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.

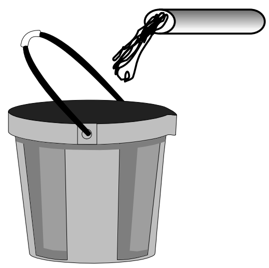
Assumptions **1** Water is an incompressible substance. **2** Flow through the hose is steady. **3** There is no waste of water by splashing.

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

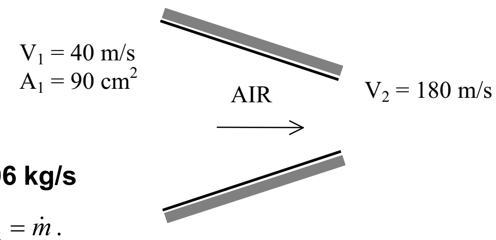
$$V_e = \frac{\dot{V}}{A_e} = \frac{\dot{V}}{\pi D_e^2/4} = \frac{0.04363\text{ ft}^3\text{/s}}{[\pi(0.5/12\text{ ft})^2/4]} = \mathbf{32\ ft/s}$$

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

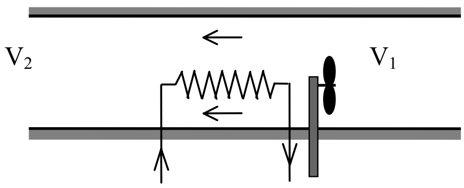
$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.796\text{ kg/s}}{(0.762\,\text{kg/m}^3)(180\,\text{m/s})} = 0.0058\,\text{m}^2 = 58\,\mathbf{cm^2}$$

5-8 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,



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

$$\rho_1 A V_1 = \rho_2 A V_2$$

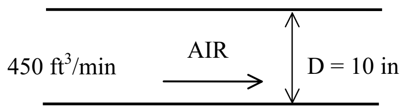
$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20\text{ kg/m}^3}{1.05\text{ kg/m}^3} = 1.14 \quad \text{(or, and increase of } \mathbf{14\%}\text{)}$$

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

5-9E The ducts of an air-conditioning system pass through an open area. The inlet velocity and the mass flow rate of air are to be determined.

Assumptions Flow through the air conditioning duct is steady.

Properties The density of air is given to be 0.078 lbm/ft^3 at the inlet.



Analysis The inlet velocity of air and the mass flow rate through the duct are

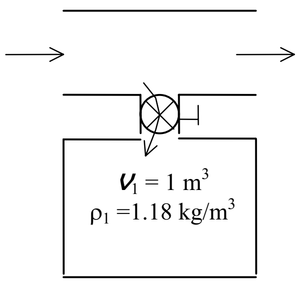
$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi D^2/4} = \frac{450\text{ ft}^3\text{/min}}{\pi(10/12\text{ ft})^2/4} = \mathbf{825\ ft/min = 13.8\ ft/s}$$

$$\dot{m} = \rho_1 \dot{V}_1 = (0.078\,\text{lbm/ft}^3)(450\text{ ft}^3/\text{min}) = 35.1\text{ lbm/min} = \mathbf{0.585\,lbm/s}$$

5-10 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until the density rises to a specified level. The mass of air that entered the tank is to be determined.

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

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



Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1 = \rho_2 V - \rho_1 V$

Substituting,

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

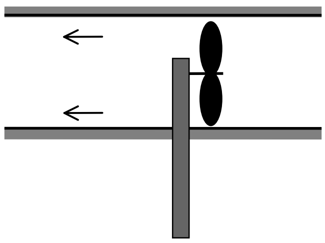
Therefore, 6.02 kg of mass entered the tank.

5-11 The ventilating fan of the bathroom of a building runs continuously. The mass of air "vented out" per day is to be determined.

Assumptions Flow through the fan is steady.

Properties The density of air in the building is given to be 1.20 kg/m^3.

Analysis The mass flow rate of air vented out is



$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (1.20\ \text{kg/m}^3)(0.030\ \text{m}^3/\text{s}) = 0.036\ \text{kg/s}$$

Then the mass of air vented out in 24 h becomes

$$m = \dot{m}_{\text{air}} \Delta t = (0.036\ \text{kg/s})(24 \times 3600\ \text{s}) = \mathbf{3110\ kg}$$

Discussion Note that more than 3 tons of air is vented out by a bathroom fan in one day.

5-12 A desktop computer is to be cooled by a fan at a high elevation where the air density is low. The mass flow rate of air through the fan and the diameter of the casing for a given velocity are to be determined.

Assumptions Flow through the fan is steady.

Properties The density of air at a high elevation is given to be 0.7 kg/m^3.

Analysis The mass flow rate of air is

$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (0.7\ \text{kg/m}^3)(0.34\ \text{m}^3/\text{min}) = 0.238\ \text{kg/min} = \mathbf{0.0040\ \ kg/s}$$

If the mean velocity is 110 m/min, the diameter of the casing is

$$\dot{V} = AV = \frac{\pi D^2}{4} V \rightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.34\ \text{m}^3/\text{min})}{\pi(110\ \text{m/min})}} = \mathbf{0.063\ m}$$

Therefore, the diameter of the casing must be at least 6.3 cm to ensure that the mean velocity does not exceed 110 m/min.

Discussion This problem shows that engineering systems are sized to satisfy certain constraints imposed by certain considerations.

5-13 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.

5-14 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

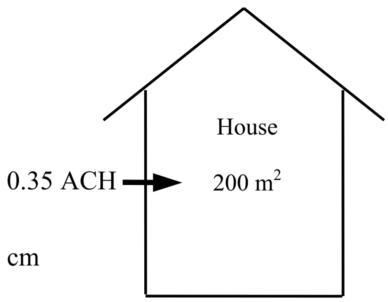
$$V_{\text{room}} = (2.7\ \text{m})(200\ \text{m}^2) = 540\ \text{m}^3$$
$$\dot{V} = V_{\text{room}} \times \text{ACH} = (540\ \text{m}^3)(0.35/\text{h}) = 189\ \text{m}^3/\text{h} = 189{,}000\ \text{L/h} = \mathbf{3150\ \ L/min}$$

The volume flow rate of fresh air can be expressed as

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

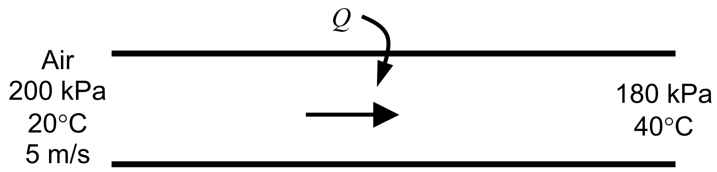
Solving for the diameter D and substituting,

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



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.

5-15 Air flows through a pipe. Heat is supplied to the air. The volume flow rates of air at the inlet and exit, the velocity at the exit, and the mass flow rate are to be determined.



Properties The gas constant for air is 0.287 kJ/kg.K (Table A-2).

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

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

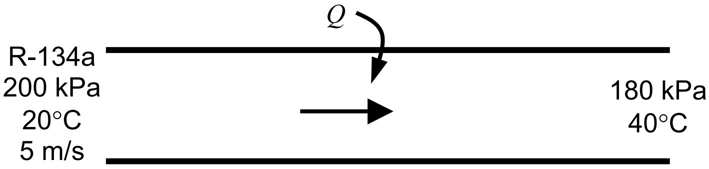
$$\dot{m} = \rho_1 A_c V_1 = \frac{P_1}{RT_1}\frac{\pi D^2}{4} V_1 = \frac{(200\text{ kPa})}{(0.287\text{ kJ/kg.K})(20+273\text{ K})}\frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.7318\ kg/s}$$

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

$$\dot{V}_2 = \frac{\dot{m}}{\rho_2} = \frac{\dot{m}}{\frac{P_2}{RT_2}} = \frac{0.7318\text{ kg/s}}{\frac{(180\text{ kPa})}{(0.287\text{ kJ/kg.K})(40+273\text{ K})}} = \mathbf{0.3654\ m^3/s}$$

$$V_2 = \frac{\dot{V}_2}{A_c} = \frac{0.3654\text{ m}^3/\text{s}}{\frac{\pi (0.28\text{ m})^2}{4}} = \mathbf{5.94\ m/s}$$

5-16 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)

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} v_1 = 0.1142\text{ m}^3/\text{kg} \qquad \left.\begin{array}{l} P_1 = 180\text{ kPa} \\ T_1 = 40°\text{C} \end{array}\right\} v_2 = 0.1374\text{ m}^3/\text{kg}$$

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

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

$$\dot{m} = \frac{1}{v_1} A_c V_1 = \frac{1}{v_1}\frac{\pi D^2}{4} V_1 = \frac{1}{0.1142\text{ m}^3/\text{kg}}\frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{2.696\ kg/s}$$

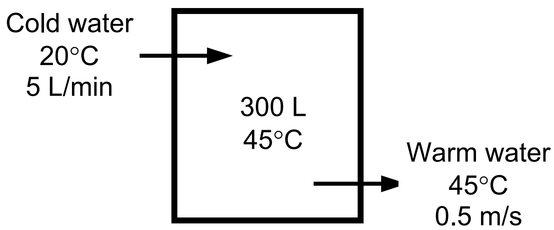
(c) Noting that mass flow rate is constant, the volume flow rate and the velocity at the exit of the pipe are determined from

$$\dot{V}_2 = \dot{m} v_2 = (2.696\text{ kg/s})(0.1374\text{ m}^3/\text{kg}) = \mathbf{0.3705\ m^3/s}$$

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

5-17 Warm water is withdrawn from a solar water storage tank while cold water enters the tank. The amount of water in the tank in a 20-minute period is to be determined.

Properties The density of water is taken to be 1000 kg/m^3 for both cold and warm water.



Analysis The initial mass in the tank is first determined from

$$m_1 = \rho V_{\text{tank}} = (1000\text{ kg/m}^3)(0.3\text{ m}^3) = 300\text{ kg}$$

The amount of warm water leaving the tank during a 20-min period is

$$m_e = \rho A_c V \Delta t = (1000\text{ kg/m}^3)\frac{\pi (0.02\text{ m})^2}{4}(0.5\text{ m/s})(20 \times 60\text{ s}) = 188.5\text{ kg}$$

The amount of cold water entering the tank during a 20-min period is

$$m_i = \rho \dot{V}_c \Delta t = (1000\text{ kg/m}^3)(0.005\text{ m}^3/\text{min})(20\text{ min}) = 100\text{ kg}$$

The final mass in the tank can be determined from a mass balance as

$$m_i - m_e = m_2 - m_1 \longrightarrow m_2 = m_1 + m_i - m_e = 300 + 100 - 188.5 = \mathbf{211.5\ kg}$$

Flow Work and Energy Transfer by Mass

5-18C Energy can be transferred to or from a control volume as heat, various forms of work, and by mass.

5-19C Flow energy or flow work is the energy needed to push a fluid into or out of a control volume. Fluids at rest do not possess any flow energy.

5-20C Flowing fluids possess flow energy in addition to the forms of energy a fluid at rest possesses. The total energy of a fluid at rest consists of internal, kinetic, and potential energies. The total energy of a flowing fluid consists of internal, kinetic, potential, and flow energies.

5-21E 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.

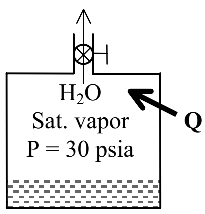
Properties The properties of saturated liquid water and water vapor at 30 psia are v_f = 0.01700 ft^3/lbm, v_g = 13.749 ft^3/lbm, u_g = 1087.8 Btu/lbm, and h_g = 1164.1 Btu/lbm (Table A-5E).

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

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

$$\dot{m} = \frac{m}{\Delta t} = \frac{3.145\text{ lbm}}{45\text{ min}} = 0.0699\text{ lbm/min} = \mathbf{1.165\times10^{-3}\ lbm/s}$$

$$V = \frac{\dot{m}}{\rho_g A_c} = \frac{\dot{m}v_g}{A_c} = \frac{(1.165\times10^{-3}\text{ lbm/s})(13.749\text{ ft}^3/\text{lbm})}{0.15\text{ in}^2}\left(\frac{144\text{ in}^2}{1\text{ ft}^2}\right) = \mathbf{15.4\ ft/s}$$



(*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,

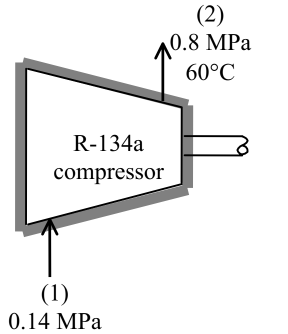
$$\dot{E}_{\text{mass}} = \dot{m}\theta = (1.165\times10^{-3}\ \text{lbm/s})(1164.1\ \ \text{Btu/lbm}) = \mathbf{1.356\ \ Btu/s}$$

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.

5-22 Refrigerant-134a enters a compressor as a saturated vapor at a specified pressure, and leaves as superheated vapor at a specified rate. The rates of energy transfer by mass into and out of the compressor are to be determined.

Assumptions **1** The flow of the refrigerant through the compressor is steady. **2** The kinetic and potential energies are negligible, and thus they are not considered.

Properties The enthalpy of refrigerant-134a at the inlet and the exit are (Tables A-12 and A-13)



$$h_1 = h_{g@0.14\text{ MPa}} = 239.16\text{ kJ/kg} \qquad \left.\begin{array}{l} P_2 = 0.8\text{ MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 296.81\text{ kJ/kg}$$

Analysis Noting that the total energy of a flowing fluid is equal to its enthalpy when the kinetic and potential energies are negligible, and that the rate of energy transfer by mass is equal to the product of the mass flow rate and the total energy of the fluid per unit mass, the rates of energy transfer by mass into and out of the compressor are

$$\dot{E}_{\text{mass, in}} = \dot{m}\theta_{in} = \dot{m}h_1 = (0.06\text{ kg/s})(239.16\text{ kJ/kg}) = 14.35\text{ kJ/s} = \mathbf{14.35\text{ kW}}$$

$$\dot{E}_{\text{mass, out}} = \dot{m}\theta_{out} = \dot{m}h_2 = (0.06\text{ kg/s})(296.81\text{ kJ/kg}) = 17.81\text{ kJ/s} = \mathbf{17.81\text{kW}}$$

Discussion The numerical values of the energy entering or leaving a device by mass alone does not mean much since this value depends on the reference point selected for enthalpy (it could even be negative). The significant quantity here is the difference between the outgoing and incoming energy flow rates, which is

$$\Delta\dot{E}_{\text{mass}} = \dot{E}_{\text{mass, out}} - \dot{E}_{\text{mass, in}} = 17.81 - 14.35 = 3.46\text{ kW}$$

This quantity represents the rate of energy transfer to the refrigerant in the compressor.

5-23 Warm air in a house is forced to leave by the infiltrating cold outside air at a specified rate. The net energy loss due to mass transfer is to be determined.

Assumptions **1** The flow of the air into and out of the house through the cracks is steady. **2** The kinetic and potential energies are negligible. **3** Air is an ideal gas with constant specific heats at room temperature.

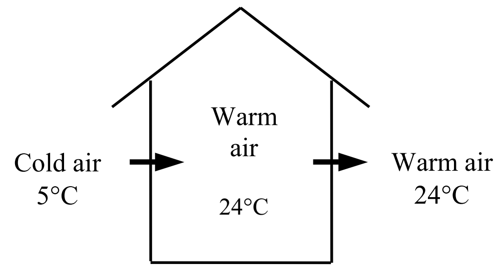
Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2).

Analysis The density of air at the indoor conditions and its mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(24+273)\text{K}} = 1.189\text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.189\text{ kg/m}^3)(150\text{ m}^3/\text{h}) = 178.35\text{ kg/h} = 0.0495\text{ kg/s}$$

Noting that the total energy of a flowing fluid is equal to its enthalpy when the kinetic and potential energies are negligible, and that the rate of energy transfer by mass is equal to the product of the mass flow rate and the total energy of the fluid per unit mass, the rates of energy transfer by mass into and out of the house by air are



$$\dot{E}_{\text{mass, in}} = \dot{m}\theta_{in} = \dot{m}h_1$$

$$\dot{E}_{\text{mass, out}} = \dot{m}\theta_{out} = \dot{m}h_2$$

The net energy loss by air infiltration is equal to the difference between the outgoing and incoming energy flow rates, which is

$$\Delta\dot{E}_{\text{mass}} = \dot{E}_{\text{mass, out}} - \dot{E}_{\text{mass, in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

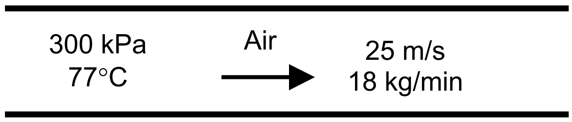
$$= (0.0495\text{ kg/s})(1.005\text{ kJ/kg}\cdot°\text{C})(24 - 5)°\text{C} = 0.945\text{ kJ/s} = \mathbf{0.945\text{ kW}}$$

This quantity represents the rate of energy transfer to the refrigerant in the compressor.

