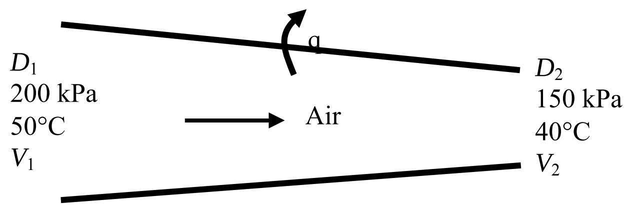
$$\begin{aligned}\dot{W}_{\text{flow}} &= 2\dot{m}_2P_2\boldsymbol{v}_2 - \dot{m}_1P_1\boldsymbol{v}_1 \\ &= 2(0.2186\,\text{kg/s})(1400\,\text{kPa})(0.06335\,\text{m}^3/\text{kg}) - (0.4372\,\text{kg/s})(1600\,\text{kPa})(0.05614\,\text{m}^3/\text{kg}) \\ &= -\mathbf{0.496\,kW}\end{aligned}$$

5-158 Air flows through a non-constant cross-section pipe. The inlet and exit velocities of the air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible. **5** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K. Also, c_p = 1.005 kJ/kg.K for air at room temperature (Table A-2)

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



Mass balance:

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

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \longrightarrow \rho_1 A_1 V_1 = \rho_2 A_2 V_2 \longrightarrow \frac{P_1}{RT_1}\frac{\pi D_1^2}{4}V_1 = \frac{P_2}{RT_2}\frac{\pi D_2^2}{4}V_2 \longrightarrow \frac{P_1}{T_1}D_1^2 V_1 = \frac{P_2}{T_2}D_2^2 V_2$$

Energy balance:

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

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

or $$c_p T_1 + \frac{V_1^2}{2} = c_p T_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting given values into mass and energy balance equations

$$\left(\frac{200 \text{ kPa}}{323 \text{ K}}\right)(1.8 \text{ m})^2 V_1 = \left(\frac{150 \text{ kPa}}{313 \text{ K}}\right)(1.0 \text{ m})^2 V_2 \qquad (1)$$

$$(1.005 \text{ kJ/kg.K})(323 \text{ K}) + \frac{V_1^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = (1.005 \text{ kJ/kg.K})(313 \text{ K}) + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) + 3.3 \text{ kJ/kg} \qquad (2)$$

There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

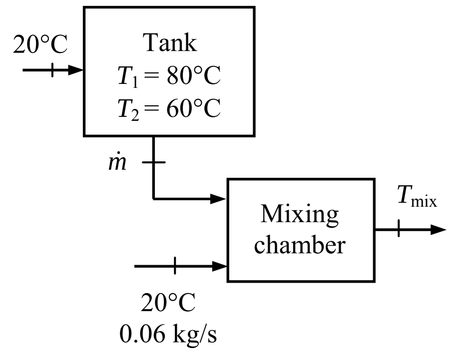
V_1 = **28.6 m/s**

V_2 = **120 m/s**

5-159 A water tank is heated by electricity. The water withdrawn from the tank is mixed with cold water in a chamber. The mass flow rate of hot water withdrawn from the tank and the average temperature of mixed water are to be determined.

Assumptions **1** The process in the mixing chamber is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties The specific heat and density of water are taken to be $c_p = 4.18$ kJ/kg.K, ρ = 100 kg/m^3 (Table A-3).



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

Energy balance:

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

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

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

or $$\dot{m}_{hot} c_p T_{tank,ave} + \dot{m}_{cold} c_p T_{cold} = (\dot{m}_{hot} + \dot{m}_{cold}) c_p T_{mixture} \quad (1)$$

Similarly, an energy balance may be written on the water tank as

$$\left[\dot{W}_{e,in} + \dot{m}_{hot} c_p (T_{cold} - T_{tank,ave})\right]\Delta t = m_{tank} c_p (T_{tank,2} - T_{tank,1}) \quad (2)$$

where $$T_{tank,ave} = \frac{T_{tank,1} + T_{tank,2}}{2} = \frac{80 + 60}{2} = 70°C$$

and $$m_{tank} = \rho \boldsymbol{V} = (1000 \text{ kg/m}^3)(0.060 \text{ m}^3) = 60 \text{ kg}$$

Substituting into Eq. (2),

$$\left[1.6 \text{ kJ/s} + \dot{m}_{hot}(4.18 \text{ kJ/kg.°C})(20 - 70)\text{°C}\right](8 \times 60 \text{ s}) = (60 \text{ kg})(4.18 \text{ kJ/kg.°C})(60 - 80)\text{°C}$$

$$\longrightarrow \dot{m}_{hot} = \mathbf{0.0577 \text{ kg/s}}$$

Substituting into Eq. (1),

$$(0.0577 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70\text{°C}) + (0.06 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(20\text{°C})$$
$$= [(0.0577 + 0.06)\text{kg/s}](4.18 \text{ kJ/kg.°C}) T_{mixture}$$

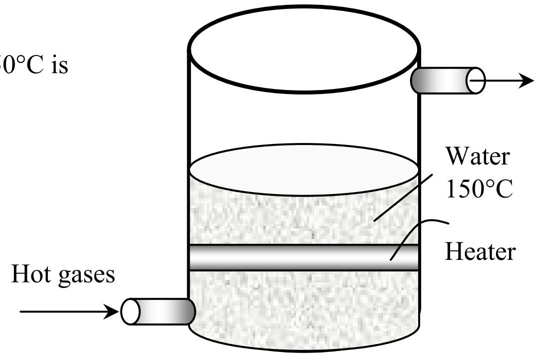
$$\longrightarrow T_{mixture} = \mathbf{44.5°C}$$

5-160 Water is boiled at a specified temperature by hot gases flowing through a stainless steel pipe submerged in water. The rate of evaporation of is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the boiler are negligible.

Properties The enthalpy of vaporization of water at 150°C is h_{fg} = 2113.8 kJ/kg (Table A-4).

Analysis The rate of heat transfer to water is given to be 74 kJ/s. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be



$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{74 \text{ kJ/s}}{2113.8 \text{ kJ/kg}} = \mathbf{0.0350 \text{ kg/s}}$$

5-161 Cold water enters a steam generator at 20°C, and leaves as saturated vapor at T_{sat} = 150°C. The fraction of heat used to preheat the liquid water from 20°C to saturation temperature of 150°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the steam generator are negligible. **3** The specific heat of water is constant at the average temperature.

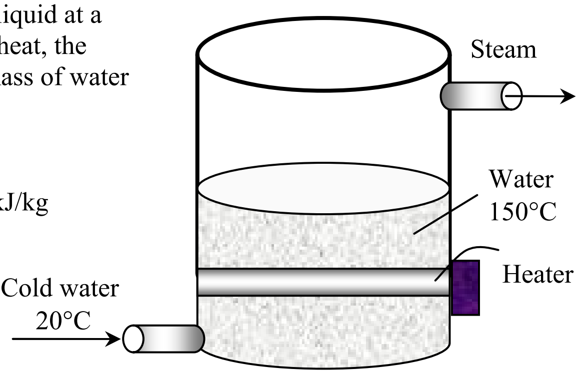
Properties The heat of vaporization of water at 150°C is h_{fg} = 2113.8 kJ/kg (Table A-4), and the specific heat of liquid water is c = 4.18 kJ/kg.°C (Table A-3).

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature. Using the average specific heat, the amount of heat transfer needed to preheat a unit mass of water from 20°C to 150°C is

$$q_{\text{preheating}} = c\Delta T$$
$$= (4.18 \text{ kJ/kg}\cdot°\text{C})(150-20)°\text{C} = 543.4 \text{ kJ/kg}$$

and

$$q_{\text{total}} = q_{\text{boiling}} + q_{\text{preheating}}$$
$$= 2113.8 + 543.4 = 2657.2 \text{ kJ/kg}$$



Therefore, the fraction of heat used to preheat the water is

$$\text{Fraction to preheat} = \frac{q_{\text{preheating}}}{q_{\text{total}}} = \frac{543.4}{2657.2} = \mathbf{0.205} \text{ (or } \mathbf{20.5\%}\text{)}$$

5-162 Cold water enters a steam generator at 20°C and is boiled, and leaves as saturated vapor at boiler pressure. The boiler pressure at which the amount of heat needed to preheat the water to saturation temperature that is equal to the heat of vaporization is to be determined.

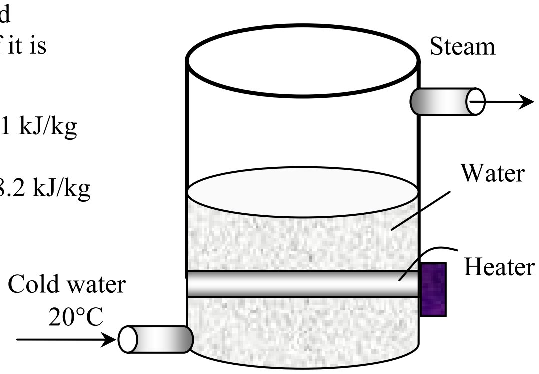
Assumptions Heat losses from the steam generator are negligible.

Properties The enthalpy of liquid water at 20°C is 83.91 kJ/kg. Other properties needed to solve this problem are the heat of vaporization h_{fg} and the enthalpy of saturated liquid at the specified temperatures, and they can be obtained from Table A-4.

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature, and Δh represents the amount of heat needed to preheat a unit mass of water from 20°C to the saturation temperature. Therefore,

$$q_{\text{preheating}} = q_{\text{boiling}}$$
$$(h_{f@T_{\text{sat}}} - h_{f@20°\text{C}}) = h_{fg\,@T_{\text{sat}}}$$
$$h_{f@T_{\text{sat}}} - 83.91\,\text{kJ/kg} = h_{fg\,@T_{\text{sat}}} \rightarrow h_{f@T_{\text{sat}}} - h_{fg\,@T_{\text{sat}}} = 83.91\,\text{kJ/kg}$$

The solution of this problem requires choosing a boiling temperature, reading h_f and h_{fg} at that temperature, and substituting the values into the relation above to see if it is satisfied. By trial and error, (Table A-4)



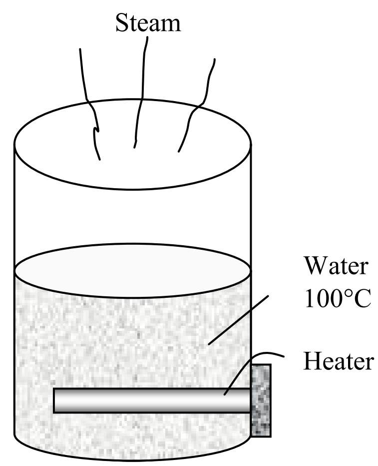
At 310°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1402.0 – 1325.9 = 76.1$ kJ/kg

At 315°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1431.6 – 1283.4 = 148.2$ kJ/kg

The temperature that satisfies this condition is determined from the two values above by interpolation to be 310.6°C. The saturation pressure corresponding to this temperature is **9.94 MPa.**

5-163 Water is boiled at T_{sat} = 100°C by an electric heater. The rate of evaporation of water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the water tank are negligible.



Properties The enthalpy of vaporization of water at 100°C is h_{fg} = 2256.4 kJ/kg (Table A-4).

Analysis Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{W}_{\text{e,boiling}}}{h_{fg}} = \frac{3\text{ kJ/s}}{2256.4\text{ kJ/kg}} = \mathbf{0.00133\ kg/s = 4.79\ kg/h}$$

5-164 Two streams of same ideal gas at different states are mixed in a mixing chamber. The simplest expression for the mixture temperature in a specified format is to be obtained.

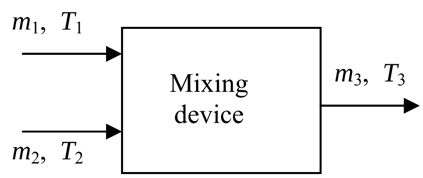
Analysis The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{m}_1 c_p T_1 + \dot{m}_2 c_p T_2 = \dot{m}_3 c_p T_3$$



and,

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

Solving for final temperature, we find

$$T_3 = \frac{\dot{m}_1}{\dot{m}_3} T_1 + \frac{\dot{m}_2}{\dot{m}_3} T_2$$

5-165 An ideal gas expands in a turbine. The volume flow rate at the inlet for a power output of 200 kW is to be determined.

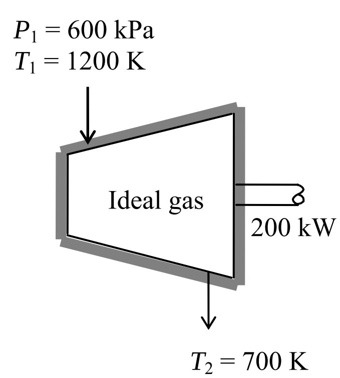
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties The properties of the ideal gas are given as R = 0.30 kPa.m^3/kg.K, c_p = 1.13 kJ/kg·°C, c_v = 0.83 kJ/kg·°C.

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

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

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



which can be rearranged to solve for mass flow rate

$$\dot{m} = \frac{\dot{W}_{out}}{h_1 - h_2} = \frac{\dot{W}_{out}}{c_p (T_1 - T_2)} = \frac{200 \text{ kW}}{(1.13 \text{ kJ/kg.K})(1200 - 700)\text{K}} = 0.354 \text{ kg/s}$$

The inlet specific volume and the volume flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(1200 \text{ K})}{600 \text{ kPa}} = 0.6 \text{ m}^3/\text{kg}$$

Thus,

$$\dot{\boldsymbol{V}} = \dot{m} \boldsymbol{v}_1 = (0.354 \text{ kg/s})(0.6 \text{ m}^3/\text{kg}) = \mathbf{0.212 \text{ m}^3/\text{s}}$$

5-166 Chilled air is to cool a room by removing the heat generated in a large insulated classroom by lights and students. The required flow rate of air that needs to be supplied to the room is to be determined.