Discussion The rate of energy loss by infiltration will be less in reality since some air will leave the house before it is fully heated to 24°C.

5-24 Air flows steadily in a pipe at a specified state. The diameter of the pipe, the rate of flow energy, and the rate of energy transport by mass are to be determined. Also, the error involved in the determination of energy transport by mass is to be determined.

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



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

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

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

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

(*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%**.

Steady Flow Energy Balance: Nozzles and Diffusers

5-25C A steady-flow system involves no changes with time anywhere within the system or at the system boundaries

5-26C No.

5-27C It is mostly converted to internal energy as shown by a rise in the fluid temperature.

5-28C The kinetic energy of a fluid increases at the expense of the internal energy as evidenced by a decrease in the fluid temperature.

5-29C 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.

5-30 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.

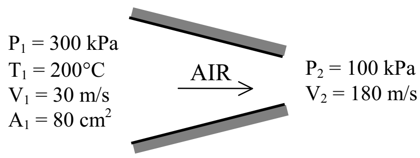
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

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

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

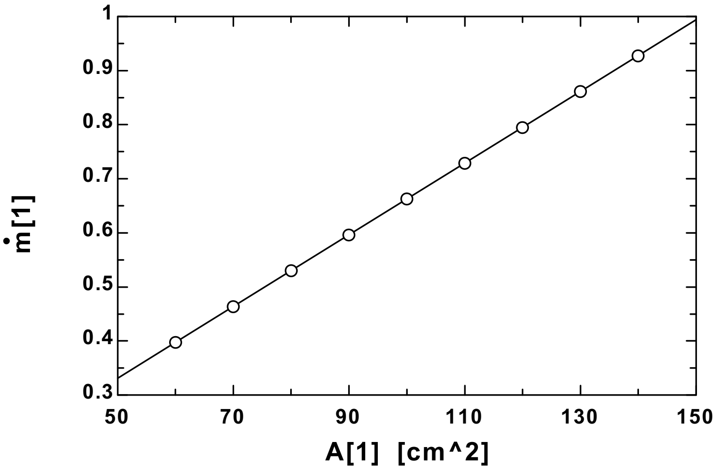
$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 \longrightarrow 0.5304 \text{ kg/s} = \frac{1}{1.313 \text{ m}^3/\text{kg}} A_2 (180 \text{ m/s}) \rightarrow A_2 = 0.00387 \text{ m}^2 = \mathbf{38.7 \text{ cm}^2}$$

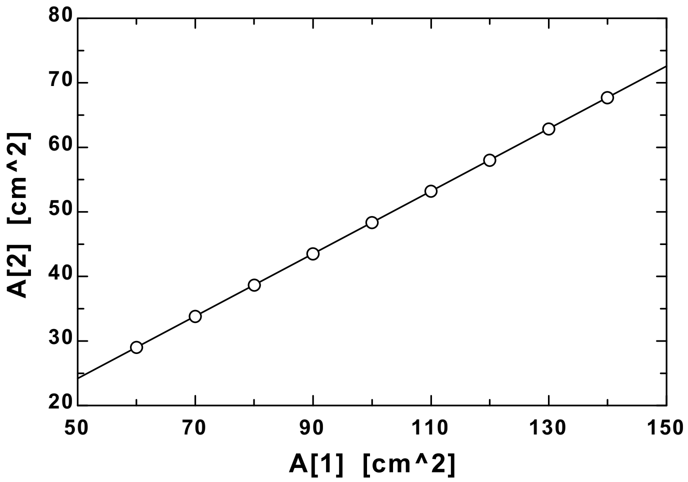
5-31 EES Problem 5-30 is reconsidered. The effect of the inlet area on the mass flow rate, exit velocity, and the exit area as the inlet area varies from 50 cm^2 to 150 cm^2 is to be investigated, and the final results are to be plotted against the inlet area.

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

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

"System: control volume for the nozzle"
"Property relation: Air is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"
"Knowns - obtain from the input diagram"
WorkFluid$ = 'Air'
T[1] = 200  [C]
P[1] = 300   [kPa]
Vel[1] = 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



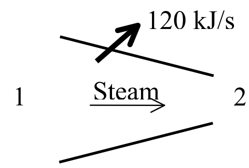


5-32 Steam is accelerated in a nozzle from a velocity of 80 m/s. The mass flow rate, the exit velocity, and the exit area of the nozzle are to be determined.

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

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

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

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

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

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

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

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

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

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

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

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

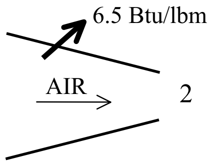
5-33E Air is accelerated in a nozzle from 150 ft/s to 900 ft/s. The exit temperature of air and the exit area of the nozzle are to be determined.

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

Properties The enthalpy of air at the inlet is h_1 = 143.47 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 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

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



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

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

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

or,

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

$$= -6.5 \text{ Btu/lbm} + 143.47 \text{ Btu/lbm} - \frac{(900 \text{ ft/s})^2 - (150 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)$$

$$= 121.2 \text{ Btu/lbm}$$

Thus, from Table A-17E, T_2 = **507 R**

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

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

$$A_2 = \frac{(508/14.7)(150 \text{ ft/s})}{(600/50)(900 \text{ ft/s})}\left(0.1 \text{ ft}^2\right) = \mathbf{0.048\ ft^2}$$

5-34 **[***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.

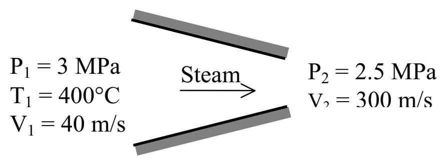
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

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

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



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

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

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

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,

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

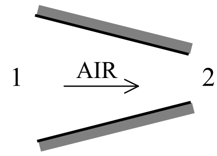
5-35 Air is accelerated in a nozzle from 120 m/s to 380 m/s. The exit temperature and pressure of air are to be determined.

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 500 K is h_1 = 503.02 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 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

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



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

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

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

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 503.02 \text{ kJ/kg} - \frac{(380 \text{ m/s})^2 - (120 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 438.02 \text{ kJ/kg}$$

Then from Table A-17 we read T_2 = **436.5 K**

(*b*) The exit pressure 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,

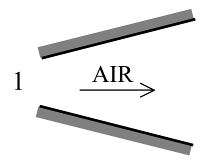
$$P_2 = \frac{A_1 T_2 V_1}{A_2 T_1 V_2} P_1 = \frac{2}{1} \frac{(436.5 \text{ K})(120 \text{ m/s})}{(500 \text{ K})(380 \text{ m/s})} (600 \text{ kPa}) = \mathbf{330.8 \text{ kPa}}$$

5-36 Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

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

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpy of air at the inlet temperature of 400 K is h_1 = 400.98 kJ/kg (Table A-17).

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



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

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

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

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

or,

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

From Table A-17, $T_2 = \mathbf{425.6\ K}$

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

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(425.6 \text{ K})}{(100 \text{ kPa})} = 1.221 \text{ m}^3/\text{kg}$$

From conservation of mass,

$$\dot{m} = \frac{1}{v_2} A_2 V_2 \longrightarrow A_2 = \frac{\dot{m} v_2}{V_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = \mathbf{0.0678\ m^2}$$

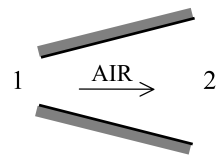
5-37E Air is decelerated in a diffuser from 600 ft/s to a low velocity. The exit temperature and the exit velocity of air are to be determined.

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,

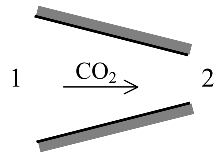
$$V_2 = \frac{A_1 T_2 P_1}{A_2 T_1 P_2} V_1 = \frac{1}{5}\frac{(510 \text{ R})(13 \text{ psia})}{(480 \text{ R})(14.5 \text{ psia})}(600 \text{ ft/s}) = \mathbf{114.3 \text{ ft/s}}$$

5-38 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



$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(773 \text{ K})}{1000 \text{ kPa}} = 0.146 \text{ m}^3/\text{kg}$$

Thus,

$$\dot{m} = \frac{1}{\boldsymbol{v}_1}A_1V_1 \longrightarrow V_1 = \frac{\dot{m}\boldsymbol{v}_1}{A_1} = \frac{(6000/3600 \text{ kg/s})(0.146 \text{ m}^3/\text{kg})}{40\times10^{-4}\text{m}^2} = \mathbf{60.8 \text{ m/s}}$$

(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

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

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

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

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

Substituting,

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

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

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

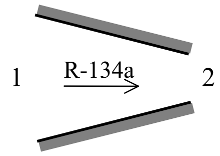
Then the exit temperature of CO_2 from Table A-20 is obtained to be $T_2 = \mathbf{685.8 \text{ K}}$

5-39 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

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

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

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

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

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 $\quad V_2 = \mathbf{409.9\ m/s}$

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

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

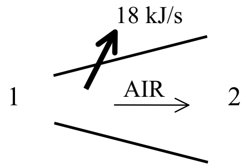
5-40 Air is decelerated in a diffuser from 220 m/s. The exit velocity and the exit pressure of air are to be determined.

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** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpies are (Table A-17)

$$T_1 = 27°\text{C} = 300\text{ K} \quad \rightarrow \quad h_1 = 300.19\text{ kJ/kg}$$
$$T_2 = 42°\text{C} = 315\text{ K} \quad \rightarrow \quad h_2 = 315.27\text{ kJ/kg}$$

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



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

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

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

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

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

$$-18\text{ kJ/s} = (2.5\text{ kg/s})\left((315.27 - 300.19)\text{ kJ/kg} + \frac{V_2^2 - (220\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right)$$

It yields $\qquad$ $\mathbf{V_2 = 62.0\ m/s}$

(*b*) The exit pressure of air is determined from the conservation of mass and the ideal gas relations,

$$\dot{m} = \frac{1}{\boldsymbol{v}_2}A_2V_2 \longrightarrow \boldsymbol{v}_2 = \frac{A_2V_2}{\dot{m}} = \frac{(0.04\text{ m}^2)(62\text{ m/s})}{2.5\text{ kg/s}} = 0.992\text{ m}^3/\text{kg}$$

and

$$P_2\boldsymbol{v}_2 = RT_2 \longrightarrow P_2 = \frac{RT_2}{\boldsymbol{v}_2} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(315\text{ K})}{0.992\text{ m}^3/\text{kg}} = \mathbf{91.1\ kPa}$$

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)

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

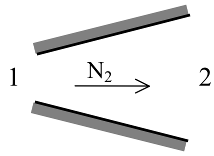
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



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,

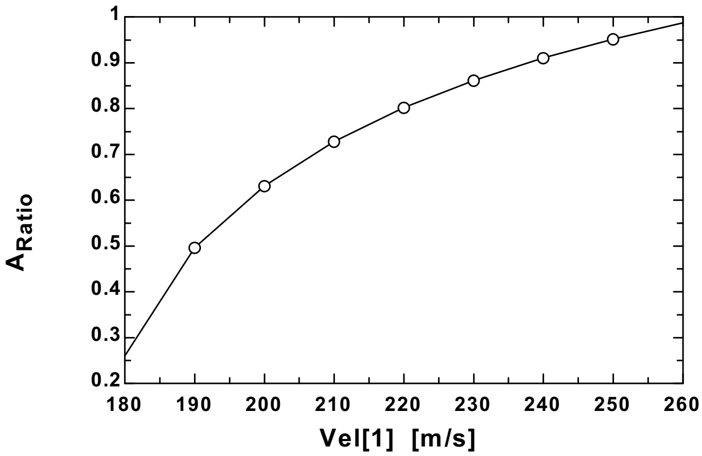
$$\frac{A_1}{A_2} = \left(\frac{T_1/P_1}{T_2/P_2}\right)\frac{V_2}{V_1} = \frac{(280\ \text{K}/60\ \text{kPa})(93.0\ \text{m/s})}{(295\ \text{K}/85\ \text{kPa})(200\ \text{m/s})} = \mathbf{0.625}$$

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.

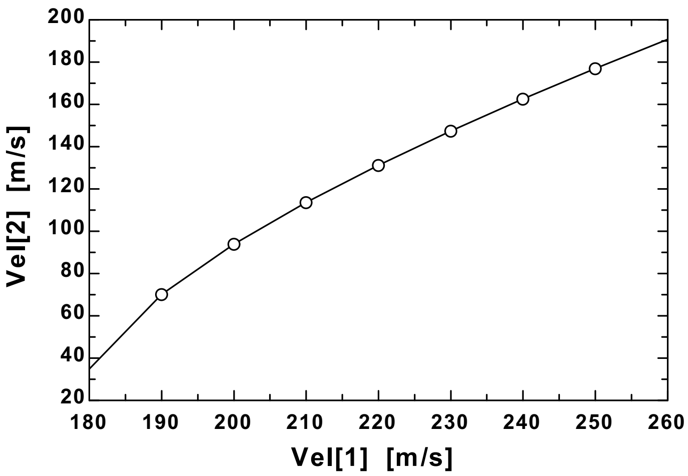
Analysis The problem is solved using EES, and the solution is given below.

```
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



A_{Ratio}
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
180
190
200
210
220
230
240
250
260
Vel[1] [m/s]



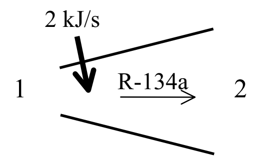
Vel[2] [m/s]
200
180
160
140
120
100
80
60
40
20
180
190
200
210
220
230
240
250
260
Vel[1] [m/s]

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)

$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.025621 \text{ m}^3/\text{kg} \\ h_1 = 267.29 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ T_2 = 40°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.023375 \text{ m}^3/\text{kg} \\ h_2 = 274.17 \text{ kJ/kg} \end{array}$$

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

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow V_2 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}\frac{A_1}{A_2} V_1 = \frac{1}{1.8}\frac{(0.023375 \text{ m}^3/\text{kg})}{(0.025621 \text{ m}^3/\text{kg})}(120 \text{ m/s}) = \mathbf{60.8 \text{ m/s}}$$

(*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

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

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

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

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

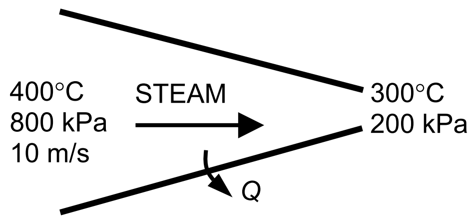
Substituting, the mass flow rate of the refrigerant is determined to be

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

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:

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

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

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

or

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

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

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

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

The mass flow rate of the steam is

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

Substituting,

$$3267.7 \text{ kJ/kg} + \frac{(10 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3072.1 \text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) + \frac{25 \text{ kJ/s}}{2.082 \text{ kg/s}}$$

$$\longrightarrow V_2 = \mathbf{606 \text{ m/s}}$$

The volume flow rate at the exit of the nozzle is

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

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.

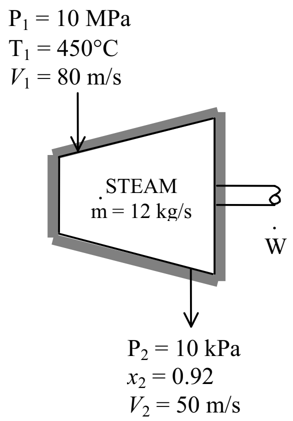
5-48C No.

5-49 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)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.029782 \text{ m}^3/\text{kg} \\ h_1 = 3242.4 \text{ kJ/kg} \end{array}$$



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

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

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

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

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

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

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

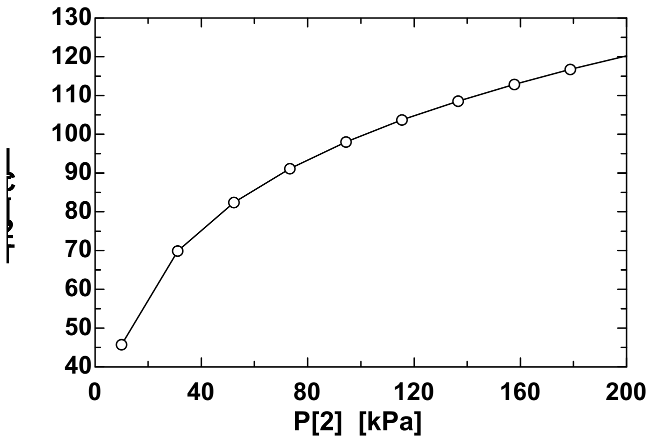
(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow A_1 = \frac{\dot{m}\boldsymbol{v}_1}{V_1} = \frac{(12 \text{ kg/s})(0.029782 \text{ m}^3/\text{kg})}{80 \text{ m/s}} = \mathbf{0.00447\ m^2}$$

5-50 EES Problem 5-49 is reconsidered. The effect of the turbine exit pressure on the power output of the turbine as the exit pressure varies from 10 kPa to 200 kPa is to be investigated. The power output is to be plotted against the exit pressure.

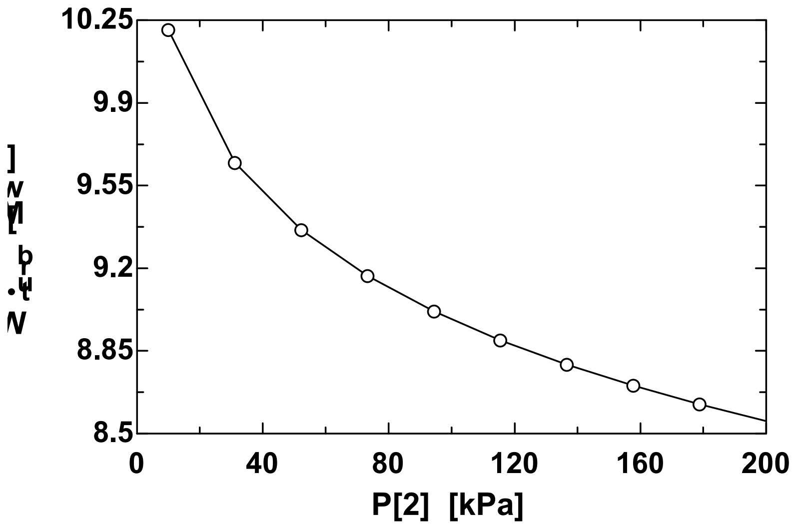
Analysis The problem is solved using EES, and the solution is given below.

```
"Knowns "
T[1] = 450 [C]
P[1] = 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



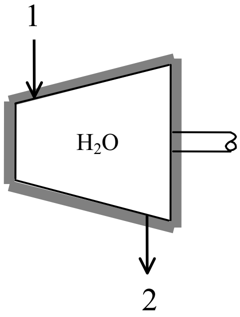
5-51 Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

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

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

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

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



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

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

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

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

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

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

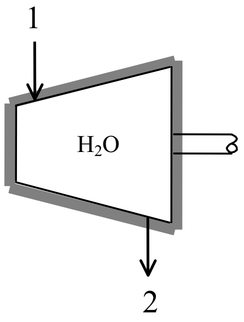
$$5000 \text{ kJ/s} = -\dot{m}(2344.7 - 3375.1) \text{ kJ/kg} \longrightarrow \dot{m} = \mathbf{4.852 \text{ kg/s}}$$

5-52E Steam expands in a turbine. The rate of heat loss from the steam for a power output of 4 MW is to be determined.

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

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

$$\left.\begin{array}{l} P_1 = 1000 \text{ psia} \\ T_1 = 900°\text{F} \end{array}\right\} h_1 = 1448.6 \text{ Btu/lbm} \qquad \left.\begin{array}{l} P_2 = 5 \text{ psia} \\ sat.vapor \end{array}\right\} h_2 = 1130.7 \text{ Btu/lbm}$$



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

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

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

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

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

Substituting,

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

5-53 Steam expands in a turbine. The exit temperature of the steam for a power output of 2 MW is to be determined.

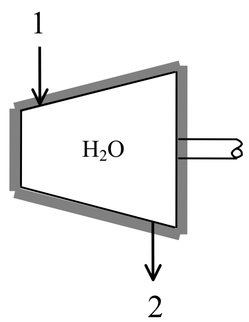
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

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



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

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

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

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

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

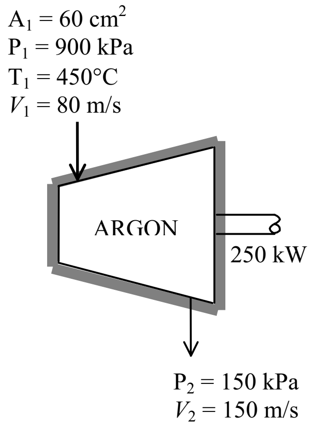
5-54 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is to be determined.

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**

5-55E Air expands in a turbine. The mass flow rate of air and the power output of the turbine are to be determined.

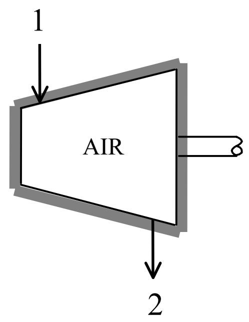
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R. The constant pressure specific heat of air at the average temperature of (900 + 300)/2 = 600°F is c_p = 0.25 Btu/lbm·°F (Table A-2a)

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 mass flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(1360 \text{ R})}{150 \text{ psia}} = 3.358 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 = \frac{1}{3.358 \text{ ft}^3/\text{lbm}} (0.1 \text{ ft}^2)(350 \text{ ft/s}) = \mathbf{10.42 \text{ lbm/s}}$$



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

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

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

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

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

Substituting,

$$\dot{W}_{\text{out}} = -(10.42 \text{ lbm/s})\left[(0.250 \text{ Btu/lbm}\cdot^\circ\text{F})(300 - 900)^\circ\text{F} + \frac{(700 \text{ ft/s})^2 - (350 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)\right]$$

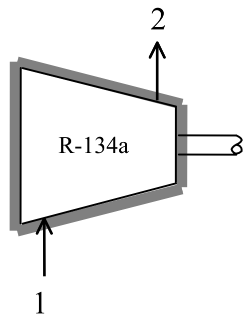
$$= 1486.5 \text{ Btu/s} = \mathbf{1568 \text{ kW}}$$

5-56 Refrigerant-134a is compressed steadily by a compressor. The power input to the compressor and the volume flow rate of the refrigerant at the compressor inlet are to be determined.

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

Properties From the refrigerant tables (Tables A-11 through 13)

$$\left.\begin{array}{l} T_1 = -24°\text{C} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.17395 \text{ m}^3/\text{kg} \\ h_1 = 235.92 \text{ kJ/kg} \end{array} \qquad \left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 296.81 \text{ kJ/kg}$$



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

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

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

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

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

Substituting, $\dot{W}_{in} = (1.2 \text{ kg/s})(296.81 - 235.92)\text{kJ/kg} = \mathbf{73.06 \text{ kJ/s}}$

(*b*) The volume flow rate of the refrigerant at the compressor inlet is

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (1.2 \text{ kg/s})(0.17395 \text{ m}^3/\text{kg}) = \mathbf{0.209 \text{ m}^3/\text{s}}$$

5-57 Air is compressed by a compressor. The mass flow rate of air through the compressor is to be determined.

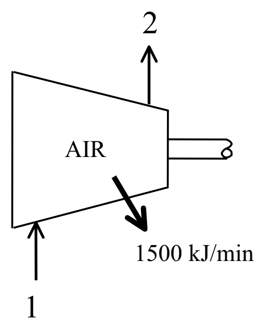
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The inlet and exit enthalpies of air are (Table A-17)

$$T_1 = 25°\text{C} = 298 \text{ K} \quad \rightarrow \quad h_1 = \text{h}_{@\,298\text{ K}} = 298.2 \text{ kJ/kg}$$

$$T_2 = 347°\text{C} = 620 \text{ K} \quad \rightarrow \quad h_2 = \text{h}_{@\,620\text{ K}} = 628.07 \text{ kJ/kg}$$

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



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

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

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

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

Substituting, the mass flow rate is determined to be

$$250 \text{ kJ/s - (1500/60 kJ/s)} = \dot{m}\left[628.07 - 298.2 + \frac{(90 \text{ m/s})^2 - 0}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right] \rightarrow \dot{m} = \mathbf{0.674 \text{ kg/s}}$$

5-58E Air is compressed by a compressor. The mass flow rate of air through the compressor and the exit temperature of air are to be determined.

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

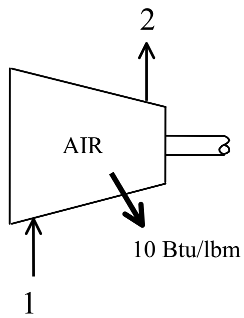
Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1E). The inlet enthalpy of air is (Table A-17E)

$$T_1 = 60°\text{F} = 520 \text{ R} \quad \rightarrow \quad h_1 = h_{@\,520\text{ R}} = 124.27 \text{ Btu/lbm}$$

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 mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520 \text{ R})}{14.7 \text{ psia}} = 13.1 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{5000 \text{ ft}^3/\text{min}}{13.1 \text{ ft}^3/\text{lbm}} = 381.7 \text{ lbm/min} = \mathbf{6.36 \text{ lbm/s}}$$



(*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

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

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

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

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

Substituting,

$$(700 \text{ hp})\left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) - (6.36 \text{ lbm/s}) \times (10 \text{ Btu/lbm}) = (6.36 \text{ lbm/s})(h_2 - 124.27 \text{ Btu/lbm})$$

$$h_2 = 192.06 \text{ Btu/lbm}$$

Then the exit temperature is determined from Table A-17E to be

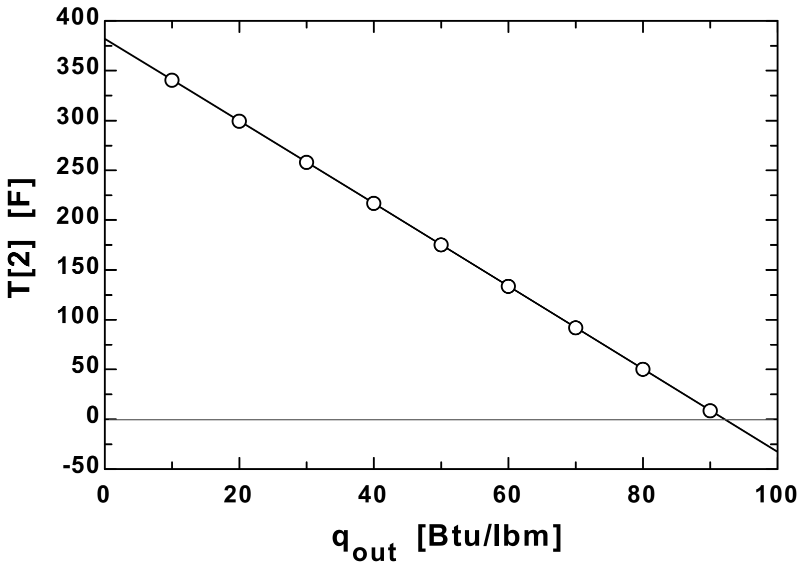
$$T_2 = 801 \text{ R} = \mathbf{341°F}$$

5-59E EES Problem 5-58E is reconsidered. The effect of the rate of cooling of the compressor on the exit temperature of air as the cooling rate varies from 0 to 100 Btu/lbm is to be investigated. The air exit temperature is to be plotted against the rate of cooling.

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

```
"Knowns "
T[1] = 60  [F]
P[1] = 14.7  [psia]
V_dot[1] = 5000  [ft^3/min]
P[2] = 150  [psia]
{q_out=10 [Btu/lbm]}
W_dot_in=700 [hp]

"Property Data"
h[1]=enthalpy(Air,T=T[1])
h[2]=enthalpy(Air,T=T[2])
TR_2=T[2]+460 "[R]"
v[1]=volume(Air,T=T[1],p=P[1])
v[2]=volume(Air,T=T[2],p=P[2])

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

"Mass flow rate"
m_dot[1]=V_dot[1]/v[1] *convert(ft^3/min,ft^3/s)
m_dot[2]= V_dot[2]/v[2]*convert(ft^3/min,ft^3/s)

"Conservation of Energy - Steady Flow energy balance"
W_dot_in*convert(hp,Btu/s)+m_dot[1]*(h[1]) = m_dot[1]*q_out+m_dot[1]*(h[2])
```

q_{out} [Btu/lbm]	T_2 [F]
0	382
10	340.9
20	299.7
30	258.3
40	216.9
50	175.4
60	133.8
70	92.26
80	50.67
90	9.053
100	-32.63



5-60 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).

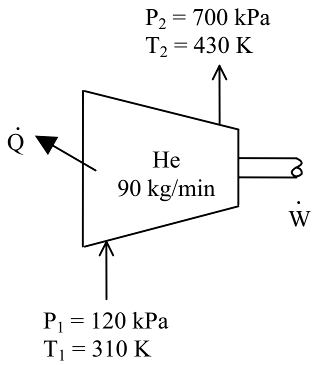
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Thus,

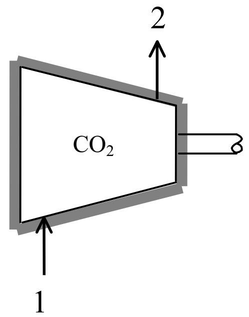
$$\dot{W}_{in} = \dot{Q}_{out} + \dot{m}c_p(T_2 - T_1)$$
$$= (90/60 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg}\cdot\text{K})(430 - 310)\text{K}$$
$$= \mathbf{965 \ kW}$$

5-61 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)

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



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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{100 \text{ kPa}} = 0.5667 \text{ m}^3/\text{kg}$$

$$\dot{V} = \dot{m}v_1 = (0.5 \text{ kg/s})(0.5667 \text{ m}^3/\text{kg}) = \mathbf{0.283 \ m^3/s}$$

(*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

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

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

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

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \dot{m}(\bar{h}_2 - \bar{h}_1)/M$$

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

Throttling Valves

5-62C Because usually there is a large temperature drop associated with the throttling process.

5-63C Yes.

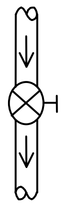
5-64C No. Because air is an ideal gas and h = h(T) for ideal gases. Thus if h remains constant, so does the temperature.

5-65C If it remains in the liquid phase, no. But if some of the liquid vaporizes during throttling, then yes.