Assumptions **1** The moisture produced by the bodies leave the room as vapor without any condensing, and thus the classroom has no latent heat load. **2** Heat gain through the walls and the roof is negligible. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady operating conditions exist.

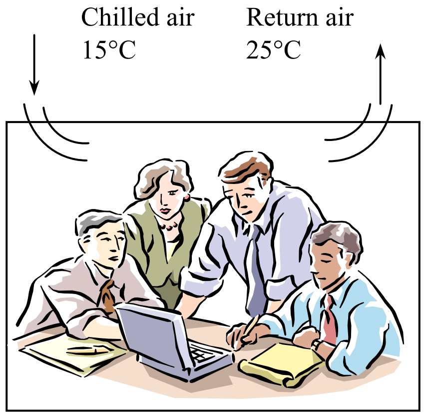
Properties The specific heat of air at room temperature is 1.005 kJ/kg·°C (Table A-2). The average rate of sensible heat generation by a person is given to be 60 W.

Analysis The rate of sensible heat generation by the people in the room and the total rate of sensible internal heat generation are

$$\dot{Q}_{\text{gen, sensible}} = \dot{q}_{\text{gen, sensible}}(\text{No. of people}) = (60\text{ W/person})(150\text{ persons}) = 9000\text{ W}$$

$$\dot{Q}_{\text{total, sensible}} = \dot{Q}_{\text{gen, sensible}} + \dot{Q}_{\text{lighting}} = 9000 + 6000 = 15{,}000\text{ W}$$

Both of these effects can be viewed as heat gain for the chilled air stream, which can be viewed as a steady stream of cool air that is heated as it flows in an imaginary duct passing through the room. The energy balance for this imaginary steady-flow system can be expressed in the rate form as



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

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

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

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

Then the required mass flow rate of chilled air becomes

$$\dot{m}_{air} = \frac{\dot{Q}_{\text{total, sensible}}}{c_p \Delta T} = \frac{15\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot°\text{C})(25-15)°\text{C}} = \mathbf{1.49\text{ kg/s}}$$

Discussion The latent heat will be removed by the air-conditioning system as the moisture condenses outside the cooling coils.

5-167 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

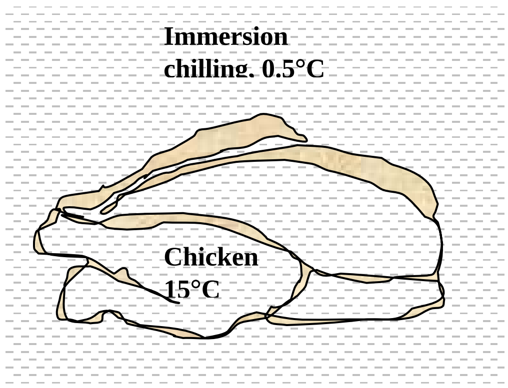
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500\text{ chicken / h})(2.2\text{ kg / chicken}) = 1100\text{ kg / h} = 0.3056\text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as



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

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

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

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m}c_p\Delta T)_{\text{chicken}} = (0.3056\text{ kg/s})(3.54\text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0\text{ kW}}$$

The chiller gains heat from the surroundings at a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 0.056 = 13.056\text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p\Delta T)_{\text{water}}} = \frac{13.056\text{ kW}}{(4.18\text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56\text{ kg/s}}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

5-168 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

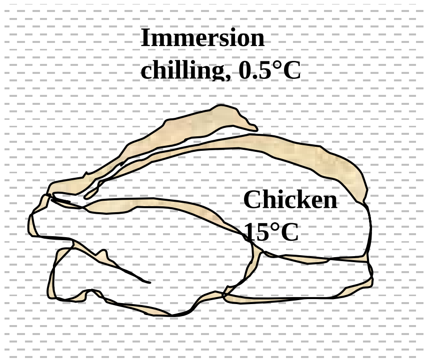
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant. **3** Heat gain of the chiller is negligible.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500 \text{ chicken / h})(2.2 \text{ kg / chicken}) = 1100 \text{ kg / h} = 0.3056 \text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as



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

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

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

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m} c_p \Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.°C})(15-3)\text{°C} = \mathbf{13.0 \ kW}$$

Heat gain of the chiller from the surroundings is negligible. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} = 13.0 \text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p \Delta T)_{\text{water}}} = \frac{13.0 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(2\text{°C})} = \mathbf{1.56 \ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

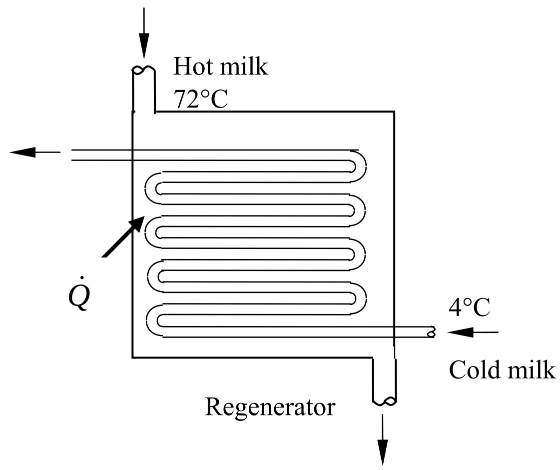
5-169 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year are to be determined.

Assumptions 1 Steady operating conditions exist. **2** The properties of the milk are constant.

Properties The average density and specific heat of milk can be taken to be $\rho_{milk} \cong \rho_{water} = 1$ kg/L and $c_{p,\,milk} = 3.79$ kJ/kg.°C (Table A-3).

Analysis The mass flow rate of the milk is

$$\dot{m}_{milk} = \rho \dot{V}_{milk} = (1\ \text{kg/L})(12\ \text{L/s}) = 12\ \text{kg/s} = 43{,}200\ \text{kg/h}$$



Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot{Q}_{current} = [\dot{m}c_p(T_{pasturization} - T_{refrigeration})]_{milk} = (12\ \text{kg/s})(3.79\ \text{kJ/kg.ëC})(72-4)°\text{C} = 3093\ \text{kJ/s}$$

The proposed regenerator has an effectiveness of ε = 0.82, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{saved} = \varepsilon \dot{Q}_{current} = (0.82)(3093\ \text{kJ/s}) = 2536\ \text{kJ/s}$$

Noting that the boiler has an efficiency of $\eta_{boiler} = 0.82$, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot{Q}_{saved}}{\eta_{boiler}} = \frac{(2536\ \text{kJ/s})}{(0.82)} \frac{(1\ \text{therm})}{(105{,}500\ \text{kJ})} = 0.02931\ \text{therm/s}$$

Noting that 1 year = 365×24=8760 h and unit cost of natural gas is $1.10/therm, the annual fuel and money savings will be

Fuel Saved = (0.02931 therms/s)(8760×3600 s) = **924,400 therms/yr**

Money saved = (Fuel saved)(Unit cost of fuel)
= (924,400 therm/yr)($1.10/therm) = **$1,016,800/yr**

5-170E A refrigeration system is to cool eggs by chilled air at a rate of 10,000 eggs per hour. The rate of heat removal from the eggs, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

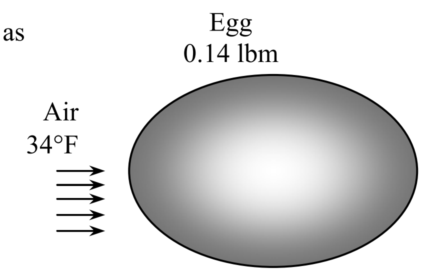
Assumptions **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated. **4** The properties of eggs are constant. **5** The local atmospheric pressure is 1 atm.

Properties The properties of the eggs are given to ρ = 67.4 lbm/ft^3 and c_p = 0.80 Btu/lbm.°F. The specific heat of air at room temperature c_p = 0.24 Btu/lbm. °F (Table A-2E). The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1E).

Analysis (*a*) Noting that eggs are cooled at a rate of 10,000 eggs per hour, eggs can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{egg}} = (10{,}000 \text{ eggs/h})(0.14 \text{ lbm/egg}) = 1400 \text{ lbm/h} = 0.3889 \text{ lbm/s}$$

Taking the egg flow stream in the cooler as the system, the energy balance for steadily flowing eggs can be expressed in the rate form as



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

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

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

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{egg}} = \dot{m}_{\text{egg}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the eggs as they are cooled from 90°F to 50ºF at this rate becomes

$$\dot{Q}_{\text{egg}} = (\dot{m} c_p \Delta T)_{\text{egg}} = (1400 \text{ lbm/h})(0.80 \text{ Btu/lbm.°F})(90 - 50)\text{°F} = \mathbf{44{,}800 \text{ Btu/h}}$$

(*b*) All the heat released by the eggs is absorbed by the refrigerated air since heat transfer through he walls of cooler is negligible, and the temperature rise of air is not to exceed 10°F. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p \Delta T)_{\text{air}}} = \frac{44{,}800 \text{ Btu/h}}{(0.24 \text{ Btu/lbm.°F})(10\text{°F})} = 18{,}667 \text{ lbm/h}$$

$$\rho_{\text{air}} = \frac{P}{RT} = \frac{14.7 \text{ psia}}{(0.3704 \text{ psia.ft}^3\text{/lbm.R})(34 + 460)\text{R}} = 0.0803 \text{ lbm/ft}^3$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{18{,}667 \text{ lbm/h}}{0.0803 \text{ lbm/ft}^3} = \mathbf{232{,}500 \text{ ft}^3\text{/h}}$$

5-171 Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined.

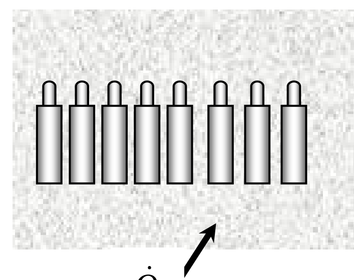
Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 55°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

Analysis (*a*) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

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

Then the rate of heat removal by the bottles as they are heated from 20 to 55°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} c_p \Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg.}^\circ\text{C})(55 - 20)^\circ\text{C} = 56{,}000 \text{ W}$$

The amount of water removed by the bottles is

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$

$$= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = \mathbf{2.67 \times 10^{-3} \text{ kg/s}}$$

Noting that the water removed by the bottles is made up by fresh water entering

at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(55 - 15)^\circ\text{C} = 446 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 56{,}000 + 446 = \mathbf{56{,}446 \text{ W}}$$

Discussion In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

5-172 Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined.

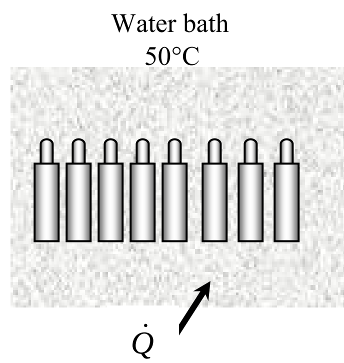
Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 50°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

Analysis (*a*) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

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

Then the rate of heat removal by the bottles as they are heated from 20 to 50°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} c_p \Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg.}^\circ\text{C})(50 - 20)^\circ\text{C} = 48{,}000 \text{ W}$$

The amount of water removed by the bottles is

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$
$$= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = \mathbf{2.67 \times 10^{-3} \text{ kg/s}}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(50 - 15)^\circ\text{C} = 391 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 48{,}000 + 391 = \mathbf{48{,}391 \text{ W}}$$

Discussion In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

5-173 Long aluminum wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

Properties The properties of aluminum are given to be ρ = 2702 kg/m³ and c_p = 0.896 kJ/kg.°C.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho\dot{V} = \rho(\pi r_0^2)V = (2702\ \text{kg/m}^3)\pi(0.0015\ \text{m})^2(10\ \text{m/min}) = 0.191\ \text{kg/min}$$

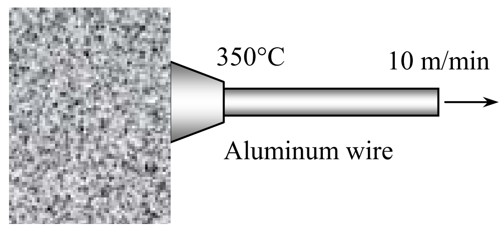
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.191\ \text{kg/min})(0.896\ \text{kJ/kg.°C})(350 - 50)\text{°C} = 51.3\ \text{kJ/min} = \mathbf{0.856\ kW}$$

5-174 Long copper wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

Properties The properties of copper are given to be ρ = 8950 kg/m³ and c_p = 0.383 kJ/kg.°C.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho\dot{V} = \rho(\pi r_0^2)V = (8950\ \text{kg/m}^3)\pi(0.0015\ \text{m})^2(10\ \text{m/min}) = 0.633\ \text{kg/min}$$

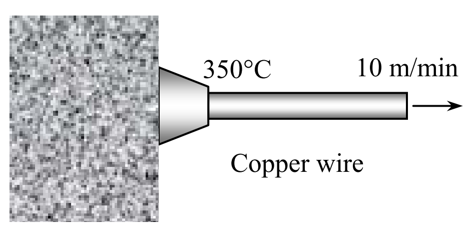
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.633\ \text{kg/min})(0.383\ \text{kJ/kg.°C})(350 - 50)\text{°C} = 72.7\ \text{kJ/min} = \mathbf{1.21\ kW}$$

5-175 Steam at a saturation temperature of T_{sat} = 40°C condenses on the outside of a thin horizontal tube. Heat is transferred to the cooling water that enters the tube at 25°C and exits at 35°C. The rate of condensation of steam is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The changes in kinetic and potential energies are negligible.