5-66 Refrigerant-134a is throttled by a valve. The temperature drop of the refrigerant and specific volume 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),



$$\left.\begin{array}{l} P_1 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} T_1 = T_{\text{sat}} = 26.69^\circ\text{C} \\ h_1 = h_f = 88.82 \text{ kJ/kg} \end{array}$$

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

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \;\rightarrow\; \dot{m}h_1 = \dot{m}h_2 \rightarrow\; h_1 = h_2$$

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

$$\left.\begin{array}{l} P_2 = 160 \text{ kPa} \\ (h_2 = h_1) \end{array}\right\} \begin{array}{l} h_f = 31.21 \text{ kJ/kg}, \quad T_{\text{sat}} = -15.60^\circ\text{C} \\ h_g = 241.11 \text{ kJ/kg} \end{array}$$

Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state and thus $T_2 = T_{sat} = -15.60°C$. Then the temperature drop becomes

$$\Delta T = T_2 - T_1 = -15.60 - 26.69 = \mathbf{-42.3^\circ C}$$

The quality at this state is determined from

$$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{88.82 - 31.21}{209.90} = 0.2745$$

Thus,

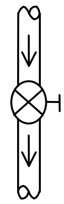
$$v_2 = v_f + x_2 v_{fg} = 0.0007437 + 0.2745 \times (0.12348 - 0.0007437) = \mathbf{0.0344\ m^3/kg}$$

5-67 [*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

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \rightarrow \dot{m}h_1 = \dot{m}h_2 \rightarrow h_1 = h_2$$

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

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

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

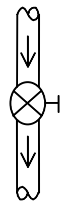
Thus, $$u_2 = u_f + x_2 u_{fg} = 25.39 + 0.2861 \times 193.45 = \mathbf{80.74 \text{ kJ/kg}}$$

5-68 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),

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



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

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \rightarrow \dot{m}h_1 = \dot{m}h_2 \rightarrow h_1 = h_2$$

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

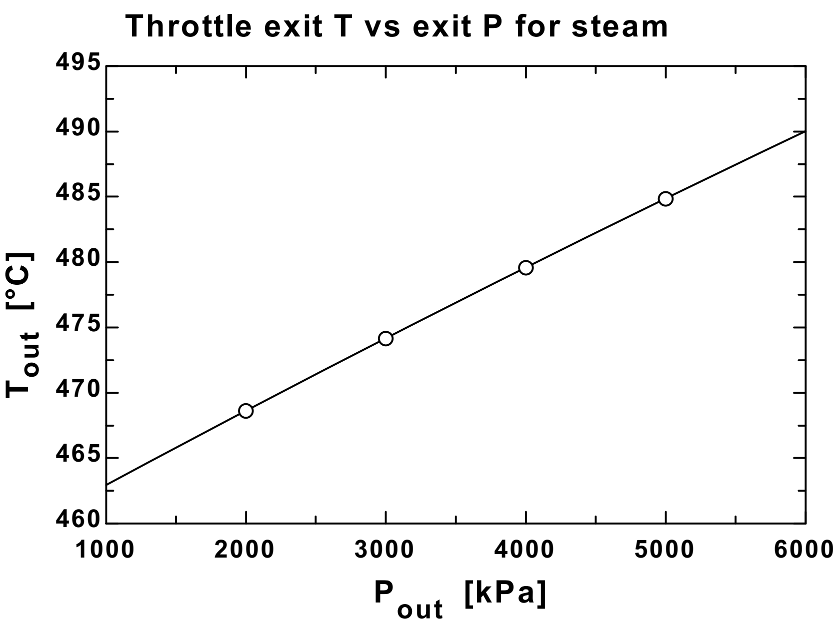
$$\left.\begin{array}{l} P_2 = 6 \text{ MPa} \\ (h_2 = h_1) \end{array}\right\} T_2 = \mathbf{490.1°C}$$

5-69 EES Problem 5-68 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"
```

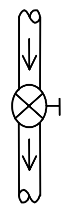
P_{out} [kPa]	T_{out} [C]
1000	463.1
2000	468.8
3000	474.3
4000	479.7
5000	484.9
6000	490.1



5-70E 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



$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\ (steady)} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out} \rightarrow \dot{m}h_1 = \dot{m}h_2 \rightarrow h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. For an ideal gas, $h = h(T)$.

Therefore,

$$T_2 = T_1 = \mathbf{90°F}$$

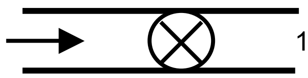
5-71 Carbon dioxide flows through a throttling valve. The temperature change of CO_2 is to be determined if CO_2 is assumed an ideal gas and a real gas.

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

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\ (steady)} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out} \rightarrow \dot{m}h_1 = \dot{m}h_2 \rightarrow h_1 = h_2$$

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



(*a*) For an ideal gas, $h = h(T)$, 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

5-72C Yes, if the mixing chamber is losing heat to the surrounding medium.

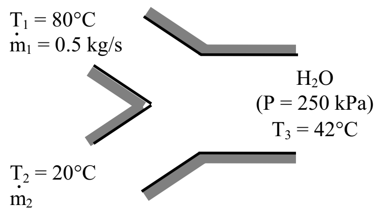
5-73C Under the conditions of no heat and work interactions between the mixing chamber and the surrounding medium.

5-74C Under the conditions of no heat and work interactions between the heat exchanger and the surrounding medium.

5-75 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:

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

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

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

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

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

5-76 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam 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 Noting that $T < T_{sat\ @\ 300\ kPa}$ = 133.52°C, the cold water stream and the mixture exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus, from steam tables (Tables A-4 through A-6)

$$h_1 \cong h_{f@\,20°C} = 83.91 \text{ kJ/kg}$$
$$h_3 \cong h_{f@\,60°C} = 251.18 \text{ kJ/kg}$$

and

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ T_2 = 300°\text{C} \end{array}\right\} h_2 = 3069.6 \text{ kJ/kg}$$

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

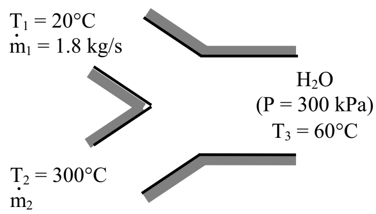
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

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

Solving for $\dot{m}_2$: $\dot{m}_2 = \dfrac{h_1 - h_3}{h_3 - h_2}\dot{m}_1$

Substituting,

$$\dot{m}_2 = \frac{(83.91 - 251.18)\text{kJ/kg}}{(251.18 - 3069.6)\text{kJ/kg}}(1.8 \text{ kg/s}) = \mathbf{0.107 \ kg/s}$$

5-77 Feedwater is heated in a chamber by mixing it with superheated steam. If the mixture is saturated liquid, the ratio of the mass flow rates of the feedwater and the superheated vapor 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 Noting that $T < T_{sat\,@\,1\,MPa} = 179.88°C$, the cold water stream and the mixture exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus, from steam tables (Tables A-4 through A-6)

$$h_1 \cong h_{f@\,50°C} = 209.34 \text{ kJ/kg}$$

$$h_3 \cong h_{f@\,1\text{ MPa}} = 762.51 \text{ kJ/kg}$$

and

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

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

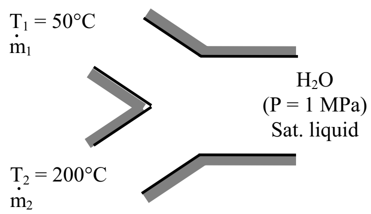
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

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

Dividing by $\dot{m}_2$ yields $y h_1 + h_2 = (y + 1) h_3$

Solving for y: $$y = \frac{h_3 - h_2}{h_1 - h_3}$$

where $y = \dot{m}_1 / \dot{m}_2$ is the desired mass flow rate ratio. Substituting,

$$y = \frac{762.51 - 2828.3}{209.34 - 762.51} = \mathbf{3.73}$$

5-78E 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

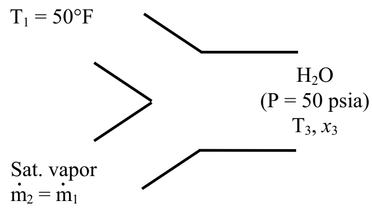
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 2\dot{m} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}$

Energy balance:

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

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

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



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

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

5-79 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),

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

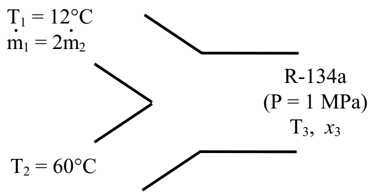
$$h_2 = h_{@ 1 \text{ MPa}, \ 60°C} = 293.38 \text{ kJ/kg}$$

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: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2$ 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,

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

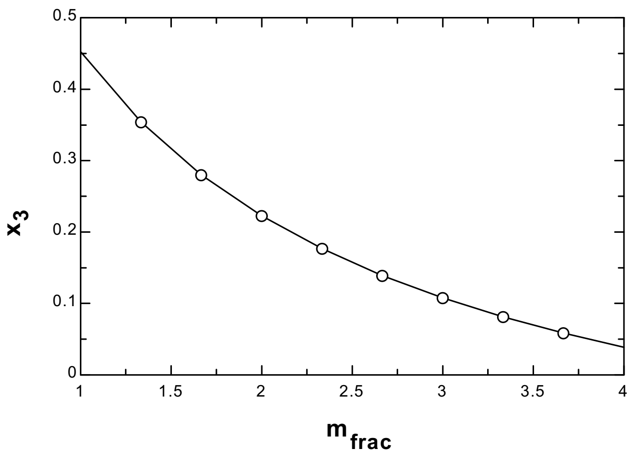
and

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

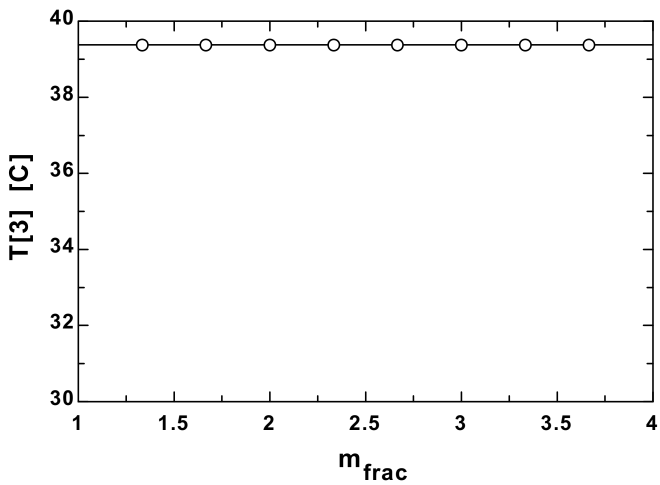
5-80 EES Problem 5-79 is reconsidered. The effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream as the ratio of the mass flow rate of the cold stream to that of the hot stream varies from 1 to 4 is to be investigated. The mixture temperature 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



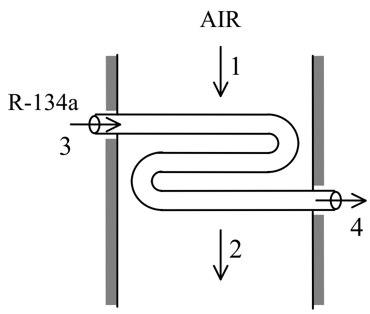
5-81 Refrigerant-134a is to be cooled by air in the condenser. For a specified volume flow rate of air, the mass flow rate of the refrigerant is to be determined.

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 is c_p = 1.005 kJ/kg·°C (Table A-2). The enthalpies of the R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_3 = 1\text{ MPa} \\ T_3 = 90°\text{C} \end{array}\right\} h_3 = 324.64\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 1\text{ MPa} \\ T_4 = 30°\text{C} \end{array}\right\} h_4 \cong h_{f@30°C} = 93.58\text{ kJ/kg}$$



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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{100\text{ kPa}} = 0.861\text{ m}^3/\text{kg}$$

and

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{600\text{ m}^3/\text{min}}{0.861\text{ m}^3/\text{kg}} = 696.9\text{ kg/min}$$

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

Mass balance (*for each fluid stream*):

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

Energy balance (for the entire heat exchanger):

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

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

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

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

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

Substituting,

$$\dot{m}_R = \frac{(1.005\text{ kJ/kg}\cdot°\text{C})(60-27)°\text{C}}{(324.64 - 93.58)\text{ kJ/kg}}(696.9\text{ kg/min}) = \mathbf{100.0\ \ kg/min}$$

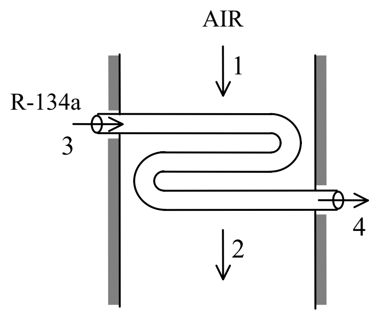
5-82E Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of the air and the rate of heat transfer from the air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work 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 the R-134a at the inlet and the exit states are (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_3 = 20 \text{ psia} \\ x_3 = 0.3 \end{array}\right\} h_3 = h_f + x_3 h_{fg} = 11.445 + 0.3 \times 91.282 = 38.83 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ psia} \\ \text{sat.vapor} \end{array}\right\} h_4 = h_{g@20 \text{ psia}} = 102.73 \text{ Btu/lbm}$$



Analysis (*a*) The inlet specific volume and the mass flow rate of air are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(550 \text{ R})}{14.7 \text{ psia}} = 13.86 \text{ ft}^3/\text{lbm}$$

and

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{200 \text{ ft}^3/\text{min}}{13.86 \text{ ft}^3/\text{lbm}} = 14.43 \text{ lbm/min}$$

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*):

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

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, $\dot{m}_R(h_3 - h_4) = \dot{m}_a(h_2 - h_1) = \dot{m}_a c_p(T_2 - T_1)$

Solving for T_2: $T_2 = T_1 + \dfrac{\dot{m}_R(h_3 - h_4)}{\dot{m}_a c_p}$

Substituting, $T_2 = 90°\text{F} + \dfrac{(4 \text{ lbm/min})(38.83 - 102.73)\text{Btu/lbm}}{(14.43 \text{ Btu/min})(0.24 \text{ Btu/lbm}\cdot°\text{F})} = \mathbf{16.2°F}$

(*b*) The rate of heat transfer from the air to the refrigerant is determined from the steady-flow energy balance applied to the air only. It yields

$$-\dot{Q}_{\text{air, out}} = \dot{m}_a(h_2 - h_1) = \dot{m}_a c_p(T_2 - T_1)$$

$$\dot{Q}_{\text{air, out}} = -(14.43 \text{ lbmg/min})(0.24 \text{ Btu/lbm}\cdot°\text{F})(16.2 - 90)°\text{F} = \mathbf{255.6\ Btu/min}$$

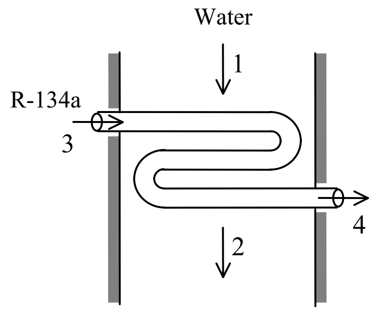
5-83 Refrigerant-134a is condensed in a water-cooled condenser. The mass flow rate of the cooling water required is to be determined.

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,

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

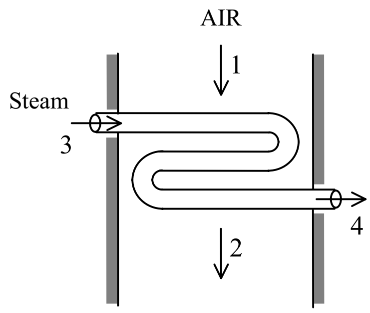
5-84E [*Also solved by EES on enclosed CD***]** Air is heated in a steam heating system. For specified flow rates, the volume flow rate of air at the inlet is to be determined.

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*):

$$\dot m_{in} - \dot m_{out} = \Delta \dot m_{system}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot m_{in} = \dot m_{out} \rightarrow \dot m_1 = \dot m_2 = \dot m_a \text{ and } \dot m_3 = \dot m_4 = \dot m_s$$

Energy balance (for the entire heat exchanger):

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

$$\dot E_{in} = \dot E_{out}$$

$$\dot m_1 h_1 + \dot m_3 h_3 = \dot m_2 h_2 + \dot m_4 h_4 \quad (\text{since } \dot Q = \dot W = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

Combining the two, $\dot m_a (h_2 - h_1) = \dot m_s (h_3 - h_4)$

Solving for $\dot m_a$: $\dot m_a = \dfrac{h_3 - h_4}{h_2 - h_1} \dot m_s \cong \dfrac{h_3 - h_4}{c_p (T_2 - T_1)} \dot m_s$

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

Then the volume flow rate of air at the inlet becomes

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

5-85 Steam is condensed by cooling water in the condenser of a power plant. If the temperature rise of the cooling water is not to exceed 10°C, the minimum mass flow rate of the cooling water required is to be determined.

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

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

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

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

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

Mass balance (for each fluid stream):

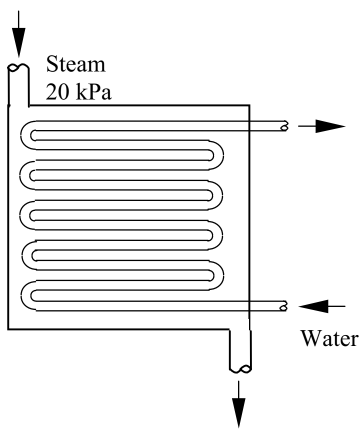
$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_w \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

Energy balance (for the heat exchanger):

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

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

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



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

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

Substituting,

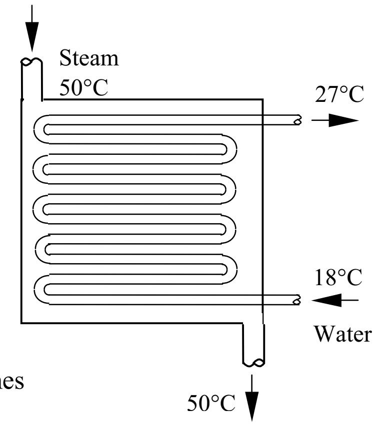
$$\dot{m}_w = \frac{(2491.1 - 251.42)\text{kJ/kg}}{(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(10^\circ\text{C})} (20{,}000/3600 \text{ kg/s}) = \mathbf{297.7 \text{ kg/s}}$$

5-86 Steam is condensed by cooling water in the condenser of a power plant. The rate of condensation of steam 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 heat of vaporization of water at 50°C is h_{fg} = 2382.0 kJ/kg and specific heat of cold water is c_p = 4.18 kJ/kg.°C (Tables A-3 and A-4).

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 heat transfer rate to the cooling water in the condenser becomes

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_{\text{cooling water}} \\ &= (101\text{ kg/s})(4.18\text{ kJ/kg.°C})(27°\text{C} - 18°\text{C}) \\ &= 3800\text{ kJ/s}\end{aligned}$$

The rate of condensation of steam is determined to be

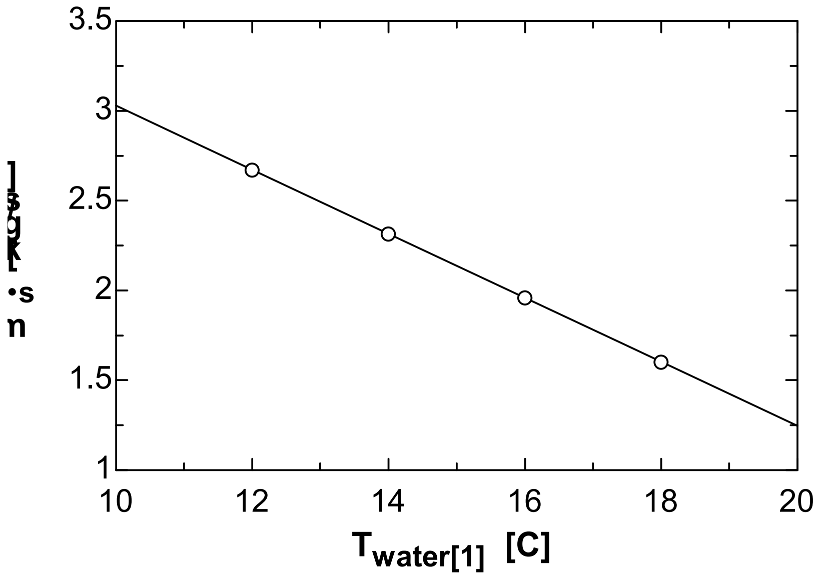
$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{3800\text{ kJ/s}}{2382.0\text{ kJ/kg}} = \mathbf{1.60\ kg/s}$$

5-87 EES Problem 5-86 is reconsidered. The effect of the inlet temperature of cooling water on the rate of condensation of steam as the inlet temperature varies from 10°C to 20°C at constant exit temperature is to be investigated. The rate of condensation of steam is to be plotted against the inlet temperature of the cooling water.

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

```
"Input Data"
T_s[1]=50 [C]
T_s[2]=50 [C]
m_dot_water=101 [kg/s]
T_water[1]=18 [C]
T_water[2]=27 [C]
C_P_water = 4.20 [kJ/kg-°C]

"Conservation of mass for the steam: m_dot_s_in=m_dot_s_out=m_dot_s"
"Conservation of mass for the water: m_dot_water_in=lm_dot_water_out=m_dot_water"
"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_s*h_s[1] + m_dot_water*h_water[1]
E_dot_out=m_dot_s*h_s[2] + m_dot_water*h_water[2]

"Property data are given by:"
h_s[1] =enthalpy(steam_iapws,T=T_s[1],x=1) "steam data"
h_s[2] =enthalpy(steam_iapws,T=T_s[2],x=0)
h_water[1] =C_P_water*T_water[1] "water data"
h_water[2] =C_P_water*T_water[2]

h_fg_s=h_s[1]-h_s[2] "h_fg is found from the EES functions rather than using h_fg = 2305 kJ/kg"
```

m_s [kg/s]	$T_{water,1}$ [C]
3.028	10
2.671	12
2.315	14
1.959	16
1.603	18
1.247	20

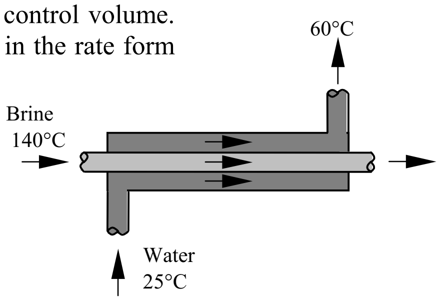


5-88 Water is heated in a heat exchanger by geothermal water. The rate of heat transfer to the water and the exit temperature of the geothermal water is to be determined.

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 geothermal fluid are given to be 4.18 and 4.31 kJ/kg.°C, respectively.

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



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

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

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

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

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

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (0.2\text{ kg/s})(4.18\text{ kJ/kg.°C})(60°\text{C} - 25°\text{C}) = \mathbf{29.26\ kW}$$

Noting that heat transfer to the cold water is equal to the heat loss from the geothermal water, the outlet temperature of the geothermal water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{geot.water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 140°\text{C} - \frac{29.26\text{ kW}}{(0.3\text{ kg/s})(4.31\text{ kJ/kg.°C})} = \mathbf{117.4°C}$$

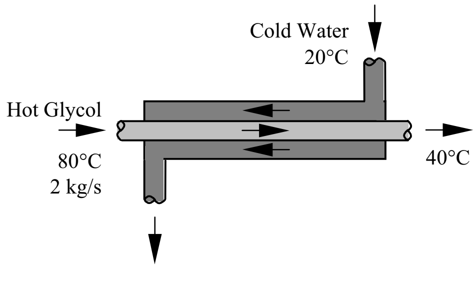
5-89 Ethylene glycol is cooled by water in a heat exchanger. The rate of heat transfer in the heat exchanger and the mass flow rate of water are to be determined.

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

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

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

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



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

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

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

Then the rate of heat transfer becomes

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

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

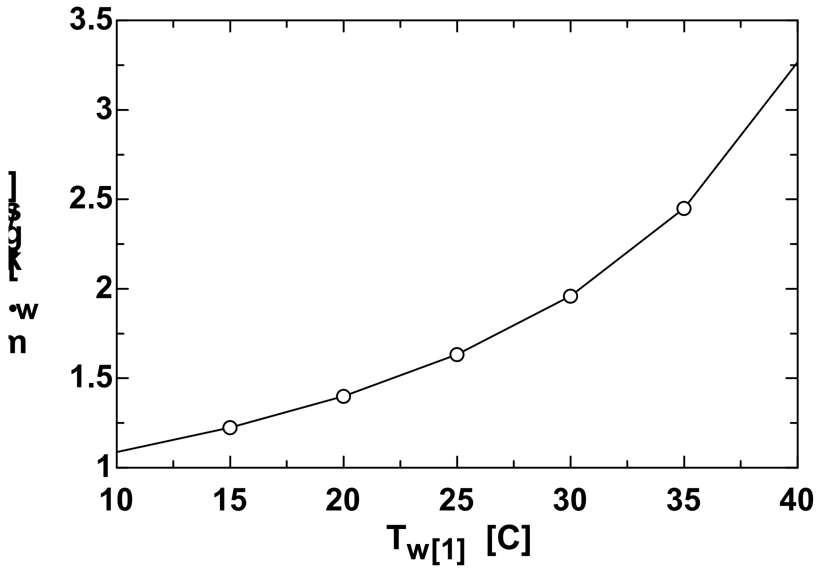
$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{204.8\ \text{kJ/s}}{(4.18\ \text{kJ/kg.°C})(55°\text{C} - 20°\text{C})} = \mathbf{1.4\ kg/s}$$

5-90 EES Problem 5-89 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

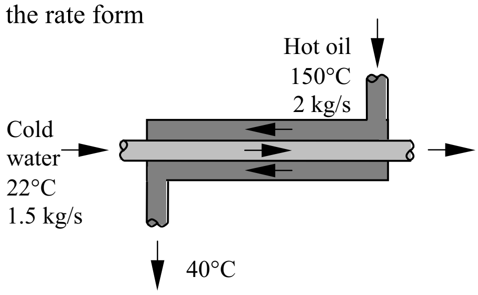


5-91 Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer in the heat exchanger and the exit temperature of water is to be determined.

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

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

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

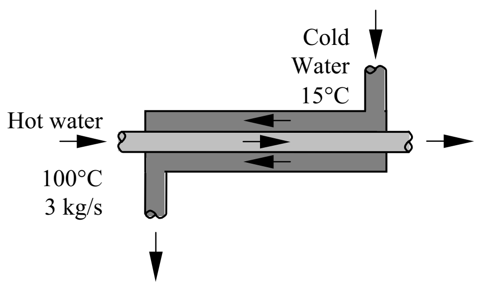
$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}_{water}c_p} = 22°\text{C} + \frac{484 \text{ kJ/s}}{(1.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})} = \mathbf{99.2°C}$$

5-92 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



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

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

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

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

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

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

5-93 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.

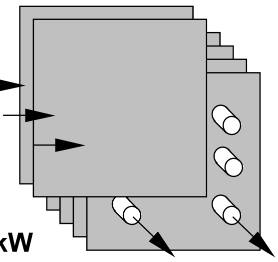
Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively.

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

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

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

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

5-94 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.

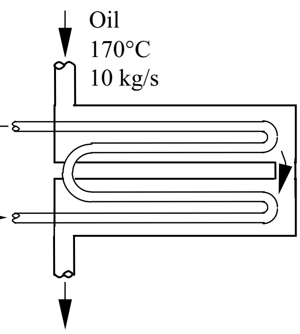
Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

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

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

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

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



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

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

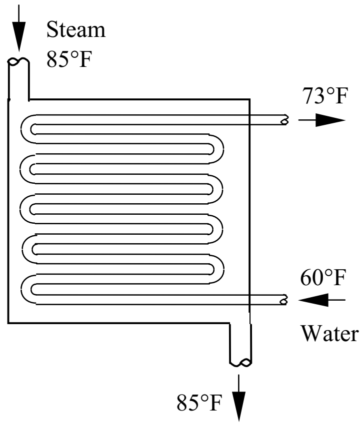
Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

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

5-95E 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

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

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

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

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

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

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

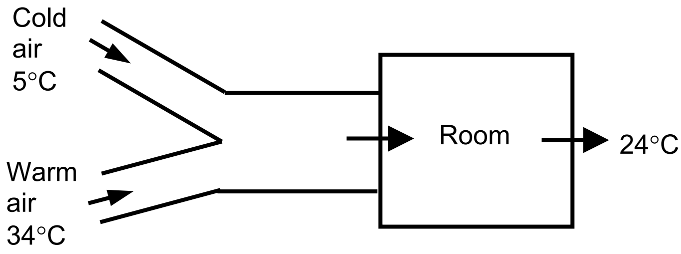
Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

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

5-96 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 $\dot{m}_1 h_1 + 1.6\dot{m}_1 h_2 = 2.6\dot{m}_1 h_3 \;\; or \;\; h_3 = \left(h_1 + 1.6 h_2\right)/2.6$

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

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

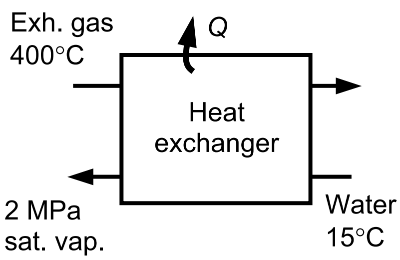
5-97 A heat exchanger that is not insulated is used to produce steam from the heat given up by the exhaust gases of an internal combustion engine. The temperature of exhaust gases at the heat exchanger exit and the rate of heat transfer to the water are to be determined.

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

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

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

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

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

The heat loss is

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

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

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

Pipe and duct Flow

5-98 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

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \rightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.341 \text{ m}^3\text{/min})}{\pi (110 \text{ m/min})}} = 0.063 \text{ m} = \mathbf{6.3 \text{ cm}}$$

5-99 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

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

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

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

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

Then the required mass flow rate of air to absorb heat at a rate of 100 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{100 \text{ W}}{(1006.5 \text{ J/kg.°C})(60 - 45)\text{°C}}$$
$$= 0.006624 \text{ kg/s} = 0.397 \text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(60 + 273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.57 \text{ m}^3/\text{min}}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

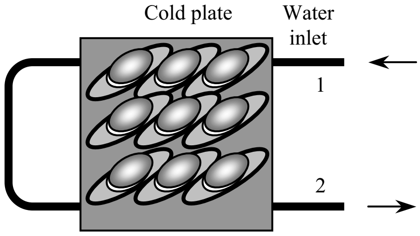
$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.57 \text{ m}^3/\text{min})}{\pi(110 \text{ m/min})}} = 0.081 \text{ m} = \mathbf{8.1 \text{ cm}}$$

5-100E 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



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

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

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

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

Then mass flow rate of water and the rate of heat removal by the water are determined to be

$$\dot{m} = \rho A V = \rho \frac{\pi D^2}{4} V = (62.1\,\text{lbm/ft}^3)\frac{\pi (0.25/12\,\text{ft})^2}{4}(60\,\text{ft/min}) = 1.270\,\text{lbm/min} = 76.2\,\text{lbm/h}$$

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (76.2\,\text{lbm/h})(1.00\,\text{Btu/lbm.°F})(105 - 95)\text{°F} = 762\,\text{Btu/h}$$

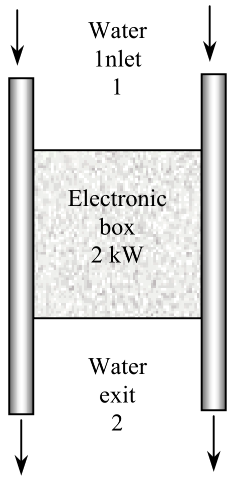
which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

$$\dot{Q} = \frac{762\,\text{Btu/h}}{0.85} = \mathbf{896\,Btu/h = 263\,W}$$

5-101 A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water. **3** Water is an incompressible substance with constant specific heats at room temperature. **4** Kinetic and potential energy changes are negligible.

Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg.°C (Table A-3).



Analysis We take the water channels on the sides 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

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

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

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

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

Then the mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}c_p \Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p \Delta T} = \frac{2\,\text{kJ/s}}{(4.18\,\text{kJ/kg.°C})(4\text{°C})} = \mathbf{0.1196\,kg/s}$$

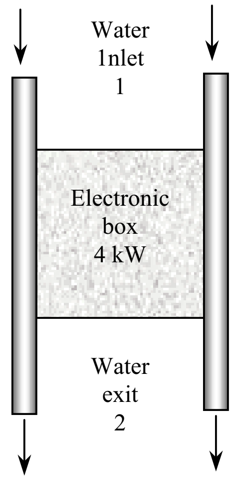
Therefore, 0.1196 kg of water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.1196\,\text{kg/s})(365\,\text{days/yr} \times 24\,\text{h/day} \times 3600\,\text{s/h}) = 3{,}772{,}000\,\text{kg/yr} = \mathbf{3{,}772\,tons/yr}$$

5-102 A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water. **3** Water is an incompressible substance with constant specific heats at room temperature. **4** Kinetic and potential energy changes are negligible

Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg.°C (Table A-3).



Analysis We take the water channels on the sides 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 mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}c_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p\Delta T} = \frac{4\ \text{kJ/s}}{(4.18\ \text{kJ/kg.°C})(4°\text{C})} = \mathbf{0.2392\ kg/s}$$

Therefore, 0.2392 kg of water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.23923\ \text{kg/s})(365\ \text{days/yr} \times 24\ \text{h/day} \times 3600\ \text{s/h}) = 7{,}544{,}400\ \text{kg/yr} = \mathbf{7544\ tons/yr}$$

5-103 A long roll of large 1-Mn manganese steel plate is to be quenched in an oil bath at a specified rate. The rate at which heat needs to be removed from the oil to keep its temperature constant is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the roll are constant. **3** Kinetic and potential energy changes are negligible

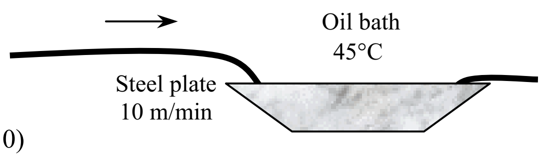
Properties The properties of the steel plate are given to be ρ = 7854 kg/m^3 and c_p = 0.434 kJ/kg.°C.

Analysis The mass flow rate of the sheet metal through the oil bath is

$$\dot{m} = \rho\dot{V} = \rho wtV = (7854\ \text{kg/m}^3)(2\ \text{m})(0.005\ \text{m})(10\ \text{m/min}) = 785.4\ \text{kg/min}$$

We take the volume occupied by the sheet metal in the oil bath to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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



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

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

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

Then the rate of heat transfer from the sheet metal to the oil bath becomes

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_{\text{in}} - T_{\text{out}})_{\text{metal}} = (785.4\ \text{kg/min})(0.434\ \text{kJ/kg.°C})(820 - 51.1)°\text{C} = 262{,}090\ \text{kJ/min} = \mathbf{4368\ \ kW}$$

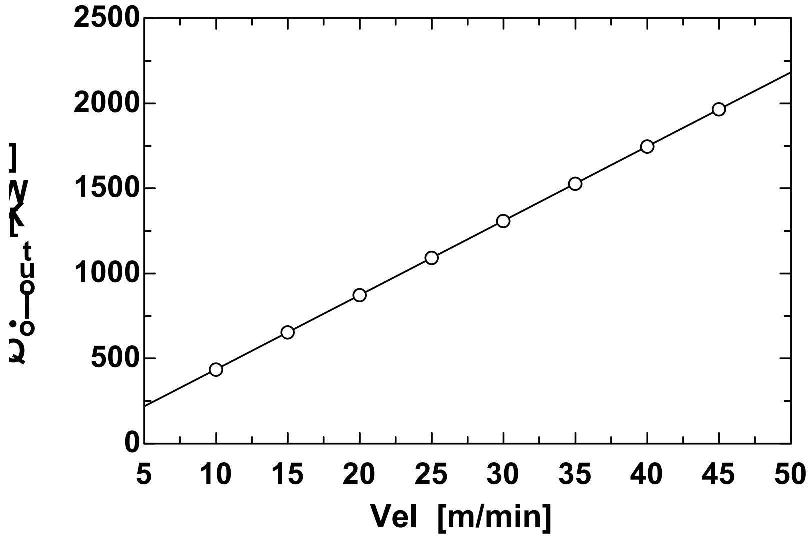
This is the rate of heat transfer from the metal sheet to the oil, which is equal to the rate of heat removal from the oil since the oil temperature is maintained constant.

5-104 EES Problem 5-103 is reconsidered. The effect of the moving velocity of the steel plate on the rate of heat transfer from the oil bath as the velocity varies from 5 to 50 m/min is to be investigated. Tate of heat transfer is to be plotted against the plate velocity.

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