Properties The properties of water at room temperature are ρ = 997 kg/m^3 and c_p = 4.18 kJ/kg.°C (Table A-3). The enthalpy of vaporization of water at 40°C is h_{fg} = 2406.0 kJ/kg (Table A-4).

Analysis The mass flow rate of water through the tube is

$$\dot{m}_{\text{water}} = \rho V A_c = (997\ \text{kg/m}^3)(2\ \text{m/s})[\pi(0.03\ \text{m})^2 / 4] = 1.409\ \text{kg/s}$$

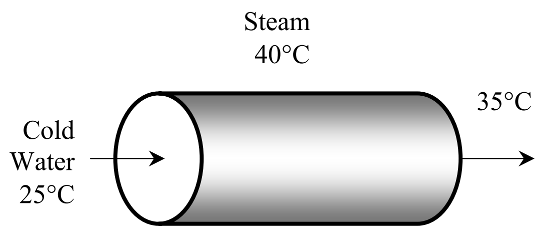
Taking the volume occupied by the cold water in the tube as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the rate of heat transfer to the water and the rate of condensation become

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}}) = (1.409\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})(35 - 25)°\text{C} = 58.9\ \text{kW}$$

$$\dot{Q} = \dot{m}_{\text{cond}} h_{fg} \ \rightarrow\ \dot{m}_{\text{cond}} = \frac{\dot{Q}}{h_{fg}} = \frac{58.9\ \text{kJ/s}}{2406.0\ \text{kJ/kg}} = \mathbf{0.0245\ kg/s}$$

5-176E Steam is mixed with water steadily in an adiabatic device. The temperature of the water leaving this device is to be determined.

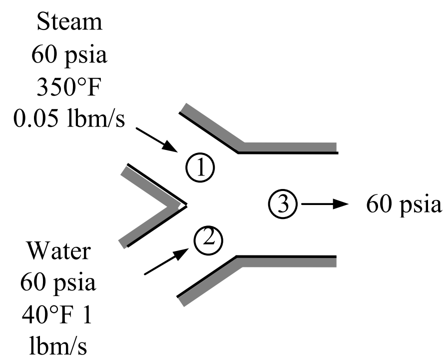
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions. **4** There is no heat transfer between the mixing device and the surroundings.

Analysis We take the mixing device as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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



From a mass balance

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 0.05 + 1 = 1.05 \text{ lbm/s}$$

The enthalpies of steam and water are (Table A-6E and A-4E)

$$\left. \begin{array}{l} P_1 = 60 \text{ psia} \\ T_1 = 350°\text{F} \end{array} \right\} h_1 = 1208.3 \text{ Btu/lbm}$$

$$\left. \begin{array}{l} P_2 = 60 \text{ psia} \\ T_2 = 40°\text{F} \end{array} \right\} h_2 \cong h_{f\,@\,40°\text{F}} = 8.032 \text{ Btu/lbm}$$

Substituting into the energy balance equation solving for the exit enthalpy gives

$$h_3 = \frac{\dot{m}_1 h_1 + \dot{m}_2 h_2}{\dot{m}_3} = \frac{(0.05 \text{ lbm/s})(1208.3 \text{ Btu/lbm}) + (1 \text{ lbm/s})(8.032 \text{ Btu/lbm})}{1.05 \text{ lbm/s}} = 65.19 \text{ Btu/lbm}$$

At the exit state P_3=60 psia and h_3=65.19 kJ/kg. An investigation of Table A-5E reveals that this is compressed liquid state. Approximating this state as saturated liquid at the specified temperature, the temperature may be determined from Table A-4E to be

$$\left. \begin{array}{l} P_3 = 60 \text{ psia} \\ h_3 = 65.19 \text{ Btu/lbm} \end{array} \right\} T_3 \cong T_{f\,@\,h=65.19 \text{ Btu/lbm}} = \mathbf{97.2°F}$$

Discussion The exact answer is determined at the compressed liquid state using EES to be **97.0°F**, indicating that the saturated liquid approximation is a reasonable one.

5-177 A constant-pressure R-134a vapor separation unit separates the liquid and vapor portions of a saturated mixture into two separate outlet streams. The flow power needed to operate this unit and the mass flow rate of the two outlet streams are to be determined.

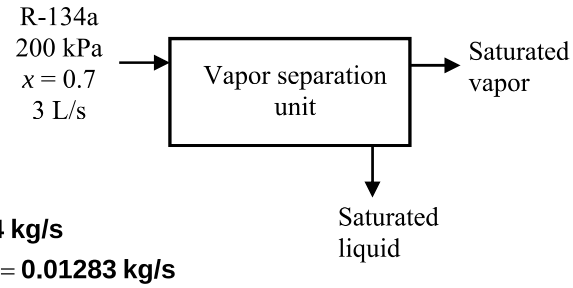
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work and heat interactions.

Analysis The specific volume at the inlet is (Table A-12)

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ x_1 = 0.70 \end{array}\right\} \boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.0007533 + (0.70)(0.099867 - 0.0007533) = 0.070133\text{ m}^3/\text{kg}$$

The mass flow rate at the inlet is then

$$\dot{m}_1 = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.003\text{ m}^3/\text{s}}{0.070133\text{ m}^3/\text{kg}} = 0.042776\text{ kg/s}$$



For each kg of mixture processed, 0.7 kg of vapor are processed. Therefore,

$$\dot{m}_2 = 0.7\dot{m}_1 = 0.7 \times 0.042776 = \mathbf{0.02994\ kg/s}$$

$$\dot{m}_3 = \dot{m}_1 - \dot{m}_2 = 0.3\dot{m}_1 = 0.3 \times 0.042776 = \mathbf{0.01283\ kg/s}$$

The flow power for this unit is

$$\begin{aligned} \dot{W}_{\text{flow}} &= \dot{m}_2 P_2 \boldsymbol{v}_2 + \dot{m}_3 P_3 \boldsymbol{v}_3 - \dot{m}_1 P_1 \boldsymbol{v}_1 \\ &= (0.02994\text{ kg/s})(200\text{ kPa})(0.099867\text{ m}^3/\text{kg}) + (0.01283\text{ kg/s})(200\text{ kPa})(0.0007533\text{ m}^3/\text{kg}) \\ &\quad - (0.042776\text{ kg/s})(200\text{ kPa})(0.070133\text{ m}^3/\text{kg}) \\ &= \mathbf{0\ kW} \end{aligned}$$

5-178E A small positioning control rocket in a satellite is driven by a container filled with R-134a at saturated liquid state. The number of bursts this rocket experience before the quality in the container is 90% or more is to be determined.

Analysis The initial and final specific volumes are

$$\left.\begin{array}{l} T = -10°\text{F} \\ x_1 = 0 \end{array}\right\} \boldsymbol{v}_1 = 0.01171\ \text{ft}^3/\text{lbm} \quad \text{(Table A-11E)}$$

$$\left.\begin{array}{l} T_2 = -10°\text{F} \\ x_2 = 0.90 \end{array}\right\} \boldsymbol{v}_2 = \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg} = 0.01171 + (0.90)(2.7091 - 0.01171) = 2.4394\ \text{ft}^3/\text{lbm}$$

The initial and final masses in the container are

$$m_1 = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{2\ \text{ft}^3}{0.01171\ \text{m}^3/\text{kg}} = 170.8\ \text{lbm}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{2\ \text{ft}^3}{2.4394\ \text{ft}^3/\text{lbm}} = 0.8199\ \text{lbm}$$

Then,

$$\Delta m = m_1 - m_2 = 170.8 - 0.8199 = 170.0\ \text{lbm}$$

The amount of mass released during each control burst is

$$\Delta m_b = \dot{m}\Delta t = (0.05\ \text{lbm/s})(5\ \text{s}) = 0.25\ \text{lbm}$$

The number of bursts that can be executed is then

$$N_b = \frac{\Delta m}{\Delta m_b} = \frac{170.0\ \text{lbm}}{0.25\ \text{lbm/burst}} = \mathbf{680\ bursts}$$

5-179E The relationships between the mass flow rate and the time for the inflation and deflation of an air bag are given. The volume of this bag as a function of time are to be plotted.

Assumptions Uniform flow exists at the inlet and outlet.

Properties The specific volume of air during inflation and deflation are given to be 15 and 13 ft^3/lbm, respectively.

Analysis The volume of the airbag at any time is given by

$$V(t) = \int_{\text{in flow time}} (\dot{m}v)_{\text{in}}\, dt - \int_{\text{out flow time}} (\dot{m}v)_{\text{out}}\, d$$

Applying at different time periods as given in problem statement give

$$V(t) = \int_0^t (15\ \text{ft}^3/\text{lbm}) \frac{20\ \text{lbm/s}}{10\ \text{ms}} \left(\frac{1\ \text{s}}{1000\ \text{ms}}\right) t dt \qquad 0 \le t \le 10\ \text{ms}$$

$$V(t) = \int_0^t 0.015 t^2\ \text{ft}^3/\text{ms}^2 \qquad 0 \le t \le 10\ \text{ms}$$

$$V(t) = V(10\ \text{ms}) + \int_{10\ \text{ms}}^t (15\ \text{ft}^3/\text{lbm})(20\ \text{lbm/s}) \left(\frac{1\ \text{s}}{1000\ \text{ms}}\right) t dt \qquad 10 < t \le 12\ \text{ms}$$

$$V(t) = V(10\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2 (t - 10\ \text{ms}) \qquad 10 < t \le 12\ \text{ms}$$

$$V(t) = V(12\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2 (t - 12\ \text{ms}) - \int_{12\ \text{ms}}^t (13\ \text{ft}^3/\text{lbm}) \frac{16\ \text{lbm/s}}{(30\text{-}12)\ \text{ms}} \left(\frac{1\ \text{s}}{1000\ \text{ms}}\right)(t - 12\ \text{ms}) dt \qquad 12 < t \le 25\ \text{ms}$$

$$V(t) = V(12\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2 (t - 12\ \text{ms}) - \int_{12\ \text{ms}}^t 0.011556\ \text{ft}^3/\text{ms}^2 (t - 12\ \text{ms}) dt \qquad 12 < t \le 25\ \text{ms}$$

$$V(t) = V(12\ \text{ms}) + 0.03\ \text{ft}^3/\text{ms}^2 (t - 12\ \text{ms}) - \frac{0.011556}{2}(t^2 - 144\ \text{ms}^2) + 0.13867(t - 12\ \text{ms}) \qquad 12 < t \le 25\ \text{ms}$$

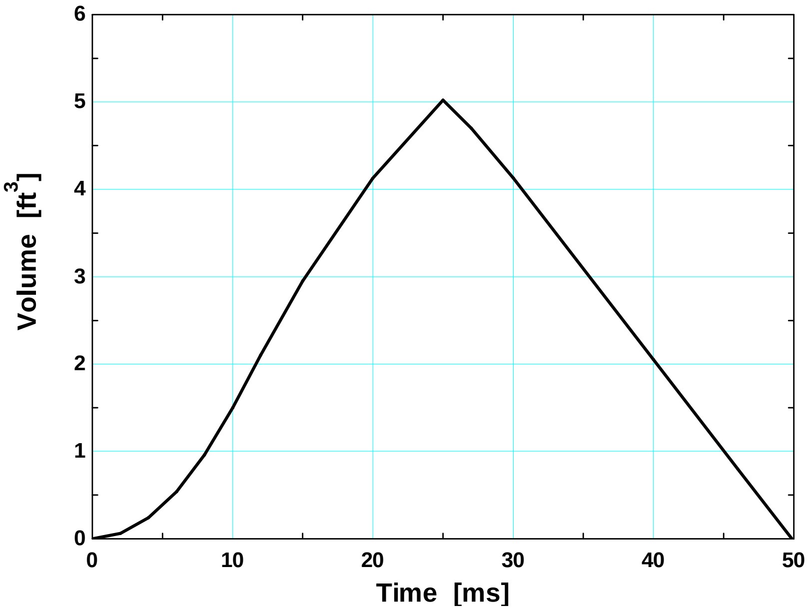
$$V(t) = V(25\ \text{ms}) - \frac{0.011556}{2}(t^2 - 625\ \text{ms}^2) + 0.13867(t - 25\ \text{ms}) \qquad 25 < t \le 30\ \text{ms}$$

$$V(t) = V(30\ \text{ms}) - \int_{12\ \text{ms}}^t (13\ \text{ft}^3/\text{lbm})(16\ \text{lbm/s}) \left(\frac{1\ \text{s}}{1000\ \text{ms}}\right) dt \qquad 30 < t \le 50\ \text{ms}$$

$$V(t) = V(30\ \text{ms}) - (0.208\ \text{ft}^3/\text{ms})(t - 30\ \text{ms}) \qquad 30 < t \le 50\ \text{ms}$$

The results with some suitable time intervals are

Time, ms	V, ft^3
0	0
2	0.06
4	0.24
6	0.54
8	0.96
10	1.50
12	2.10
15	2.95
20	4.13
25	5.02
27	4.70
30	4.13
40	2.05
46	0.80
49.85	0



Alternative solution

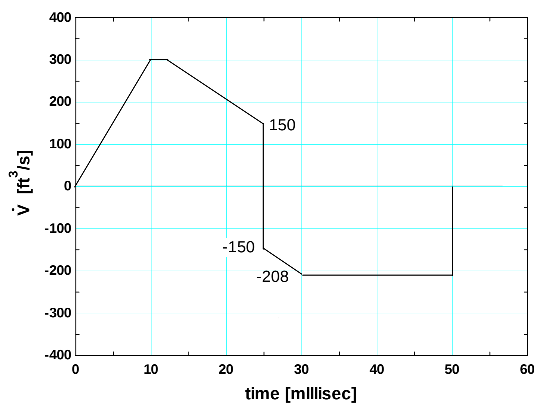
The net volume flow rate is obtained from

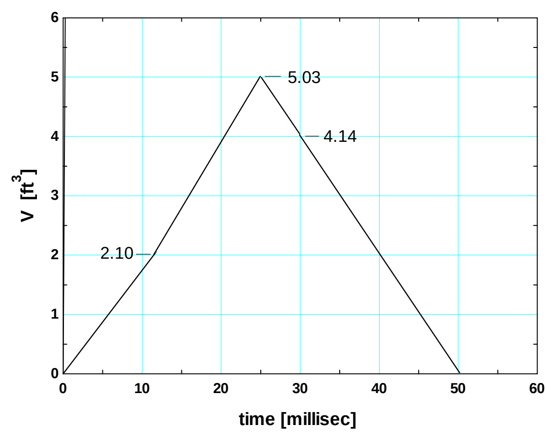
$$\dot{V} = (\dot{m}v)_{in} - (\dot{m}v)_{out}$$

which is sketched on the figure below. The volume of the airbag is given by

$$V = \int \dot{V} dt$$

The results of a graphical interpretation of the volume is also given in the figure below. Note that the evaluation of the above integral is simply the area under the process curve.





5-180 A rigid container filled with an ideal gas is heated while gas is released so that the temperature of the gas remaining in the container stays constant. An expression for the mass flow rate at the outlet as a function of the rate of pressure change in the container is to be derived.

Analysis At any instant, the mass in the control volume may be expressed as

$$m_{\text{CV}} = \frac{V}{v} = \frac{V}{RT} P$$

Since there are no inlets to this control volume, the conservation of mass principle becomes

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \frac{dm_{\text{CV}}}{dt}$$

$$\dot{m}_{\text{out}} = -\frac{dm_{\text{CV}}}{dt} = -\frac{V}{RT}\frac{dP}{dt}$$

5-181E A winterizing project is to reduce the infiltration rate of a house from 2.2 ACH to 1.1 ACH. The resulting cost savings are to be determined.

Assumptions **1** The house is maintained at 72°F at all times. **2** The latent heat load during the heating season is negligible. **3** The infiltrating air is heated to 72°F before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm·R (Table A-1E). The specific heat of air at room temperature is 0.24 Btu/lbm·°F (Table A-2E).

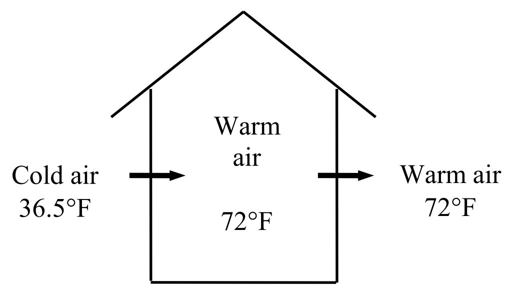
Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5 \text{ psia}}{(0.3704 \text{ psia.ft}^3/\text{lbm.R})(496.5 \text{ R})} = 0.0734 \text{ lbm/ft}^3$$

The volume of the house is

$$V_{\text{building}} = (\text{Floor area})(\text{Height}) = (3000 \text{ ft}^2)(9 \text{ ft}) = 27{,}000 \text{ ft}^3$$

We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the house. The energy balance for this imaginary steady-flow system can be expressed in the rate form as



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

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

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

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

The reduction in the infiltration rate is 2.2 – 1.1 = 1.1 ACH. The reduction in the sensible infiltration heat load corresponding to it is

$$\begin{aligned}\dot{Q}_{\text{infiltration, saved}} &= \rho_o c_p (ACH_{\text{saved}})(V_{\text{building}})(T_i - T_o)\\ &= (0.0734 \text{ lbm/ft}^3)(0.24 \text{ Btu/lbm.°F})(1.1/\text{h})(27{,}000 \text{ ft}^3)(72 - 36.5)\text{°F}\\ &= 18{,}573 \text{ Btu/h} = 0.18573 \text{ therm/h}\end{aligned}$$

since 1 therm = 100,000 Btu. The number of hours during a six month period is 6×30×24 = 4320 h. Noting that the furnace efficiency is 0.65 and the unit cost of natural gas is $1.24/therm, the energy and money saved during the 6-month period are

$$\begin{aligned}\text{Energy savings} &= (\dot{Q}_{\text{infiltration, saved}})(\text{No. of hours per year})/\text{Efficiency}\\ &= (0.18573 \text{ therm/h})(4320 \text{ h/year})/0.65\\ &= 1234 \text{ therms/year}\end{aligned}$$

$$\begin{aligned}\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy})\\ &= (1234 \text{ therms/year})(\$1.24/\text{therm})\\ &= \mathbf{\$1530/year}\end{aligned}$$

Therefore, reducing the infiltration rate by one-half will reduce the heating costs of this homeowner by $1530 per year.

5-182 Outdoors air at -5°C and 90 kPa enters the building at a rate of 35 L/s while the indoors is maintained at 20°C. The rate of sensible heat loss from the building due to infiltration is to be determined.

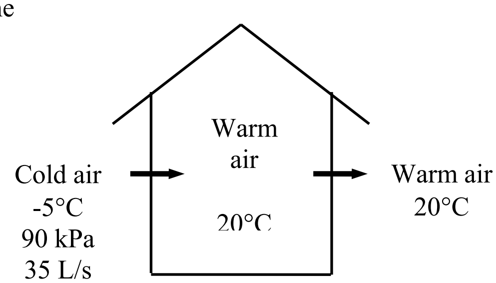
Assumptions **1** The house is maintained at 20°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 20°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg·K. The specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2).

Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{90 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(-5+273 \text{ K})} = 1.17 \text{ kg/m}^3$$

We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as



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

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

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

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

Then the sensible infiltration heat load corresponding to an infiltration rate of 35 L/s becomes

$$\begin{aligned} \dot{Q}_{infiltration} &= \rho_o \dot{V}_{air} c_p (T_i - T_o) \\ &= (1.17 \text{ kg/m}^3)(0.035 \text{ m}^3/\text{s})(1.005 \text{ kJ/kg.°C})[20 - (-5)]°\text{C} \\ &= \mathbf{1.029 \text{ kW}} \end{aligned}$$

Therefore, sensible heat will be lost at a rate of 1.029 kJ/s due to infiltration.

5-183 A fan is powered by a 0.5 hp motor, and delivers air at a rate of 85 m^3/min. The highest possible air velocity at the fan exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity and the change in potential energy are negligible, $V_1 \cong 0$ and $\Delta pe \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** The efficiencies of the motor and the fan are 100% since best possible operation is assumed. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be ρ = 1.18 kg/m^3. The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

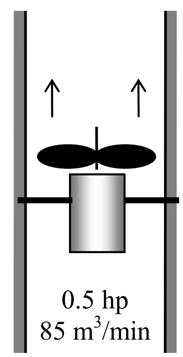
Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

The velocity of air leaving the fan will be highest when all of the entire electrical energy drawn by the motor is converted to kinetic energy, and the friction between the air layers is zero. In this best possible case, no energy will be converted to thermal energy, and thus the temperature change of air will be zero, $T_2 = T_1$. Then the energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } V_1 \cong 0 \text{ and } \Delta\text{pe} \cong 0)$$



Noting that the temperature and thus enthalpy remains constant, the relation above simplifies further to

$$\dot{W}_{e,in} = \dot{m}V_2^2/2$$

where

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1.18\ \text{kg/m}^3)(85\ \text{m}^3/\text{min}) = 100.3\ \text{kg/min} = 1.67\ \text{kg/s}$$

Solving for V_2 and substituting gives

$$V_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} = \sqrt{\frac{2(0.5\ \text{hp})}{1.67\ \text{kg/s}}\left(\frac{745.7\ \text{W}}{1\ \text{hp}}\right)\left(\frac{1\ \text{m}^2/\text{s}^2}{1\ \text{W}}\right)} = \mathbf{21.1\ m/s}$$

Discussion In reality, the velocity will be less because of the inefficiencies of the motor and the fan.

5-184 The average air velocity in the circular duct of an air-conditioning system is not to exceed 10 m/s. If the fan converts 70 percent of the electrical energy into kinetic energy, the size of the fan motor needed and the diameter of the main duct are to be determined.

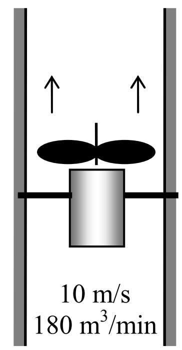
Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity is negligible, $V_1 \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be ρ = 1.20 kg/m^3. The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The change in the kinetic energy of air as it is accelerated from zero to 10 m/s at a rate of 180 m^3/s is

$$\dot{m} = \rho \dot{V} = (1.20\ \text{kg/m}^3)(180\ \text{m}^3/\text{min}) = 216\ \text{kg/min} = 3.6\ \text{kg/s}$$

$$\Delta \dot{\text{KE}} = \dot{m}\frac{V_2^2 - V_1^2}{2} = (3.6\ \text{kg/s})\frac{(10\ \text{m/s})^2 - 0}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 0.18\ \text{kW}$$



It is stated that this represents 70% of the electrical energy consumed by the motor. Then the total electrical power consumed by the motor is determined to be

$$0.7\dot{W}_{\text{motor}} = \Delta \dot{KE} \quad \rightarrow \quad \dot{W}_{\text{motor}} = \frac{\Delta \dot{KE}}{0.7} = \frac{0.18\ \text{kW}}{0.7} = \mathbf{0.257\ kW}$$

The diameter of the main duct is

$$\dot{V} = VA = V(\pi D^2/4) \rightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(180\ \text{m}^3/\text{min})}{\pi(10\ \text{m/s})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right)} = \mathbf{0.618\ m}$$

Therefore, the motor should have a rated power of at least 0.257 kW, and the diameter of the duct should be at least 61.8 cm

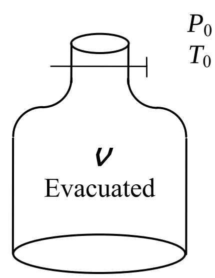
5-185 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

Analysis We take the bottle as the system. It is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$$



Energy balance:

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

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = \text{ke} \cong \text{pe} \cong 0)$$

Combining the two balances:

$$Q_{\text{in}} = m_2(u_2 - h_i) = m_2(c_v T_2 - c_p T_i)$$

but

$$T_i = T_2 = T_0$$

and

$$c_p - c_v = R.$$

Substituting,

$$Q_{\text{in}} = m_2(c_v - c_p)T_0 = -m_2 R T_0 = -\frac{P_0 V}{R T_0} R T_0 = -P_0 V$$

Therefore,

$$Q_{\text{out}} = \boldsymbol{P_0 V} \quad \text{(Heat is lost from the tank)}$$

5-186 An adiabatic air compressor is powered by a direct-coupled steam turbine, which is also driving a generator. The net power delivered to the generator is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The devices are adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

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

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} h_3 = 3343.6 \text{ kJ/kg}$$

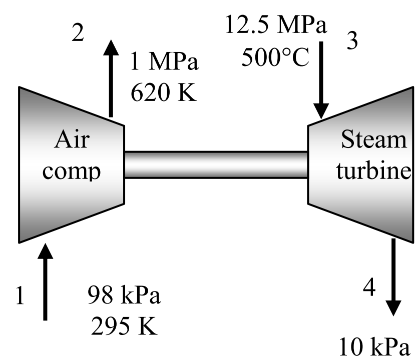
and

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg}$$

From the air table (Table A-17),

$$T_1 = 295 \text{ K} \longrightarrow h_1 = 295.17 \text{ kJ/kg}$$

$$T_2 = 620 \text{ K} \longrightarrow h_2 = 628.07 \text{ kJ/kg}$$



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

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

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

For the turbine and the compressor it becomes

Compressor:

$$\dot{W}_{\text{comp, in}} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2 \quad \to \quad \dot{W}_{\text{comp, in}} = \dot{m}_{air}(h_2 - h_1)$$

Turbine:

$$\dot{m}_{steam} h_3 = \dot{W}_{\text{turb, out}} + \dot{m}_{steam} h_4 \quad \to \quad \dot{W}_{\text{turb, out}} = \dot{m}_{steam}(h_3 - h_4)$$

Substituting,

$$\dot{W}_{comp,in} = (10 \text{ kg/s})(628.07 - 295.17) \text{ kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{turb,out} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{ kJ/kg} = 23,777 \text{ kW}$$

Therefore,

$$\dot{W}_{net,out} = \dot{W}_{turb,out} - \dot{W}_{comp,in} = 23,777 - 3329 = \mathbf{20,448 \text{ kW}}$$

5-187 Helium is compressed by a compressor. The power required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of helium is c_p = 5.1926 kJ/kg·K (Table A-2b). The gas constant of air is R = 2.0769 kPa·m^3/kg·K (Table A-1).

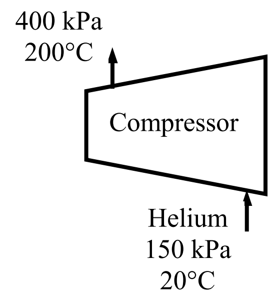
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



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

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(2.0769 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(20 + 273 \text{ K})}{150 \text{ kPa}} = 4.0569 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\boldsymbol{v}_1} = \frac{(0.1 \text{ m}^2)(15 \text{ m/s})}{4.0569 \text{ m}^3/\text{kg}} = 0.3697 \text{ kg/s}$$

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

$$\dot{W}_{in} = \dot{m}c_p(T_2 - T_1) = (0.3697 \text{ kg/s})(5.1926 \text{ kJ/kg} \cdot \text{K})(200 - 20)\text{K} = \mathbf{345.5 \text{ kW}}$$

5-188 A submarine that has an air-ballast tank originally partially filled with air is considered. Air is pumped into the ballast tank until it is entirely filled with air. The final temperature and mass of the air in the ballast tank are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** The process is adiabatic. **3** There are no work interactions.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1). The specific heat ratio of air at room temperature is $k = 1.4$ (Table A-2*a*). The specific volume of water is taken 0.001 m^3/kg.