```
"Knowns"
Vel = 10 [m/min]
T_bath = 45 [C]
T_1 = 820 [C]
T_2 = 51.1 [C]
rho = 785 [kg/m^3]
C_P = 0.434 [kJ/kg-C]
width = 2 [m]
thick = 0.5 [cm]

"Analysis:
 The mass flow rate of the sheet metal through the oil bath is:"
Vol_dot = width*thick*convert(cm,m)*Vel/convert(min,s)
m_dot = rho*Vol_dot

"We take the volume occupied by the sheet metal in the oil bath to be the system,
which is a control volume. The energy balance for this steady-flow system--the metal can be
expressed  in the rate form as:"

E_dot_metal_in = E_dot_metal_out
E_dot_metal_in=m_dot*h_1
E_dot_metal_out=m_dot*h_2+Q_dot_metal_out

h_1 = C_P*T_1
h_2 = C_P*T_2

Q_dot_oil_out = Q_dot_metal_out
```

Q_{oilout} [kW]	Vel [m/min]
218.3	5
436.6	10
654.9	15
873.2	20
1091	25
1310	30
1528	35
1746	40
1965	45
2183	50



5-105 [*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.

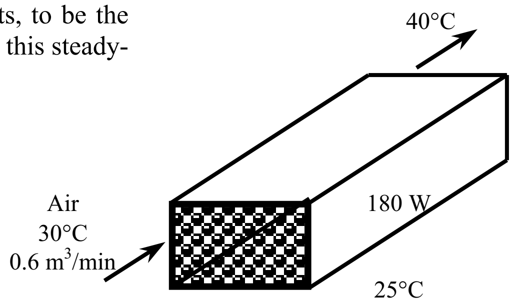
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible

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

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

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

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

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

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,

$$\dot{Q}_{external} = \dot{Q}_{total} - \dot{Q}_{internal} = 180 - 117 = \mathbf{63\ W}$$

5-106 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

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\text{ kg/m}^3$$

$$\dot{m} = \rho \dot{V} = (1.165\text{ kg/m}^3)(0.6\text{ m}^3\text{ / min}) = 0.700\text{ kg/min}$$

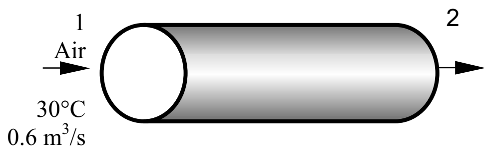
We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



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,

$$\dot{Q}_{external} = \dot{Q}_{total} - \dot{Q}_{internal} = 180 - 117 = \mathbf{63\ W}$$

5-107E 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).

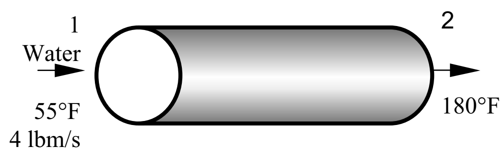
Analysis We take the thin aluminum tube to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the total rate of heat transfer to the water flowing through the tube becomes

$$\dot{Q}_{total} = \dot{m}c_p(T_e - T_i) = (4\text{ lbm/s})(1.00\text{ Btu/lbm.°F})(180-55)\text{°F} = 500\text{ Btu/s} = 1{,}800{,}000\text{ Btu/h}$$

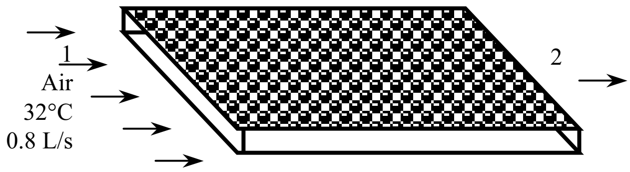
The length of the tube required is

$$L = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{1{,}800{,}000\text{ Btu/h}}{400\text{ Btu/h.ft}} = \mathbf{4500\ ft}$$

5-108 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

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1) \to T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}c_p} = 32\ °\text{C} + \frac{20\ \text{J/s}}{(0.000928\ \text{kg/s})(1005\ \text{J/kg.°C})} = \mathbf{53.4°C}$$

5-109 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

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

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

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

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

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

$$\dot{Q} = \dot{m}c_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{25\text{ W}}{(0.0104\text{ kg/s})(1005\text{ J/kg.°C})} = 2.4\text{°C}$$

$$f = \frac{2.4\text{°C}}{10\text{°C}} = 0.24 = \mathbf{24\%}$$

5-110 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.

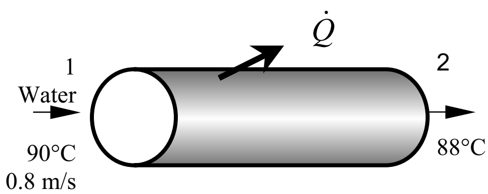
Properties The properties of water at the average temperature of (90+88)/2 = 89°C are $\rho = 965\text{ kg/m}^3$ and c_p = 4.21 kJ/kg.°C (Table A-3).

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

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



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

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

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

Then the rate of heat transfer from the hot water to the surrounding air becomes

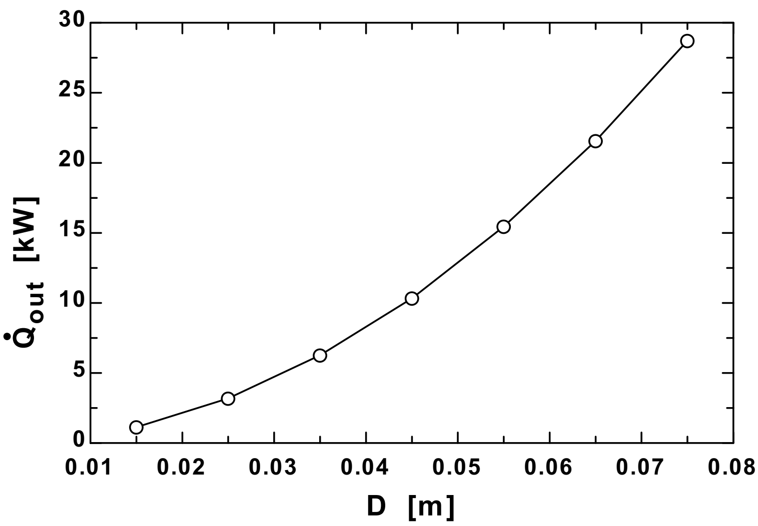
$$\dot{Q}_{out} = \dot{m}c_p[T_{in} - T_{out}]_{water} = (0.970\text{ kg/s})(4.21\text{ kJ/kg.°C})(90 - 88)\text{°C} = \mathbf{8.17\ kW}$$

5-111 EES Problem 5-110 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



5-112 A room is to be heated by an electric resistance heater placed in a duct in the room. The power rating of the electric heater and the temperature rise of air as it passes through the heater are to be determined.

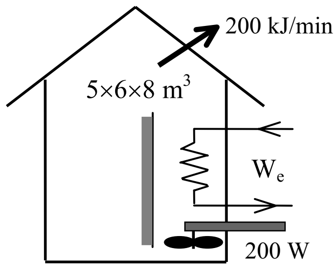
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the room.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heats of air at room temperature are c_p = 1.005 and c_v = 0.718 kJ/kg·K (Table A-2).

Analysis (*a*) The total mass of air in the room is

$$V = 5\times 6\times 8\ \text{m}^3 = 240\ \text{m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(98\ \text{kPa})(240\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})} = 284.6\ \text{kg}$$



We first take the *entire room* as our system, which is a closed system since no mass leaks in or out. The power rating of the electric heater is determined by applying the conservation of energy relation to this constant volume closed system:

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

$$W_{\text{e,in}} + W_{\text{fan,in}} - Q_{out} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}\right) = mc_{v,\text{avg}}\left(T_2 - T_1\right)$$

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,

$$\Delta T = T_2 - T_1 = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{\dot{m}c_p} = \frac{(5.40 + 0.2)\ \text{kJ/s}}{\left(50/60\ \text{kg/s}\right)\left(1.005\ \text{kJ/kg}\cdot\text{K}\right)} = \mathbf{6.7^\circ C}$$

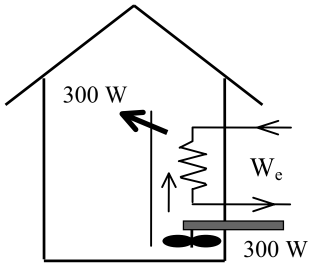
5-113 A house is heated by an electric resistance heater placed in a duct. The power rating of the electric heater is to be determined.

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

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



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

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

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

Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{e,in} = \dot{Q}_{out} + \dot{m}c_p\Delta T - \dot{W}_{fan,in} = (0.3 \text{ kJ/s}) + (0.6 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot{}^\circ\text{C})(7^\circ\text{C}) - 0.3 \text{ kW} = \mathbf{4.22 \text{ kW}}$$

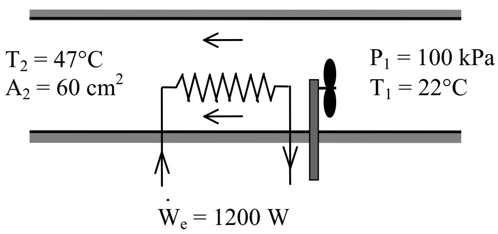
5-114 A hair dryer consumes 1200 W of electric power when running. The inlet volume flow rate and the exit velocity of air are to be determined.

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

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



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

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

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

Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{e,in}}{c_p(T_2 - T_1)} = \frac{1.2 \text{ kJ/s}}{(1.005 \text{ kJ/kg}\cdot{}^\circ\text{C})(47-22)^\circ\text{C}} = 0.04776 \text{ kg/s}$$

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})}{(100 \text{ kPa})} = 0.8467 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}v_1 = (0.04776 \text{ kg/s})(0.8467 \text{ m}^3/\text{kg}) = \mathbf{0.0404 \text{ m}^3/\text{s}}$$

(*b*) The exit velocity of air is determined from the mass balance $\dot{m}_1 = \dot{m}_2 = \dot{m}$ to be

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(320\text{K})}{(100 \text{ kPa})} = 0.9184 \text{ m}^3/\text{kg}$$

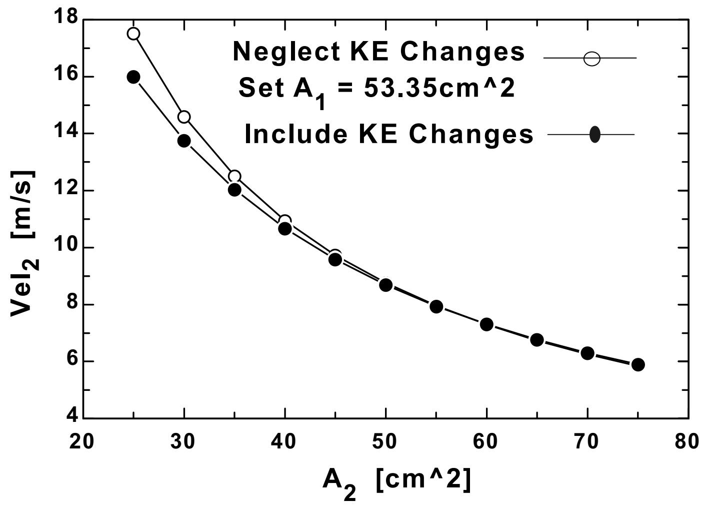
$$\dot{m} = \frac{1}{v_2}A_2V_2 \longrightarrow V_2 = \frac{\dot{m}v_2}{A_2} = \frac{(0.04776 \text{ kg/s})(0.9184 \text{ m}^3/\text{kg})}{60\times10^{-4} \text{ m}^2} = \mathbf{7.31 \text{ m/s}}$$

5-115 EES Problem 5-114 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

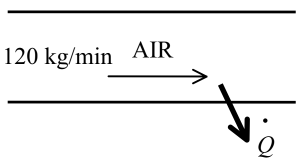


5-116 The ducts of a heating system pass through an unheated area. The rate of heat loss from the air in the ducts is to be determined.

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



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

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

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

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

Substituting, $\dot{Q}_{out} = (120 \text{ kg/min})(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(4^\circ\text{C}) = \mathbf{482\ kJ/min}$

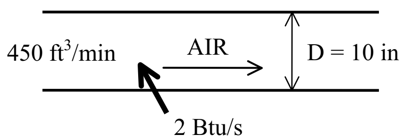
5-117E The ducts of an air-conditioning system pass through an unconditioned area. The inlet velocity and the exit temperature of air are to be determined.

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** There are no work interactions involved.

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.R (Table A-2E)

Analysis (*a*) The inlet velocity of air through the duct is

$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi r^2} = \frac{450 \text{ ft}^3\text{/min}}{\pi (5/12 \text{ ft})^2} = \mathbf{825\ ft/min}$$



Then the mass flow rate of air becomes

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3\text{/lbm}\cdot\text{R})(510 \text{ R})}{(15 \text{ psia})} = 12.6 \text{ ft}^3\text{/lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{450 \text{ ft}^3\text{/min}}{12.6 \text{ ft}^3\text{/lbm}} = 35.7 \text{ lbm/min} = 0.595 \text{ lbm/s}$$

(*b*) We take the *air-conditioning 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

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

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

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

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

Then the exit temperature of air becomes

$$T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}c_p} = 50^\circ\text{F} + \frac{2 \text{ Btu/s}}{(0.595 \text{ lbm/s})(0.24 \text{ Btu/lbm}\cdot^\circ\text{F})} = \mathbf{64.0^\circ F}$$

5-118 Water is heated by a 7-kW resistance heater as it flows through an insulated tube. The mass flow rate of water is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Water is an incompressible substance with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The tube is adiabatic and thus heat losses are negligible.

Properties The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-3).

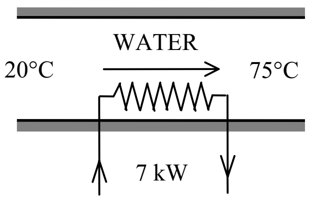
Analysis We take the *water pipe* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$



Substituting, the mass flow rates of water is determined to be

$$\dot{m} = \frac{\dot{W}_{e,in}}{c(T_2 - T_1)} = \frac{7 \text{ kJ/s}}{(4.184 \text{ kJ/kg}\cdot^\circ\text{C})(75 - 20)^\circ\text{C}} = \mathbf{0.0304 \text{ kg/s}}$$

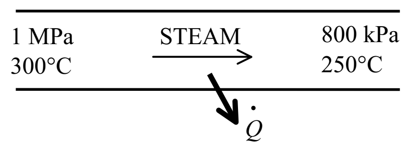
5-119 Steam pipes pass through an unheated area, and the temperature of steam drops as a result of heat losses. The mass flow rate of steam and the rate of heat loss from are to be determined.

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),

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 300^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.25799 \text{ m}^3/\text{kg} \\ h_1 = 3051.6 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 250^\circ\text{C} \end{array}\right\} h_2 = 2950.4 \text{ kJ/kg}$$



Analysis (*a*) The mass flow rate of steam is determined directly from

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{0.25799 \text{ m}^3/\text{kg}} \left[\pi (0.06 \text{ m})^2\right](2 \text{ m/s}) = \mathbf{0.0877 \text{ kg/s}}$$

(*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

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

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

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

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

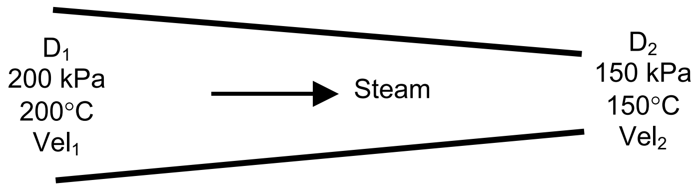
Substituting, the rate of heat loss is determined to be

$$\dot{Q}_{loss} = (0.0877 \text{ kg/s})(3051.6 - 2950.4) \text{ kJ/kg} = \mathbf{8.87 \text{ kJ/s}}$$

5-120 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:

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