Analysis The conservation of mass principle applied to the air gives

$$\frac{dm_a}{dt} = \dot{m}_{\text{in}}$$

and as applied to the water becomes

$$\frac{dm_w}{dt} = -\dot{m}_{\text{out}}$$

The first law for the ballast tank produces

$$0 = \frac{d(mu)_a}{dt} + \frac{d(mu)_w}{dt} + h_w \dot{m}_w - h_a \dot{m}_a$$

Combining this with the conservation of mass expressions, rearranging and canceling the common *dt* term produces

$$d(mu)_a + d(mu)_w = h_a dm_a + h_w dm_w$$

Integrating this result from the beginning to the end of the process gives

$$\left[(mu)_2 - (mu)_1\right]_a + \left[(mu)_2 - (mu)_1\right]_w = h_a (m_2 - m_1)_a + h_w (m_2 - m_1)_w$$

Substituting the ideal gas equation of state and the specific heat models for the enthalpies and internal energies expands this to

$$\frac{P\boldsymbol{V}_2}{RT_2} c_v T_2 - \frac{P\boldsymbol{V}_1}{RT_1} c_v T_1 - m_{w,1} c_w T_w = c_p T_{\text{in}} \left(\frac{P\boldsymbol{V}_2}{RT_2} - \frac{P\boldsymbol{V}_1}{RT_1} \right) - m_{w,1} c_w T_w$$

When the common terms are cancelled, this result becomes

$$T_2 = \frac{\boldsymbol{V}_2}{\dfrac{\boldsymbol{V}_1}{T_1} + \dfrac{1}{kT_{\text{in}}}(\boldsymbol{V}_2 - \boldsymbol{V}_1)} = \frac{1000}{\dfrac{100}{288} + \dfrac{1}{(1.4)(293)}(900)} = \mathbf{393.5\ K}$$

The final mass from the ideal gas relation is

$$m_2 = \frac{P\boldsymbol{V}_2}{RT_2} = \frac{(2000\ \text{kPa})(1000\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(393.5\ \text{K})} = \mathbf{17{,}710\ kg}$$

5-189 A submarine that has an air-ballast tank originally partially filled with air is considered. Air is pumped into the ballast tank in an isothermal manner until it is entirely filled with air. The final mass of the air in the ballast tank and the total heat transfer are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** There are no work interactions.

Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 kJ/kg·K and c_v = 0.718 kJ/kg·K (Table A-2*a*). The specific volume of water is taken 0.001 m^3/kg.

Analysis The initial air mass is

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(2000\ \text{kPa})(100\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})} = 2420\ \text{kg}$$

and the initial water mass is

$$m_{w,1} = \frac{V_1}{v_1} = \frac{900\ \text{m}^3}{0.001\ \text{m}^3/\text{kg}} = 900{,}000\ \text{kg}$$

and the final mass of air in the tank is

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(2000\ \text{kPa})(1000\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})} = \mathbf{24{,}200\ kg}$$

The first law when adapted to this system gives

$$Q_{\text{in}} + m_i h_i - m_e h_e = m_2 u_2 - m_1 u_1$$
$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + m_e h_e - m_i h_i$$
$$Q_{\text{in}} = m_2 c_v T - (m_{a,1} c_v T + m_w u_w) + m_w h_w - (m_2 - m_1) c_p T$$

Noting that

$$u_w \cong h_w = 62.98\ \text{kJ/kg}$$

Substituting,

$$\begin{aligned} Q_{\text{in}} &= 24{,}200 \times 0.718 \times 288 - (2420 \times 0.718 \times 288 + 900{,}000 \times 62.98) \\ &\quad + 900{,}000 \times 62.98 - (24200 - 2420) \times 1.005 \times 288 \\ &= -\mathbf{1.800 \times 10^6\ kJ} \end{aligned}$$

The negative sign shows that the direction of heat is from the tank.

5-190 A cylindrical tank is charged with nitrogen from a supply line. The final mass of nitrogen in the tank and final temperature are to be determined for two cases.

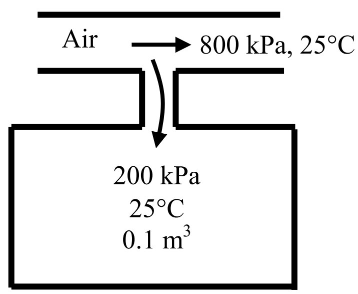
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved.

Properties The gas constant of nitrogen is 0.2968 kPa·m³/kg·K (Table A-1). The specific heats of nitrogen at room temperature are c_p = 1.039 kJ/kg·K and c_v = 0.743 kJ/kg·K (Table A-2*a*).

Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

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



Energy balance:

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

$$m_i h_i = m_2 u_2 - m_1 u_1$$

$$m_i c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

Combining the two balances:

$$(m_2 - m_1) c_p T_i = m_2 c_v T_2 - m_1 c_v T_1$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(200\ \text{kPa})(0.1\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\ \text{K})} = 0.2261\ \text{kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(800\ \text{kPa})(0.1\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{278.7}{T_2}$$

Substituting,

$$\left(\frac{278.7}{T_2} - 0.2261\right)(1.039)(298) = \frac{278.7}{T_2}(0.743)T_2 - (0.2261)(0.743)(298)$$

whose solution is

$$T_2 = \mathbf{380.1\ K} = \mathbf{107.1°C}$$

The final mass is then

$$m_2 = \frac{278.7}{T_2} = \frac{278.7}{380.1} = \mathbf{0.7332\ kg}$$

(*b*) When there is rapid heat transfer between the nitrogen and tank such that the cylinder and nitrogen remain in thermal equilibrium during the process, the energy balance equation may be written as

$$(m_2 - m_1) c_p T_i = (m_{nit,2} c_v T_2 + m_t c_t T_2) - (m_{nit,1} c_v T_1 + m_t c_t T_1)$$

Substituting,

$$\left(\frac{278.7}{T_2} - 0.2261\right)(1.039)(298) = \left[\frac{278.7}{T_2}(0.743)T_2 + (50)(0.43)T_2\right] - \left[(0.2261)(0.743)(298) + (50)(0.43)(298)\right]$$

whose solution is

$$T_2 = \mathbf{300.8\ K} = \mathbf{27.8°C}$$

The final mass is then

$$m_2 = \frac{278.7}{300.8} = \frac{278.7}{300.8} = \mathbf{0.9266\ kg}$$

5-191 The air in a tank is released until the pressure in the tank reduces to a specified value. The mass withdrawn from the tank is to be determined for three methods of analysis.

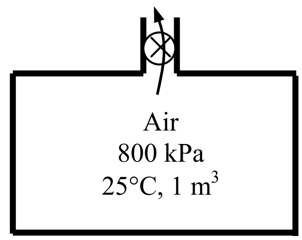
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work or heat interactions involved.

Properties The gas constant of air is 0.287 kPa·m^3/kg·K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 kJ/kg·K and c_v = 0.718 kJ/kg·K. Also k =1.4 (Table A-2*a*).

Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$



Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$
$$-m_e h_e = m_2 u_2 - m_1 u_1$$
$$0 = m_2 u_2 - m_1 u_1 + m_e h_e$$
$$0 = m_2 c_v T_2 - m_1 c_v T_1 + m_e c_p T_e$$

Combining the two balances:

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$

The initial and final masses are given by

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(800\text{ kPa})(1\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(25+273\text{ K})} = 9.354\text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(150\text{ kPa})(1\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{522.6}{T_2}$$

The temperature of air leaving the tank changes from the initial temperature in the tank to the final temperature during the discharging process. We assume that the temperature of the air leaving the tank is the average of initial and final temperatures in the tank. Substituting into the energy balance equation gives

$$0 = m_2 c_v T_2 - m_1 c_v T_1 + (m_1 - m_2) c_p T_e$$
$$0 = \frac{522.6}{T_2}(0.718)T_2 - (9.354)(0.718)(298) + \left(9.354 - \frac{522.6}{T_2}\right)(1.005)\left(\frac{298+T_2}{2}\right)$$

whose solution is

$$T_2 = 191.0\text{ K}$$

Substituting, the final mass is

$$m_2 = \frac{522.6}{191} = 2.736\text{ kg}$$

and the mass withdrawn is

$$m_e = m_1 - m_2 = 9.354 - 2.736 = \mathbf{6.618\ kg}$$

(*b*) Considering the process in two parts, first from 800 kPa to 400 kPa and from 400 kPa to 150 kPa, the solution will be as follows:

From 800 kPa to 400 kPa:

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(400\ \text{kPa})(1\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{1394}{T_2}$$

$$0 = \frac{1394}{T_2}(0.718)T_2 - (9.354)(0.718)(298) + \left(9.354 - \frac{1394}{T_2}\right)(1.005)\left(\frac{298 + T_2}{2}\right)$$

$$T_2 = 245.1\ \text{K}$$

$$m_2 = \frac{1394}{245.1} = 5.687\ \text{kg}$$

$$m_{e,1} = m_1 - m_2 = 9.354 - 5.687 = 3.667\ \text{kg}$$

From 400 kPa to 150 kPa:

$$0 = \frac{522.6}{T_2}(0.718)T_2 - (5.687)(0.718)(298) + \left(5.687 - \frac{522.6}{T_2}\right)(1.005)\left(\frac{298 + T_2}{2}\right)$$

$$T_2 = 215.5\ \text{K}$$

$$m_2 = \frac{522.6}{215.5} = 2.425\ \text{kg}$$

$$m_{e,2} = m_1 - m_2 = 5.687 - 2.425 = 3.262\ \text{kg}$$

The total mass withdrawn is

$$m_e = m_{e,1} + m_{e,2} = 3.667 + 3.262 = \mathbf{6.929\ kg}$$

(*c*) The mass balance may be written as

$$\frac{dm}{dt} = -\dot{m}_e$$

When this is combined with the ideal gas equation of state, it becomes

$$\frac{V}{R}\frac{d(P/T)}{dt} = -\dot{m}_e$$

since the tank volume remains constant during the process. An energy balance on the tank gives

$$\frac{d(mu)}{dt} = -h_e \dot{m}_e$$

$$c_v \frac{d(mT)}{dt} = c_p T \frac{dm}{dt}$$

$$c_v \frac{V}{R} \frac{dP}{dt} = c_p T \frac{V}{R} \frac{d(P/T)}{dt}$$

$$c_v \frac{dP}{dt} = c_p \left(\frac{dP}{dt} - \frac{P}{T} \frac{dT}{dt} \right)$$

$$(c_p - c_v) \frac{dP}{P} = c_p \frac{dT}{dt}$$

When this result is integrated, it gives

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (298\text{ K}) \left(\frac{150\text{ kPa}}{800\text{ kPa}} \right)^{0.4/1.4} = 184.7\text{ K}$$

The final mass is

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(150\text{ kPa})(1\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.7\text{ K})} = 2.830\text{ kg}$$

and the mass withdrawn is

$$m_e = m_1 - m_2 = 9.354 - 2.830 = \mathbf{6.524\text{ kg}}$$

Discussion The result in first method is in error by 1.4% while that in the second method is in error by 6.2%.

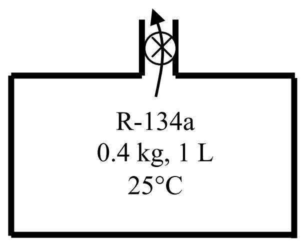
5-192 A tank initially contains saturated mixture of R-134a. A valve is opened and R-134a vapor only is allowed to escape slowly such that temperature remains constant. The heat transfer necessary with the surroundings to maintain the temperature and pressure of the R-134a constant is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the exit remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$



Energy balance:

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

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1$$
$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + m_e h_e$$

Combining the two balances:

$$Q_{\text{in}} = m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$

The specific volume at the initial state is

$$v_1 = \frac{V}{m_1} = \frac{0.001\,\text{m}^3}{0.4\,\text{kg}} = 0.0025\,\text{m}^3/\text{kg}$$

The initial state properties of R-134a in the tank are

$$\left.\begin{array}{l} T_1 = 26°\text{C} \\ v_1 = 0.0025\,\text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_1 = \dfrac{v_1 - v_f}{v_{fg}} = \dfrac{0.0025 - 0.0008313}{0.029976 - 0.0008313} = 0.05726 \\ u_1 = u_f + x_1 u_{fg} = 87.26 + (0.05726)(156.87) = 96.24\,\text{kJ/kg} \end{array} \quad \text{(Table A-11)}$$

The enthalpy of saturated vapor refrigerant leaving the bottle is

$$h_e = h_{g\,@\,26°\text{C}} = 264.68\,\text{kJ/kg}$$

The specific volume at the final state is

$$v_2 = \frac{V}{m_2} = \frac{0.001\,\text{m}^3}{0.1\,\text{kg}} = 0.01\,\text{m}^3/\text{kg}$$

The internal energy at the final state is

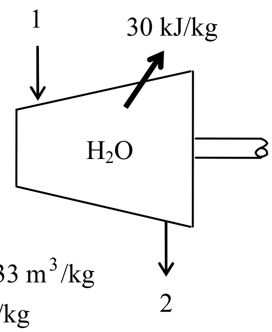
$$\left.\begin{array}{l} T_2 = 26°\text{C} \\ v_2 = 0.01\,\text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{v_2 - v_f}{v_{fg}} = \dfrac{0.01 - 0.0008313}{0.029976 - 0.0008313} = 0.3146 \\ u_2 = u_f + x_1 u_{fg} = 87.26 + (0.3146)(156.87) = 136.61\,\text{kJ/kg} \end{array} \quad \text{(Table A-11)}$$

Substituting into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e \\ &= (0.1\,\text{kg})(136.61\,\text{kJ/kg}) - (0.4\,\text{kg})(96.24\,\text{kJ/kg}) + (0.4 - 0.1\,\text{kg})(264.68\,\text{kJ/kg}) \\ &= \mathbf{54.6\,kJ} \end{aligned}$$

5-193 [*Also solved by EES on enclosed CD*] 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)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 550°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.035655 \text{ m}^3/\text{kg} \\ h_1 = 3502.0 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 25 \text{ kPa} \\ x_2 = 0.95 \end{array}\right\} \begin{array}{l} v_2 = v_f + x_2 v_{fg} = 0.00102 + (0.95)(6.2034 - 0.00102) = 5.8933 \text{ m}^3/\text{kg} \\ h_2 = h_f + x_2 h_{fg} = 271.96 + (0.95)(2345.5) = 2500.2 \text{ kJ/kg} \end{array}$$

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

$$\dot{m} = \frac{1}{v_1} V_1 A_1 = \frac{1}{0.035655 \text{ m}^3/\text{kg}} (60 \text{ m/s})(0.015 \text{ m}^2) = \mathbf{25.24 \text{ kg/s}}$$

(*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

$$\dot{m} = \frac{1}{v_2} V_2 A_2 \longrightarrow V_2 = \frac{\dot{m} v_2}{A_2} = \frac{(25.24 \text{ kg/s})(5.8933 \text{ m}^3/\text{kg})}{0.14 \text{ m}^2} = \mathbf{1063 \text{ m/s}}$$

(*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

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

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

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

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

Then the power output of the turbine is determined by substituting to be

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

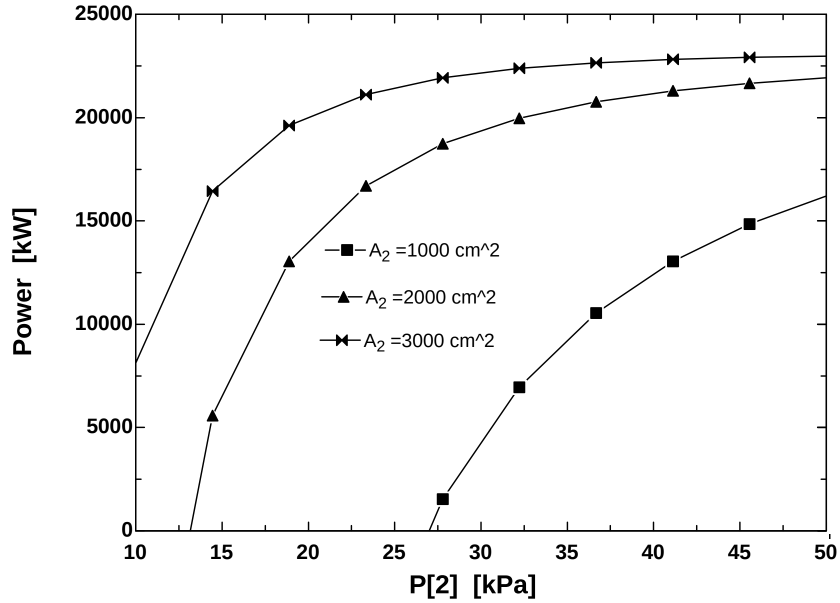
$$= \mathbf{10{,}330 \text{ kW}}$$

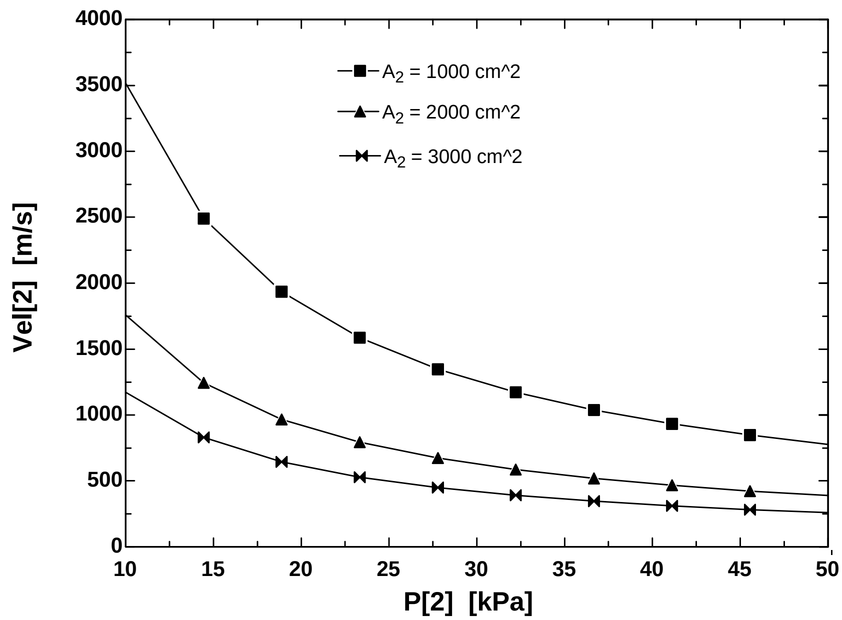
5-194 EES Problem 5-193 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 cm^2 to 3000 cm^2 is to be investigated. The exit velocity and the power output are to be plotted against the exit pressure for the exit areas of 1000, 2000, and 3000 cm2.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
Fluid$='Steam_IAPWS'

A[1]=150 [cm^2]
T[1]=550 [C]
P[1]=10000 [kPa]
Vel[1]= 60 [m/s]
A[2]=1400 [cm^2]
P[2]=25 [kPa]
q_out = 30 [kJ/kg]
m_dot = A[1]*Vel[1]/v[1]*convert(cm^2,m^2)
v[1]=volume(Fluid$, T=T[1], P=P[1]) "specific volume of steam at state 1"
Vel[2]=m_dot*v[2]/(A[2]*convert(cm^2,m^2))
v[2]=volume(Fluid$, x=0.95, P=P[2]) "specific volume of steam at state 2"
T[2]=temperature(Fluid$, P=P[2], v=v[2]) "[C]" "not required, but good to know"