$$\dot{m}_{in} = \dot{m}_{out} \longrightarrow A_1 \frac{V_1}{v_1} = A_1 \frac{V_1}{v_1} \longrightarrow \frac{\pi D_1^2}{4}\frac{V_1}{v_1} = \frac{\pi D_2^2}{4}\frac{V_2}{v_2}$$

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)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.0805 \text{ m}^3/\text{kg} \\ h_1 = 2870.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 1.2855 \text{ m}^3/\text{kg} \\ h_2 = 2772.9 \text{ kJ/kg} \end{array}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting,

$$\frac{\pi(1.8\text{ m})^2}{4}\frac{V_1}{(1.0805\text{ m}^3/\text{kg})} = \frac{\pi(1.0\text{ m})^2}{4}\frac{V_2}{(1.2855\text{ m}^3/\text{kg})} \quad (1)$$

$$2870.7\text{ kJ/kg} + \frac{V_1^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 2772.9\text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \quad (2)$$

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

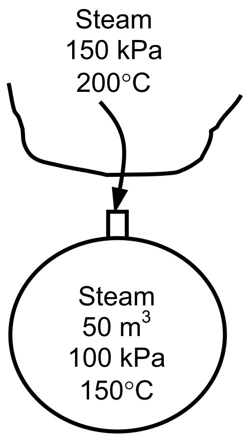
V_1 = **118.8 m/s**

V_2 = **458.0 m/s**

Charging and Discharging Processes

5-121 A large reservoir supplies steam to a balloon whose initial state is specified. The final temperature in the balloon and the boundary work are to be determined.

Analysis Noting that the volume changes linearly with the pressure, the final volume and the initial mass are determined from



$$\left.\begin{array}{l} P_1 = 100\text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} v_1 = 1.9367\text{ m}^3/\text{kg} \quad \text{(Table A-6)}$$

$$V_2 = \frac{P_2}{P_1}V_1 = \frac{150\text{ kPa}}{100\text{ kPa}}(50\text{ m}^3) = 75\text{ m}^3$$

$$m_1 = \frac{V_1}{v_1} = \frac{50\text{ m}^3}{1.9367\text{ m}^3/\text{kg}} = 25.82\text{ kg}$$

The final temperature may be determined if we first calculate specific volume at the final state

$$v_2 = \frac{V_2}{m_2} = \frac{V_2}{2m_1} = \frac{75\text{ m}^3}{2\times(25.82\text{ kg})} = 1.4525\text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 150\text{ kPa} \\ v_2 = 1.4525\text{ m}^3/\text{kg} \end{array}\right\} T_2 = \mathbf{202.5°C} \quad \text{(Table A-6)}$$

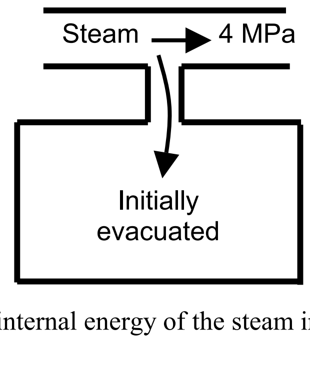
Noting again that the volume changes linearly with the pressure, the boundary work can be determined from

$$W_b = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100+150)\text{kPa}}{2}(75-50)\text{m}^3 = \mathbf{3125\ kJ}$$

5-122 Steam in a supply line is allowed to enter an initially evacuated tank. The temperature of the steam in the supply line and the flow work are to be determined.

Analysis Flow work of the steam in the supply line is converted to sensible internal energy in the tank. That is,

$$h_{\text{line}} = u_{\text{tank}}$$



where $\left.\begin{array}{l} P_{\text{tank}} = 4\text{ MPa} \\ T_{\text{tank}} = 550°\text{C} \end{array}\right\} u_{\text{tank}} = 3189.5\text{ kJ/kg} \quad \text{(Table A-6)}$

Now, the properties of steam in the line can be calculated

$$\left.\begin{array}{l} P_{\text{line}} = 4\text{ MPa} \\ h_{\text{line}} = 3189.5\text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_{\text{line}} = \mathbf{389.5°C} \\ u_{\text{line}} = 2901.5\text{ kJ/kg} \end{array} \quad \text{(Table A-6)}$$

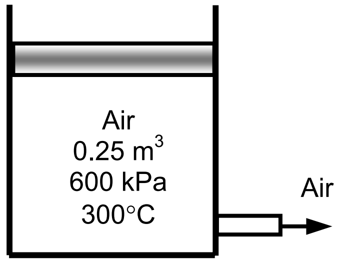
The flow work per unit mass is the difference between enthalpy and internal energy of the steam in the line

$$w_{\text{flow}} = h_{\text{line}} - u_{\text{line}} = 3189.5 - 2901.5 = \mathbf{288\ kJ/kg}$$

5-123 A vertical piston-cylinder device contains air at a specified state. Air is allowed to escape from the cylinder by a valve connected to the cylinder. The final temperature and the boundary work are to be determined.

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1).

Analysis The initial and final masses in the cylinder are



$$m_1 = \frac{P\boldsymbol{V}_1}{RT_1} = \frac{(600\text{ kPa})(0.25\text{ m}^3)}{(0.287\text{ kJ/kg.K})(300+273\text{ K})} = 0.9121\text{ m}^3$$

$$m_2 = 0.25m_1 = 0.25(0.9121\text{ kg}) = 0.2280\text{ kg}$$

Then the final temperature becomes

$$T_2 = \frac{P\boldsymbol{V}_2}{m_2 R} = \frac{(600\text{ kPa})(0.05\text{ m}^3)}{(0.2280\text{ kg})(0.287\text{ kJ/kg.K})} = \mathbf{458.4\ K}$$

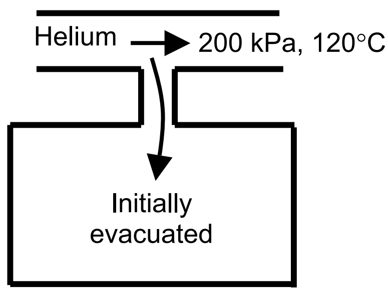
Noting that pressure remains constant during the process, the boundary work is determined from

$$W_b = P(\boldsymbol{V}_1 - \boldsymbol{V}_2) = (600\ \text{kPa})(0.25-0.05)\text{m}^3 = \mathbf{120\ \ kJ}$$

5-124 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



$$\boldsymbol{v} = \frac{RT_{\text{line}}}{P} = \frac{(2.0769\text{ kJ/kg.K})(120+273\text{ K})}{(200\text{ kPa})} = 4.0811\text{ m}^3\text{/kg}$$

$$w_{\text{flow}} = P\boldsymbol{v} = (200\text{ kPa})(4.0811\ \text{ m}^3\text{/kg}) = \mathbf{816.2\ kJ/kg}$$

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.

5-125 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).

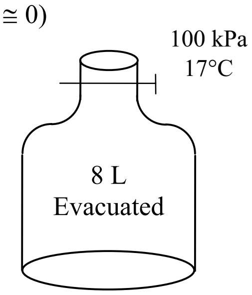
Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_i = m_2 \quad (\text{since } m_{out} = m_{initial} = 0)$

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

$$m_2 = \frac{P_2 \boldsymbol{V}}{RT_2} = \frac{(100 \text{ kPa})(0.008 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})} = 0.0096 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{array}{l} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{array}$$

Substituting, Q_{in} = (0.0096 kg)(206.91 - 290.16) kJ/kg = - 0.8 kJ $\rightarrow$ Q_{out} = **0.8 kJ**

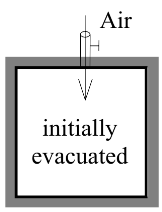
Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

5-126 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: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_i = m_2 \quad (\text{since } m_{out} = m_{initial} = 0)$

Energy balance:

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

$$m_i h_i = m_2 u_2 \quad (\text{since } Q \cong W \cong E_{out} = E_{initial} = ke \cong pe \cong 0)$$

Combining the two balances:

$$u_2 = h_i \rightarrow c_v T_2 = c_p T_i \rightarrow T_2 = (c_p / c_v) T_i = kT_i$$

Substituting, $T_2 = 1.4 \times 290 \text{ K} = 406 \text{ K} = \mathbf{133^\circ C}$

5-127 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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

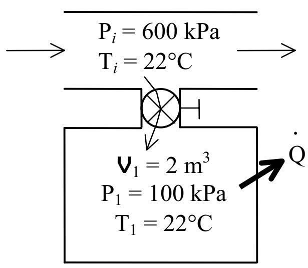
Energy balance: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

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.

5-128 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.

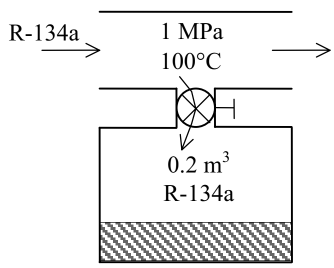
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions 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

$$m_i = m_2 - m_1 = 7.81 - 5.38 = \mathbf{2.43 \text{ kg}}$$

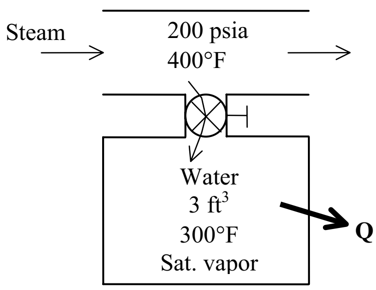
(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(2.43 \text{ kg})(335.06 \text{ kJ/kg}) + (7.81 \text{ kg})(246.79 \text{ kJ/kg}) - (5.38 \text{ kg})(182.92 \text{ kJ/kg}) \\ &= \mathbf{130 \text{ kJ}} \end{aligned}$$

5-129E A rigid tank initially contains saturated water vapor. The tank is connected to a supply line, and water vapor is allowed to enter the tank until one-half of the tank is filled with liquid water. The final pressure in the tank, the mass of steam 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 water are (Tables A-4E through A-6E)



$$\left.\begin{array}{l} T_1 = 300°\text{F} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@300°\text{F}} = 6.4663 \text{ ft}^3/\text{lbm} \\ u_1 = u_{g@300°\text{F}} = 1099.8 \text{ Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} T_2 = 300°F \\ \text{sat. mixture} \end{array}\right\} \begin{array}{l} v_f = 0.01745, \quad v_g = 6.4663 \text{ ft}^3/\text{lbm} \\ u_f = 269.51, \quad u_g = 1099.8 \text{ Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_i = 200 \text{ psia} \\ T_i = 400°\text{F} \end{array}\right\} h_i = 1210.9 \text{ Btu/lbm}$$

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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

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 mixture at the final state at 250°F, and thus the exit pressure is the saturation pressure at this temperature,

$$P_2 = P_{\text{sat@ 300°F}} = \mathbf{67.03 \ psia}$$

(b) The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{3 \text{ ft}^3}{6.4663 \text{ ft}^3/\text{lbm}} = 0.464 \text{ lbm}$$

$$m_2 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{1.5 \text{ ft}^3}{0.01745 \text{ ft}^3/\text{lbm}} + \frac{1.5 \text{ ft}^3.}{6.4663 \text{ ft}^3/\text{lbm}} = 85.97 + 0.232 = 86.20 \text{ lbm}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 86.20 - 0.464 = \mathbf{85.74 \ lbm}$$

(c) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(85.74 \text{ lbm})(1210.9 \text{ Btu/lbm}) + 23{,}425 \text{ Btu} - (0.464 \text{ lbm})(1099.8 \text{ Btu/lbm}) \\ &= -80{,}900 \text{ Btu} \rightarrow Q_{\text{out}} = \mathbf{80{,}900 \ Btu} \end{aligned}$$

since $U_2 = m_2 u_2 = m_f u_f + m_g u_g = 85.97 \times 269.51 + 0.232 \times 1099.8 = 23{,}425 \text{ Btu}$

Discussion A negative result for heat transfer indicates that the assumed direction is wrong, and should be reversed.

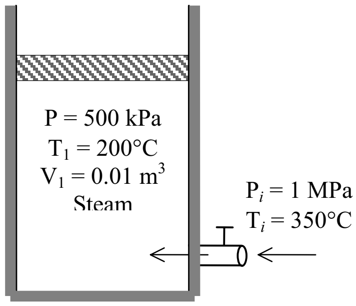
5-130 A cylinder initially contains superheated steam. The cylinder is connected to a supply line, and is superheated steam is allowed to enter the cylinder until the volume doubles at constant pressure. The final temperature in the cylinder and the mass of the steam that entered are to be determined.

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)

$$\left.\begin{array}{l} P_1 = 500 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.42503 \text{ m}^3/\text{kg} \\ u_1 = 2643.3 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1 \text{ MPa} \\ T_i = 350°\text{C} \end{array}\right\} h_i = 3158.2 \text{ kJ/kg}$$



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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

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 $0 = W_{\text{b,out}} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$

The boundary work done during this process is

$$W_{b,out} = \int_1^2 P d\boldsymbol{V} = \text{P}(\boldsymbol{V}_2 - \boldsymbol{V}_1) = (500 \text{ kPa})(0.02 - 0.01)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = 5 \text{ kJ}$$

The initial and the final masses in the cylinder are

$$m_1 = \frac{\boldsymbol{V}_1}{v_1} = \frac{0.01 \text{ m}^3}{0.42503 \text{ m}^3/\text{kg}} = 0.0235 \text{ kg}$$

$$m_2 = \frac{\boldsymbol{V}_2}{v_2} = \frac{0.02 \text{ m}^3}{v_2}$$

Substituting, $$0 = 5 - \left(\frac{0.02}{v_2} - 0.0235 \right)(3158.2) + \frac{0.02}{v_2} u_2 - (0.0235)(2643.3)$$

Then by trial and error (or using EES program), T_2 = **261.7°C** and v_2 = 0.4858 m^3/kg

(*b*) The final mass in the cylinder is

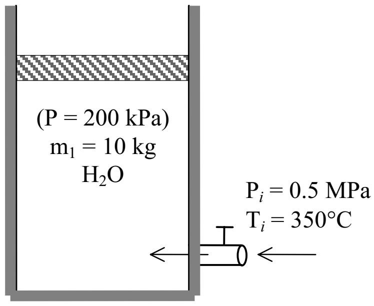
$$m_2 = \frac{\boldsymbol{V}_2}{v_2} = \frac{0.02 \text{ m}^3}{0.4858 \text{ m}^3/\text{kg}} = 0.0412 \text{ kg}$$

Then, $m_i = m_2 - m_1 = 0.0412 - 0.0235 =$ **0.0176 kg**

5-131 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,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3168.1 - 1825.6) \text{ kJ/kg}}{(3168.1 - 2706.3) \text{ kJ/kg}} (10 \text{ kg}) = 29.07 \text{ kg}$$

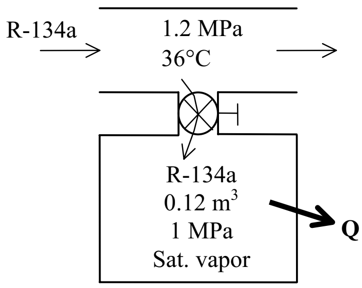
Thus,

$$m_i = m_2 - m_1 = 29.07 - 10 = \mathbf{19.07 \text{ kg}}$$

5-132 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)



$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@1 \text{ MPa}} = 0.02031 \text{ m}^3/\text{kg} \\ u_1 = u_{g@1 \text{ MPa}} = 250.68 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_2 = v_{f@1.2 \text{ MPa}} = 0.0008934 \text{ m}^3/\text{kg} \\ u_2 = u_{f@1.2 \text{ MPa}} = 116.70 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.2 \text{ MPa} \\ T_i = 36°\text{C} \end{array}\right\} h_i = h_{f@36°\text{C}} = 102.30 \text{ kJ/kg}$$

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

$$m_1 = \frac{V_1}{v_1} = \frac{0.12 \text{ m}^3}{0.02031 \text{ m}^3/\text{kg}} = 5.91 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.12 \text{ m}^3}{0.0008934 \text{ m}^3/\text{kg}} = 134.31 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 134.31 - 5.91 = \mathbf{128.4 \text{ kg}}$$

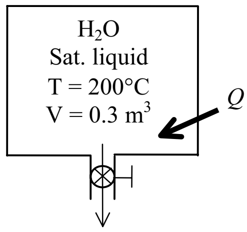