"[conservation of Energy for steady-flow:"
"Ein_dot - Eout_dot = DeltaE_dot"  "For steady-flow, DeltaE_dot = 0"
DELTAE_dot=0 "[kW]"
"For the turbine as the control volume, neglecting the PE of each flow steam:"
E_dot_in=E_dot_out
h[1]=enthalpy(Fluid$,T=T[1], P=P[1])
E_dot_in=m_dot*(h[1]+ Vel[1]^2/2*Convert(m^2/s^2, kJ/kg))
h[2]=enthalpy(Fluid$,x=0.95, P=P[2])
E_dot_out=m_dot*(h[2]+ Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+ m_dot *q_out+ W_dot_out
Power=W_dot_out
Q_dot_out=m_dot*q_out
```

Power [kW]	P_2 [kPa]	Vel_2 [m/s]
-54208	10	2513
-14781	14.44	1778
750.2	18.89	1382
8428	23.33	1134
12770	27.78	962.6
15452	32.22	837.6
17217	36.67	742.1
18432	41.11	666.7
19299	45.56	605.6
19935	50	555


25000
20000
15000
10000
5000
0
10
15
20
25
30
35
40
45
50
Power [kW]
P[2] [kPa]
A2 =1000 cm^2
A2 =2000 cm^2
A2 =3000 cm^2

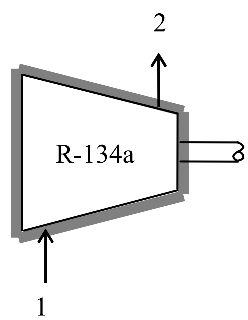

4000
3500
3000
2500
2000
1500
1000
500
0
10
15
20
25
30
35
40
45
50
Vel[2] [m/s]
P[2] [kPa]
A2 = 1000 cm^2
A2 = 2000 cm^2
A2 = 3000 cm^2

5-195E Refrigerant-134a is compressed steadily by a compressor. The mass flow rate of the refrigerant and the exit temperature are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the refrigerant tables (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 15 \text{ psia} \\ T_1 = 20°\text{F} \end{array}\right\} \begin{array}{l} v_1 = 3.2551 \text{ ft}^3/\text{lbm} \\ h_1 = 107.52 \text{ Btu/lbm} \end{array}$$



Analysis (*a*) The mass flow rate of refrigerant is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{10 \text{ ft}^3/\text{s}}{3.2551 \text{ ft}^3/\text{lbm}} = \textbf{3.072 lbm/s}$$

(*b*) 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,

$$(45 \text{ hp})\left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) = (3.072 \text{ lbm/s})(h_2 - 107.52)\text{Btu/lbm}$$

$$h_2 = 117.87 \text{ Btu/lbm}$$

Then the exit temperature becomes

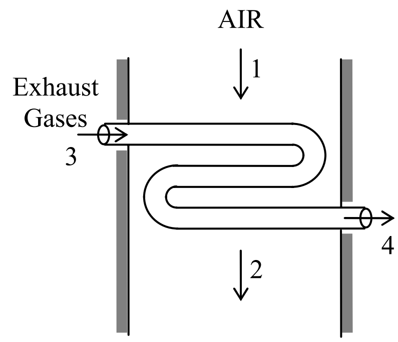
$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ h_2 = 117.87 \text{ Btu/lbm} \end{array}\right\} T_2 = \textbf{95.7°F}$$

5-196 Air is preheated by the exhaust gases of a gas turbine in a regenerator. For a specified heat transfer rate, the exit temperature of air and the mass flow rate of exhaust gases are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the regenerator to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Exhaust gases can be treated as air. **6** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpies of air are (Table A-17)

$$T_1 = 550\text{ K} \rightarrow h_1 = 555.74\text{ kJ/kg}$$
$$T_3 = 800\text{ K} \rightarrow h_3 = 821.95\text{ kJ/kg}$$
$$T_4 = 600\text{ K} \rightarrow h_4 = 607.02\text{ kJ/kg}$$



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

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

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

Substituting,

$$3200\text{ kJ/s} = (800/60\text{ kg/s})(h_2 - 554.71\text{ kJ/kg}) \rightarrow h_2 = 794.71\text{ kJ/kg}$$

Then from Table A-17 we read

$$T_2 = \mathbf{775.1\ K}$$

(*b*) Treating the exhaust gases as an ideal gas, the mass flow rate of the exhaust gases is determined from the steady-flow energy relation applied only to the exhaust gases,

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}_{exhaust} h_3 = \dot{Q}_{out} + \dot{m}_{exhaust} h_4 \quad (\text{since } \dot{W} = \Delta ke \cong \Delta pe \cong 0)$$
$$\dot{Q}_{out} = \dot{m}_{exhaust}(h_3 - h_4)$$

$$3200\text{ kJ/s} = \dot{m}_{exhaust}(821.95 - 607.02)\text{ kJ/kg}$$

It yields

$$\dot{m}_{exhaust} = \mathbf{14.9\ kg/s}$$

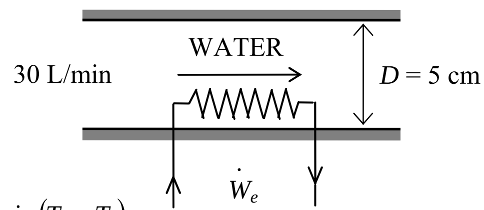
5-197 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** The pipe is insulated and thus the heat losses are negligible.

Properties The density and specific heat of water at room temperature are ρ = 1000 kg/m³ and c = 4.18 kJ/kg·°C (Table A-3).

Analysis (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. Also, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

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



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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

The mass flow rate of water through the pipe is

$$\dot{m} = \rho\dot{V}_1 = (1000 \text{ kg/m}^3)(0.030 \text{ m}^3/\text{min}) = 30 \text{ kg/min}$$

Therefore,

$$\dot{W}_{e,in} = \dot{m}c(T_2 - T_1) = (30/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(55-20)^\circ\text{C} = \mathbf{73.2\ kW}$$

(*b*) The average velocity of water through the pipe is determined from

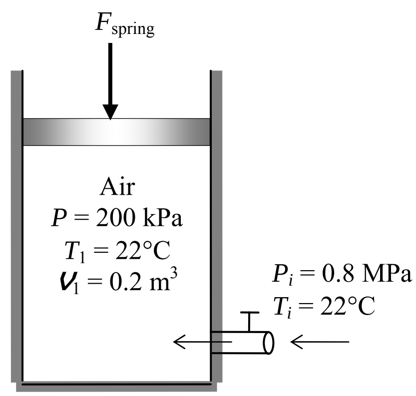
$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}}{\pi r^2} = \frac{0.030 \text{ m}^3/\text{min}}{\pi(0.025 \text{ m})^2} = \mathbf{15.3\ m/min}$$

5-198 [*Also solved by EES on enclosed CD***]** 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:

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

Energy balance:

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

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

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 $\quad T_2 = \mathbf{344.1 \text{ K}}$

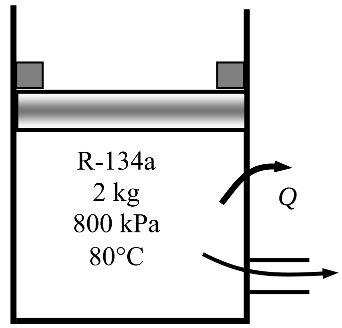
Thus, $\quad m_2 = \dfrac{836.2}{T_2} = \dfrac{836.2}{344.1} = 2.430 \text{ kg}$

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

5-199 R-134a is allowed to leave a piston-cylinder device with a pair of stops. The work done and the heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device is assumed to be constant. **2** Kinetic and potential energies are negligible.

Properties The properties of R-134a at various states are (Tables A-11 through A-13)



$$\left.\begin{array}{l} P_1 = 800\text{ kPa} \\ T_1 = 80°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.032659\text{ m}^3/\text{kg} \\ u_1 = 290.84\text{ kJ/kg} \\ h_1 = 316.97\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 500\text{ kPa} \\ T_2 = 20°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.042115\text{ m}^3/\text{kg} \\ u_2 = 242.40\text{ kJ/kg} \\ h_2 = 263.46\text{ kJ/kg} \end{array}$$

Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance:

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

$$W_{\text{b,in}} - Q_{\text{out}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad \text{(since ke} \cong \text{pe} \cong 0)$$

The volumes at the initial and final states and the mass that has left the cylinder are

$$V_1 = m_1 v_1 = (2\text{ kg})(0.032659\text{ m}^3/\text{kg}) = 0.06532\text{ m}^3$$

$$V_2 = m_2 v_2 = (1/2) m_1 v_2 = (1/2)(2\text{ kg})(0.042115\text{ m}^3/\text{kg}) = 0.04212\text{ m}^3$$

$$m_e = m_1 - m_2 = 2 - 1 = 1\text{ kg}$$

The enthalpy of the refrigerant withdrawn from the cylinder is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder

$$h_e = (1/2)(h_1 + h_2) = (1/2)(316.97 + 263.46) = 290.21\text{ kJ/kg}$$

Noting that the pressure remains constant after the piston starts moving, the boundary work is determined from

$$W_{\text{b,in}} = P_2(V_1 - V_2) = (500\text{ kPa})(0.06532 - 0.04212)\text{m}^3 = \mathbf{11.6\ kJ}$$

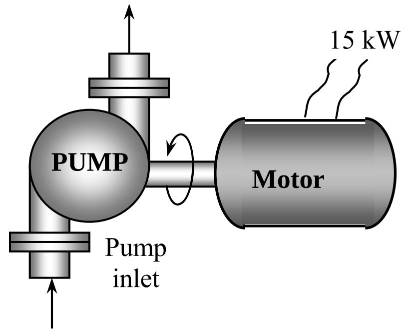
(*b*) Substituting,

$$11.6\text{ kJ} - Q_{\text{out}} - (1\text{ kg})(290.21\text{ kJ/kg}) = (1\text{ kg})(242.40\text{ kJ/kg}) - (2\text{ kg})(290.84\text{ kJ/kg})$$

$$Q_{\text{out}} = \mathbf{60.7\ kJ}$$

5-200 The pressures across a pump are measured. The mechanical efficiency of the pump and the temperature rise of water are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The pump is driven by an external motor so that the heat generated by the motor is dissipated to the atmosphere. **3** The elevation difference between the inlet and outlet of the pump is negligible, $z_1 = z_2$. **4** The inlet and outlet diameters are the same and thus the inlet and exit velocities are equal, $V_1 = V_2$.



Properties We take the density of water to be 1 kg/L = 1000 kg/m^3 and its specific heat to be 4.18 kJ/kg · °C (Table A-3).

Analysis (*a*) The mass flow rate of water through the pump is

$$\dot{m} = \rho\dot{V} = (1\text{ kg/L})(50\text{ L/s}) = 50\text{ kg/s}$$

The motor draws 15 kW of power and is 90 percent efficient. Thus the mechanical (shaft) power it delivers to the pump is

$$\dot{W}_{\text{pump,shaft}} = \eta_{\text{motor}}\dot{W}_{\text{electric}} = (0.90)(15\text{ kW}) = 13.5\text{ kW}$$

To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) - \dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right)$$

Simplifying it for this case and substituting the given values,

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}\left(\frac{P_2 - P_1}{\rho}\right) = (50\text{ kg/s})\left(\frac{(300-100)\text{kPa}}{1000\text{ kg/m}^3}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 10\text{ kW}$$

Then the mechanical efficiency of the pump becomes

$$\eta_{pump} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump,shaft}}} = \frac{10\text{ kW}}{13.5\text{ kW}} = 0.741 = \mathbf{74.1\%}$$

(*b*) Of the 13.5-kW mechanical power supplied by the pump, only 10 kW is imparted to the fluid as mechanical energy. The remaining 3.5 kW is converted to thermal energy due to frictional effects, and this "lost" mechanical energy manifests itself as a heating effect in the fluid,

$$\dot{E}_{\text{mech,loss}} = \dot{W}_{\text{pump,shaft}} - \Delta\dot{E}_{\text{mech,fluid}} = 13.5 - 10 = 3.5\text{ kW}$$

The temperature rise of water due to this mechanical inefficiency is determined from the thermal energy balance,

$$\dot{E}_{\text{mech,loss}} = \dot{m}(u_2 - u_1) = \dot{m}c\Delta T$$

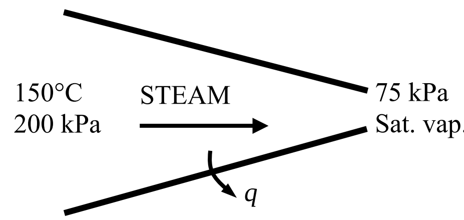
Solving for ΔT,

$$\Delta T = \frac{\dot{E}_{\text{mech,loss}}}{\dot{m}c} = \frac{3.5\text{ kW}}{(50\text{ kg/s})(4.18\text{ kJ/kg}\cdot\text{K})} = \mathbf{0.017°C}$$

Therefore, the water will experience a temperature rise of 0.017°C, which is very small, as it flows through the pump.

Discussion In an actual application, the temperature rise of water will probably be less since part of the heat generated will be transferred to the casing of the pump and from the casing to the surrounding air. If the entire pump motor were submerged in water, then the 1.5 kW dissipated to the air due to motor inefficiency would also be transferred to the surrounding water as heat. This would cause the water temperature to rise more.

5-201 Heat is lost from the steam flowing in a nozzle. The exit velocity and the mass flow rate are to be determined.



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

Analysis (*a*) We take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

Energy balance:

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

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

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

or $$V_2 = \sqrt{2(h_1 - h_2 - q_{\text{out}})}$$

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

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} h_1 = 2769.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 75 \text{ kPa} \\ \text{sat. vap.} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 2.2172 \text{ m}^3\text{/kg} \\ h_2 = 2662.4 \text{ kJ/kg} \end{array}$$

Substituting,

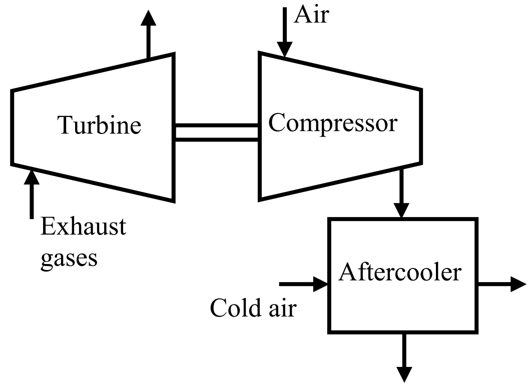
$$V_2 = \sqrt{2(h_1 - h_2 - q_{\text{out}})} = \sqrt{2(2769.1 - 2662.4 - 26)\text{kJ/kg}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2\text{/s}^2}\right)} = \mathbf{401.7\ m/s}$$

(*b*) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{2.2172 \text{ m}^3\text{/kg}} (0.001 \text{ m}^2)(401.7 \text{ m/s}) = \mathbf{0.181\ kg/s}$$

5-202 The turbocharger of an internal combustion engine consisting of a turbine, a compressor, and an aftercooler is considered. The temperature of the air at the compressor outlet and the minimum flow rate of ambient air are to be determined.

Assumptions **1** All processes are steady since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air properties are used for exhaust gases. **4** Air is an ideal gas with constant specific heats. **5** The mechanical efficiency between the turbine and the compressor is 100%. **6** All devices are adiabatic. **7** The local atmospheric pressure is 100 kPa.



Properties The constant pressure specific heats of exhaust gases, warm air, and cold ambient air are taken to be c_p = 1.063, 1.008, and 1.005 kJ/kg·K, respectively (Table A-2b).

Analysis (*a*) An energy balance on turbine gives

$$\dot{W}_T = \dot{m}_{exh} c_{p,exh}\left(T_{exh,1} - T_{exh,2}\right) = (0.02\text{ kg/s})(1.063\text{ kJ/kg}\cdot\text{K})(400-350)\text{K} = 1.063\text{ kW}$$

This is also the power input to the compressor since the mechanical efficiency between the turbine and the compressor is assumed to be 100%. An energy balance on the compressor gives the air temperature at the compressor outlet

$$\dot{W}_C = \dot{m}_a c_{p,a}(T_{a,2} - T_{a,1})$$

$$1.063\text{ kW} = (0.018\text{ kg/s})(1.008\text{ kJ/kg}\cdot\text{K})(T_{a,2} - 50)\text{K} \longrightarrow T_{a,2} = \mathbf{108.6\ °C}$$

(*b*) An energy balance on the aftercooler gives the mass flow rate of cold ambient air

$$\dot{m}_a c_{p,a}(T_{a,2} - T_{a,3}) = \dot{m}_{ca} c_{p,ca}(T_{ca,2} - T_{ca,1})$$

$$(0.018\text{ kg/s})(1.008\text{ kJ/kg}\cdot°\text{C})(108.6-80)°\text{C} = \dot{m}_{ca}(1.005\text{ kJ/kg}\cdot°\text{C})(40-30)°\text{C}$$

$$\dot{m}_{ca} = 0.05161\text{ kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of aftercooler. That is,

$$v_{ca} = \frac{RT}{P} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(30+273\text{ K})}{100\text{ kPa}} = 0.8696\text{ m}^3/\text{kg}$$

$$\dot{V}_{ca} = \dot{m}v_{ca} = (0.05161\text{ kg/s})(0.8696\text{ m}^3/\text{kg}) = \mathbf{0.0449\ m^3/s} = \mathbf{44.9\ L/s}$$

Fundamentals of Engineering (FE) Exam Problems

5-203 Steam is accelerated by a nozzle steadily from a low velocity to a velocity of 210 m/s at a rate of 3.2 kg/s. If the temperature and pressure of the steam at the nozzle exit are 400°C and 2 MPa, the exit area of the nozzle is

(a) 24.0 cm^2 (b) 8.4 cm^2 (c) 10.2 cm^2 (d) 152 cm^2 (e) 23.0 cm^2

Answer (e) 23.0 cm^2

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

```
Vel_1=0 "m/s"
Vel_2=210 "m/s"
m=3.2 "kg/s"
T2=400 "C"
P2=2000 "kPa"
"The rate form of energy balance is E_dot_in - E_dot_out = DELTAE_dot_cv"
v2=VOLUME(Steam_IAPWS,T=T2,P=P2)
m=(1/v2)*A2*Vel_2  "A2 in m^2"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
P2*v2ideal=R*(T2+273)
m=(1/v2ideal)*W1_A2*Vel_2 "assuming ideal gas"
P1*v2ideal=R*T2
m=(1/v2ideal)*W2_A2*Vel_2 "assuming ideal gas and using C"
m=W3_A2*Vel_2 "not using specific volume"
```

5-204 Steam enters a diffuser steadily at 0.5 MPa, 300°C, and 122 m/s at a rate of 3.5 kg/s. The inlet area of the diffuser is

(a) 15 cm^2 (b) 50 cm^2 (c) 105 cm^2 (d) 150 cm^2 (e) 190 cm^2

Answer (d) 150 cm^2

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

```
Vel_1=122 "m/s"
m=3.5 "kg/s"
T1=300 "C"
P1=500 "kPa"
"The rate form of energy balance is E_dot_in - E_dot_out = DELTAE_dot_cv"
v1=VOLUME(Steam_IAPWS,T=T1,P=P1)
m=(1/v1)*A*Vel_1  "A in m^2"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
P1*v1ideal=R*(T1+273)
m=(1/v1ideal)*W1_A*Vel_1 "assuming ideal gas"
P1*v2ideal=R*T1
m=(1/v2ideal)*W2_A*Vel_1 "assuming ideal gas and using C"
m=W3_A*Vel_1 "not using specific volume"
```

5-205 An adiabatic heat exchanger is used to heat cold water at 15°C entering at a rate of 5 kg/s by hot air at 90°C entering also at rate of 5 kg/s. If the exit temperature of hot air is 20°C, the exit temperature of cold water is

(a) 27°C (b) 32°C (c) 52°C (d) 85°C (e) 90°C

Answer (b) 32°C

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

```
C_w=4.18 "kJ/kg-C"
Cp_air=1.005 "kJ/kg-C"
Tw1=15 "C"
m_dot_w=5 "kg/s"
Tair1=90 "C"
Tair2=20 "C"
m_dot_air=5 "kg/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(Tw2-Tw1)

"Some Wrong Solutions with Common Mistakes:"
(Tair1-Tair2)=(W1_Tw2-Tw1) "Equating temperature changes of fluids"
Cv_air=0.718 "kJ/kg.K"
m_dot_air*Cv_air*(Tair1-Tair2)=m_dot_w*C_w*(W2_Tw2-Tw1) "Using Cv for air"
W3_Tw2=Tair1 "Setting inlet temperature of hot fluid = exit temperature of cold fluid"
W4_Tw2=Tair2 "Setting exit temperature of hot fluid = exit temperature of cold fluid"
```

5-206 A heat exchanger is used to heat cold water at 15°C entering at a rate of 2 kg/s by hot air at 100°C entering at rate of 3 kg/s. The heat exchanger is not insulated, and is loosing heat at a rate of 40 kJ/s. If the exit temperature of hot air is 20°C, the exit temperature of cold water is

(a) 44°C (b) 49°C (c) 39°C (d) 72°C (e) 95°C

Answer (c) 39°C

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

```
C_w=4.18 "kJ/kg-C"
Cp_air=1.005 "kJ/kg-C"
Tw1=15 "C"
m_dot_w=2 "kg/s"
Tair1=100 "C"
Tair2=20 "C"
m_dot_air=3 "kg/s"
Q_loss=40 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(Tw2-Tw1)+Q_loss

"Some Wrong Solutions with Common Mistakes:"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(W1_Tw2-Tw1) "Not considering Q_loss"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(W2_Tw2-Tw1)-Q_loss  "Taking heat loss as heat gain"
(Tair1-Tair2)=(W3_Tw2-Tw1) "Equating temperature changes of fluids"
Cv_air=0.718 "kJ/kg.K"
m_dot_air*Cv_air*(Tair1-Tair2)=m_dot_w*C_w*(W4_Tw2-Tw1)+Q_loss  "Using Cv for air"
```

5-207 An adiabatic heat exchanger is used to heat cold water at 15°C entering at a rate of 5 kg/s by hot water at 90°C entering at rate of 4 kg/s. If the exit temperature of hot water is 50°C, the exit temperature of cold water is

(a) 42°C (b) 47°C (c) 55°C (d) 78°C (e) 90°C

Answer (b) 47°C

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

```
C_w=4.18 "kJ/kg-C"
Tcold_1=15 "C"
m_dot_cold=5 "kg/s"
Thot_1=90 "C"
Thot_2=50 "C"
m_dot_hot=4 "kg/s"
Q_loss=0 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_w*(Thot_1-Thot_2)=m_dot_cold*C_w*(Tcold_2-Tcold_1)+Q_loss

"Some Wrong Solutions with Common Mistakes:"
Thot_1-Thot_2=W1_Tcold_2-Tcold_1 "Equating temperature changes of fluids"
W2_Tcold_2=90 "Taking exit temp of cold fluid=inlet temp of hot fluid"
```

5-208 In a shower, cold water at 10°C flowing at a rate of 5 kg/min is mixed with hot water at 60°C flowing at a rate of 2 kg/min. The exit temperature of the mixture will be

(a) 24.3°C (b) 35.0°C (c) 40.0°C (d) 44.3°C (e) 55.2°C

Answer (a) 24.3°C

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

```
C_w=4.18 "kJ/kg-C"
Tcold_1=10 "C"
m_dot_cold=5 "kg/min"
Thot_1=60 "C"
m_dot_hot=2 "kg/min"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_w*Thot_1+m_dot_cold*C_w*Tcold_1=(m_dot_hot+m_dot_cold)*C_w*Tmix
"Some Wrong Solutions with Common Mistakes:"
W1_Tmix=(Tcold_1+Thot_1)/2 "Taking the average temperature of inlet fluids"
```

5-209 In a heating system, cold outdoor air at 10°C flowing at a rate of 6 kg/min is mixed adiabatically with heated air at 70°C flowing at a rate of 3 kg/min. The exit temperature of the mixture is

(a) 30°C (b) 40°C (c) 45°C (d) 55°C (e) 85°C

Answer (a) 30°C

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

```
C_air=1.005 "kJ/kg-C"
Tcold_1=10 "C"
m_dot_cold=6 "kg/min"
Thot_1=70 "C"
m_dot_hot=3 "kg/min"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_air*Thot_1+m_dot_cold*C_air*Tcold_1=(m_dot_hot+m_dot_cold)*C_air*Tmix
"Some Wrong Solutions with Common Mistakes:"
W1_Tmix=(Tcold_1+Thot_1)/2 "Taking the average temperature of inlet fluids"
```

5-210 Hot combustion gases (assumed to have the properties of air at room temperature) enter a gas turbine at 1 MPa and 1500 K at a rate of 0.1 kg/s, and exit at 0.2 MPa and 900 K. If heat is lost from the turbine to the surroundings at a rate of 15 kJ/s, the power output of the gas turbine is

(a) 15 kW (b) 30 kW (c) 45 kW (d) 60 kW (e) 75 kW

Answer (c) 45 kW

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

```
Cp_air=1.005 "kJ/kg-C"
T1=1500 "K"
T2=900 "K"
m_dot=0.1 "kg/s"
Q_dot_loss=15 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_out+Q_dot_loss=m_dot*Cp_air*(T1-T2)
"Alternative: Variable specific heats - using EES data"
W_dot_outvariable+Q_dot_loss=m_dot*(ENTHALPY(Air,T=T1)-ENTHALPY(Air,T=T2))

"Some Wrong Solutions with Common Mistakes:"
W1_Wout=m_dot*Cp_air*(T1-T2) "Disregarding heat loss"
W2_Wout-Q_dot_loss=m_dot*Cp_air*(T1-T2) "Assuming heat gain instead of loss"
```

5-211 Steam expands in a turbine from 4 MPa and 500°C to 0.5 MPa and 250°C at a rate of 1350 kg/h. Heat is lost from the turbine at a rate of 25 kJ/s during the process. The power output of the turbine is

(a) 157 kW (b) 207 kW (c) 182 kW (d) 287 kW (e) 246 kW

Answer (a) 157 kW

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

```
T1=500 "C"
P1=4000 "kPa"
T2=250 "C"
P2=500 "kPa"
m_dot=1350/3600 "kg/s"
Q_dot_loss=25 "kJ/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_out+Q_dot_loss=m_dot*(h1-h2)

"Some Wrong Solutions with Common Mistakes:"
W1_Wout=m_dot*(h1-h2)  "Disregarding heat loss"
W2_Wout-Q_dot_loss=m_dot*(h1-h2)  "Assuming heat gain instead of loss"
u1=INTENERGY(Steam_IAPWS,T=T1,P=P1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
W3_Wout+Q_dot_loss=m_dot*(u1-u2) "Using internal energy instead of enthalpy"
W4_Wout-Q_dot_loss=m_dot*(u1-u2) "Using internal energy and wrong direction for heat"
```

5-212 Steam is compressed by an adiabatic compressor from 0.2 MPa and 150°C to 0.8 MPa and 350°C at a rate of 1.30 kg/s. The power input to the compressor is

(a) 511 kW (b) 393 kW (c) 302 kW (d) 717 kW (e) 901 kW

Answer (a) 511 kW

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

```
P1=200 "kPa"
T1=150 "C"
P2=800 "kPa"
T2=350 "C"
m_dot=1.30 "kg/s"
Q_dot_loss=0 "kJ/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_in-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
W1_Win-Q_dot_loss=(h2-h1)/m_dot   "Dividing by mass flow rate instead of multiplying"
W2_Win-Q_dot_loss=h2-h1  "Not considering mass flow rate"
u1=INTENERGY(Steam_IAPWS,T=T1,P=P1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
W3_Win-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
W4_Win-Q_dot_loss=u2-u1 "Using internal energy and ignoring mass flow rate"
```

5-213 Refrigerant-134a is compressed by a compressor from the saturated vapor state at 0.14 MPa to 1.2 MPa and 70°C at a rate of 0.108 kg/s. The refrigerant is cooled at a rate of 1.10 kJ/s during compression. The power input to the compressor is

(a) 5.54 kW (b) 7.33 kW (c) 6.64 kW (d) 7.74 kW (e) 8.13 kW

Answer (d) 7.74 kW

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

```
P1=140 "kPa"
x1=1
P2=1200 "kPa"
T2=70 "C"
m_dot=0.108 "kg/s"
Q_dot_loss=1.10 "kJ/s"
h1=ENTHALPY(R134a,x=x1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_in-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
W1_Win+Q_dot_loss=m_dot*(h2-h1)   "Wrong direction for heat transfer"
W2_Win =m_dot*(h2-h1)  "Not considering heat loss"
u1=INTENERGY(R134a,x=x1,P=P1)
u2=INTENERGY(R134a,T=T2,P=P2)
W3_Win-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
W4_Win+Q_dot_loss=u2-u1 "Using internal energy and wrong direction for heat transfer"
```

5-214 Refrigerant-134a expands in an adiabatic turbine from 1.2 MPa and 100°C to 0.18 MPa and 50°C at a rate of 1.25 kg/s. The power output of the turbine is

(a) 46.3 kW (b) 66.4 kW (c) 72.7 kW (d) 89.2 kW (e) 112.0 kW

Answer (a) 46.3 kW

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

```
P1=1200 "kPa"
T1=100 "C"
P2=180 "kPa"
T2=50 "C"
m_dot=1.25 "kg/s"
Q_dot_loss=0 "kJ/s"
h1=ENTHALPY(R134a,T=T1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
-W_dot_out-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
-W1_Wout-Q_dot_loss=(h2-h1)/m_dot   "Dividing by mass flow rate instead of multiplying"
-W2_Wout-Q_dot_loss=h2-h1  "Not considering mass flow rate"
u1=INTENERGY(R134a,T=T1,P=P1)
u2=INTENERGY(R134a,T=T2,P=P2)
-W3_Wout-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
-W4_Wout-Q_dot_loss=u2-u1 "Using internal energy and ignoring mass flow rate"
```

5-215 Refrigerant-134a at 1.4 MPa and 90°C is throttled to a pressure of 0.6 MPa. The temperature of the refrigerant after throttling is

(a) 22°C (b) 56°C (c) 82°C (d) 80°C (e) 90.0°C

Answer (d) 80°C

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

```
P1=1400 "kPa"
T1=90 "C"
P2=600 "kPa"
h1=ENTHALPY(R134a,T=T1,P=P1)
T2=TEMPERATURE(R134a,h=h1,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming the temperature to remain constant"
W2_T2=TEMPERATURE(R134a,x=0,P=P2) "Taking the temperature to be the saturation temperature at P2"
u1=INTENERGY(R134a,T=T1,P=P1)
W3_T2=TEMPERATURE(R134a,u=u1,P=P2) "Assuming u=constant"
v1=VOLUME(R134a,T=T1,P=P1)
W4_T2=TEMPERATURE(R134a,v=v1,P=P2) "Assuming v=constant"
```

5-216 Air at 20°C and 5 atm is throttled by a valve to 2 atm. If the valve is adiabatic and the change in kinetic energy is negligible, the exit temperature of air will be

(a) 10°C (b) 14°C (c) 17°C (d) 20°C (e) 24°C

Answer (d) 20°C

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

```
"The temperature of an ideal gas remains constant during throttling, and thus T2=T1"
T1=20 "C"
P1=5 "atm"
P2=2 "atm"
T2=T1 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1*P1/P2 "Assuming v=constant and using C"
W2_T2=(T1+273)*P1/P2-273 "Assuming v=constant and using K"
W3_T2=T1*P2/P1 "Assuming v=constant and pressures backwards and using C"
W4_T2=(T1+273)*P2/P1 "Assuming v=constant and pressures backwards and using K"
```

5-217 Steam at 1 MPa and 300°C is throttled adiabatically to a pressure of 0.4 MPa. If the change in kinetic energy is negligible, the specific volume of the steam after throttling will be

(a) 0.358 m^3/kg (b) 0.233 m^3/kg (c) 0.375 m^3/kg (d) 0.646 m^3/kg (e) 0.655 m^3/kg

Answer (d) 0.646 m^3/kg

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

```
P1=1000 "kPa"
T1=300 "C"
P2=400 "kPa"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
v2=VOLUME(Steam_IAPWS,h=h1,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_v2=VOLUME(Steam_IAPWS,T=T1,P=P2)  "Assuming the volume to remain constant"
u1=INTENERGY(Steam,T=T1,P=P1)
W2_v2=VOLUME(Steam_IAPWS,u=u1,P=P2) "Assuming u=constant"
W3_v2=VOLUME(Steam_IAPWS,T=T1,P=P2)  "Assuming T=constant"
```

5-218 Air is to be heated steadily by an 8-kW electric resistance heater as it flows through an insulated duct. If the air enters at 50°C at a rate of 2 kg/s, the exit temperature of air will be

(a) 46.0°C (b) 50.0°C (c) 54.0°C (d) 55.4°C (e) 58.0°C

Answer (c) 54.0°C

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

```
Cp=1.005 "kJ/kg-C"
T1=50 "C"
 m_dot=2 "kg/s"
W_dot_e=8 "kJ/s"
W_dot_e=m_dot*Cp*(T2-T1)

"Checking using data from EES table"
W_dot_e=m_dot*(ENTHALPY(Air,T=T_2table)-ENTHALPY(Air,T=T1))

"Some Wrong Solutions with Common Mistakes:"
Cv=0.718 "kJ/kg.K"
W_dot_e=Cp*(W1_T2-T1) "Not using mass flow rate"
W_dot_e=m_dot*Cv*(W2_T2-T1) "Using Cv"
W_dot_e=m_dot*Cp*W3_T2 "Ignoring T1"
```

5-219 Saturated water vapor at 50°C is to be condensed as it flows through a tube at a rate of 0.35 kg/s. The condensate leaves the tube as a saturated liquid at 50°C. The rate of heat transfer from the tube is

(a) 73 kJ/s (b) 980 kJ/s (c) 2380 kJ/s (d) 834 kJ/s (e) 907 kJ/s

Answer (d) 834 kJ/s

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

```
T1=50 "C"
 m_dot=0.35 "kg/s"
h_f=ENTHALPY(Steam_IAPWS,T=T1,x=0)
h_g=ENTHALPY(Steam_IAPWS,T=T1,x=1)
h_fg=h_g-h_f
Q_dot=m_dot*h_fg

"Some Wrong Solutions with Common Mistakes:"
W1_Q=m_dot*h_f  "Using hf"
W2_Q=m_dot*h_g "Using hg"
W3_Q=h_fg "not using mass flow rate"
W4_Q=m_dot*(h_f+h_g) "Adding hf and hg"
```

5-220 … 5-224 Design and Essay Problems