(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(128.4 \text{ kg})(102.30 \text{ kJ/kg}) + (134.31 \text{ kg})(116.70 \text{ kJ/kg}) - (5.91 \text{ kg})(250.68 \text{ kJ/kg}) \\ &= \mathbf{1057 \text{ kJ}} \end{aligned}$$

5-133 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of the mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

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



$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@200°\text{C}} = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = u_{f@200°\text{C}} = 850.46 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_e = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@200°\text{C}} = 852.26 \text{ kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

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

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

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,

$$v_2 = \frac{V}{m_2} = \frac{0.3 \text{ m}^3}{129.7 \text{ kg}} = 0.002313 \text{ m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002313 - 0.001157}{0.12721 - 0.001157} = 0.009171$$

$$\left.\begin{array}{l} T_2 = 200°\text{C} \\ x_2 = 0.009171 \end{array}\right\} u_2 = u_f + x_2 u_{fg} = 850.46 + (0.009171)(1743.7) = 866.46 \text{ kJ/kg}$$

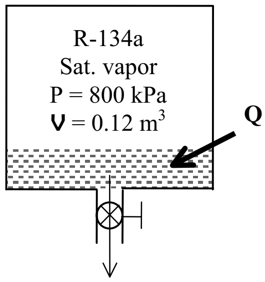
Then the heat transfer during this process is determined from the energy balance by substitution to be

$$Q = (129.7 \text{ kg})(852.26 \text{ kJ/kg}) + (129.7 \text{ kg})(866.46 \text{ kJ/kg}) - (259.4 \text{ kg})(850.46 \text{ kJ/kg})$$
$$= \mathbf{2308 \text{ kJ}}$$

5-134 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)



$$P_1 = 800 \text{ kPa} \rightarrow v_f = 0.0008458 \text{ m}^3/\text{kg}, v_g = 0.025621 \text{ m}^3/\text{kg}$$

$$u_f = 94.79 \text{ kJ/kg}, \quad u_g = 246.79 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@800 \text{ kPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 800 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@800 \text{ kPa}} = 95.47 \text{ kJ/kg}$$

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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

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

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{0.12 \times 0.25 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} + \frac{0.12 \times 0.75 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 35.47 + 3.513 = 38.98 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (35.47)(94.79) + (3.513)(246.79) = 4229.2 \text{ kJ}$$

$$m_2 = \frac{V}{v_2} = \frac{0.12 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 4.684 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 38.98 - 4.684 = 34.30 \text{ kg}$$

$$Q_{\text{in}} = (34.30 \text{ kg})(95.47 \text{ kJ/kg}) + (4.684 \text{ kg})(246.79 \text{ kJ/kg}) - 4229 \text{ kJ} = \mathbf{201.2 \text{ kJ}}$$

5-135E 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.

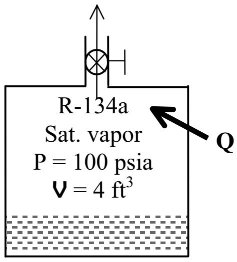
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no 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: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_e = m_1 - m_2$

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

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial 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,

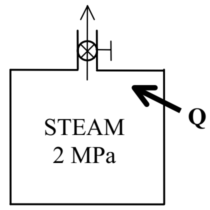
$$m_e = m_1 - m_2 = 66.74 - 8.375 = 58.37 \text{ lbm}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (58.37 \text{ lbm})(113.83 \text{ Btu/lbm}) + (8.375 \text{ lbm})(104.99 \text{ Btu/lbm}) - 2962 \text{ Btu} \\ &= \mathbf{4561\ Btu} \end{aligned}$$

5-136 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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

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 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,

$$m_e = m_1 - m_2 = 1.594 - 1.138 = 0.456 \text{ kg}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (0.456 \text{ kg})(3246.2 \text{ kJ/kg}) + (1.138 \text{ kg})(3116.9 \text{ kJ/kg}) - (1.594 \text{ kg})(2773.2 \text{ kJ/kg}) \\ &= \mathbf{606.8 \text{ kJ}} \end{aligned}$$

5-137 A pressure cooker is initially half-filled with liquid water. If the pressure cooker is not to run out of liquid water for 1 h, the highest rate of heat transfer allowed is to be determined.

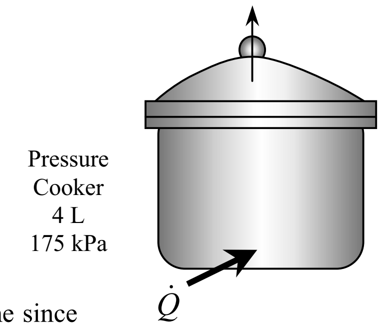
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

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

$$P_1 = 175 \text{ kPa} \rightarrow \boldsymbol{v}_f = 0.001057 \text{ m}^3/\text{kg}, \ \boldsymbol{v}_g = 1.0037 \text{ m}^3/\text{kg}$$
$$u_f = 486.82 \text{ kJ/kg}, \ u_g = 2524.5 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 175 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{g@175 \text{ kPa}} = 1.0036 \text{ m}^3/\text{kg} \\ u_2 = u_{g@175 \text{ kPa}} = 2524.5 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 175 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} h_e = h_{g@175 \text{ kPa}} = 2700.2 \text{ kJ/kg}$$



Analysis We take the cooker 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 mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{\boldsymbol{V}_f}{\boldsymbol{v}_f} + \frac{\boldsymbol{V}_g}{\boldsymbol{v}_g} = \frac{0.002 \text{ m}^3}{0.001057 \text{ m}^3/\text{kg}} + \frac{0.002 \text{ m}^3}{1.0036 \text{ m}^3/\text{kg}} = 1.893 + 0.002 = 1.895 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (1.893)(486.82) + (0.002)(2524.5) = 926.6 \text{ kJ}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{0.004 \text{ m}^3}{1.0037 \text{ m}^3/\text{kg}} = 0.004 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 1.895 - 0.004 = 1.891 \text{ kg}$$

$$Q_{in} = m_e h_e + m_2 u_2 - m_1 u_1$$
$$= (1.891 \text{ kg})(2700.2 \text{ kJ/kg}) + (0.004 \text{ kg})(2524.5 \text{ kJ/kg}) - 926.6 \text{ kJ} = 4188 \text{ kJ}$$

Thus,

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{4188 \text{ kJ}}{3600 \text{ s}} = \mathbf{1.163 \text{ kW}}$$

5-138 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.

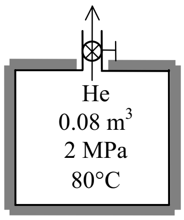
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There 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: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

$$m_2 = \tfrac{1}{2} m_1 \quad \text{(given)} \longrightarrow m_e = m_2 = \tfrac{1}{2} m_1$$



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 $m_1/2$ $$0 = h_e + u_2 - 2u_1 \quad or \quad 0 = c_p \frac{T_1 + T_2}{2} + c_v T_2 - 2c_v T_1$$

Dividing by c_v: $$0 = k(T_1 + T_2) + 2T_2 - 4T_1 \quad \text{since } k = c_p / c_v$$

Solving for T_2: $$T_2 = \frac{(4-k)}{(2+k)} T_1 = \frac{(4-1.667)}{(2+1.667)}(353\text{ K}) = \mathbf{225\ K}$$

The final pressure in the tank is

$$\frac{P_1 V}{P_2 V} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow P_2 = \frac{m_2 T_2}{m_1 T_2} P_1 = \frac{1}{2}\frac{225}{353}(2000\text{ kPa}) = \mathbf{637\ kPa}$$

5-139E 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:

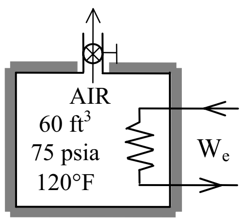
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 \mathbf{V}}{RT_1} = \frac{(75 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(580 \text{ R})} = 20.95 \text{ lbm}$$

$$m_2 = \frac{P_2 \mathbf{V}}{RT_2} = \frac{(30 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(580 \text{ R})} = 8.38 \text{ lbm}$$



Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 20.95 - 8.38 = 12.57 \text{ lbm}$$

$$\begin{aligned} W_{\text{e,in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (12.57 \text{ lbm})(138.66 \text{ Btu/lbm}) + (8.38 \text{ lbm})(98.90 \text{ Btu/lbm}) - (20.95 \text{ lbm})(98.90 \text{ Btu/lbm}) \\ &= \mathbf{500 \text{ Btu}} \end{aligned}$$

5-140 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).

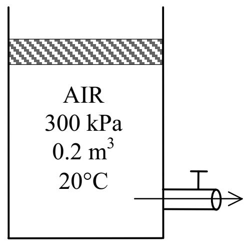
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1).

Analysis (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_e = m_1 - m_2$

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

$$Q_{in} + W_{b,in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$



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,

$$Q = \left(\tfrac{1}{2} m_1 + \tfrac{1}{2} m_1 - m_1\right) h = \mathbf{0}$$

5-141 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.

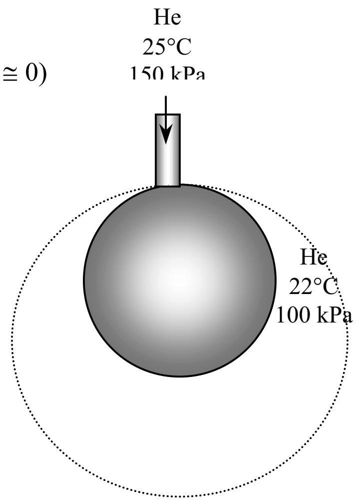
Properties The gas constant of helium is R = 2.0769 kJ/kg·K (Table A-1). The specific heats of helium are c_p = 5.1926 and c_v = 3.1156 kJ/kg·K (Table A-2a).

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: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

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

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$



$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(65 \text{ m}^3)}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 10.61 \text{ kg}$$

$$\frac{P_1}{P_2} = \frac{V_1}{V_2} \longrightarrow V_2 = \frac{P_2}{P_1} V_1 = \frac{150 \text{ kPa}}{100 \text{ kPa}} (65 \text{ m}^3) = 97.5 \text{ m}^3$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(150 \text{ kPa})(97.5 \text{ m}^3)}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(T_2 \text{ K})} = \frac{7041.74}{T_2} \text{ kg}$$

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 T_2 = **333.6 K**

5-142 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

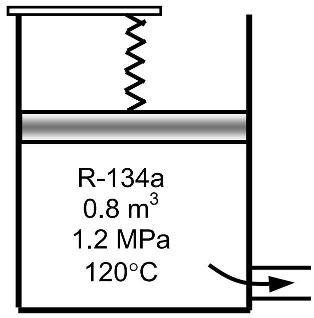
Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

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

$$W_{\text{b,in}} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(1200 + 600)\text{ kPa}}{2}(0.8\text{ - }0.5)\text{m}^3 = 270\text{ kJ}$$

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

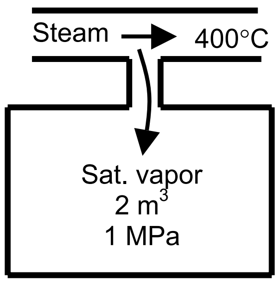
T_2 = **96.8°C**, m_e = **22.47 kg,** h_2 = 336.20 kJ/kg,

u_2 = 307.77 kJ/kg, v_2 = 0.04739 m^3/kg, m_2 = 10.55 kg

5-143 Steam flowing in a supply line is allowed to enter into an insulated tank until a specified state is achieved in the tank. The mass of the steam that has entered and the pressure of the steam in the supply line 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 tank remains constant. **2** Kinetic and potential energies are negligible.

Properties The initial and final properties of steam in the tank are (Tables A-5 and A-6)



$$\left.\begin{array}{l} P_1 = 1\,\text{MPa} \\ x_1 = 1\,(\text{sat. vap.}) \end{array}\right\} \begin{array}{l} v_1 = 0.19436\ \text{m}^3/\text{kg} \\ u_1 = 2582.8\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2\,\text{MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.12551\ \text{m}^3/\text{kg} \\ u_2 = 2773.2\ \text{kJ/kg} \end{array}$$

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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

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

$$m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial and final masses and the mass that has entered are

$$m_1 = \frac{V}{v_1} = \frac{2\ \text{m}^3}{0.19436\ \text{m}^3/\text{kg}} = 10.29\ \text{kg}$$

$$m_2 = \frac{V}{v_2} = \frac{2\ \text{m}^3}{0.12551\ \text{m}^3/\text{kg}} = 15.94\ \text{kg}$$

$$m_i = m_2 - m_1 = 15.94 - 10.29 = \mathbf{5.645\ kg}$$

Substituting,

$$(5.645\ \text{kg})h_i = (15.94\ \text{kg})(2773.2\ \text{kJ/kg}) - (10.29\ \text{kg})(2582.8\ \text{kJ/kg}) \longrightarrow h_i = 3120.3\ \text{kJ/kg}$$

The pressure in the supply line is

$$\left.\begin{array}{l} h_i = 3120.3\ \text{kJ/kg} \\ T_i = 400°\text{C} \end{array}\right\} P_i = \mathbf{8931\ kPa} \quad (\text{determined from EES})$$

5-144 Steam at a specified state is allowed to enter a piston-cylinder device in which steam undergoes a constant pressure expansion process. The amount of mass that enters and the amount of heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid entering the device remains constant. **2** Kinetic and potential energies are negligible.

Properties The properties of steam at various states are (Tables A-4 through A-6)

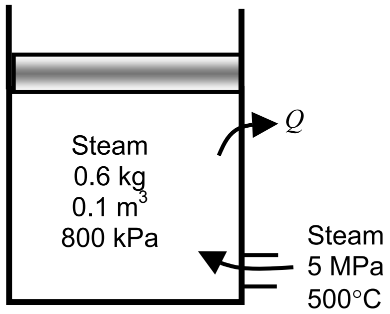
$$v_1 = \frac{V_1}{m_1} = \frac{0.1\ \text{m}^3}{0.6\ \text{kg}} = 0.16667\ \text{m}^3/\text{kg}$$

$$P_2 = P_1$$

$$\left.\begin{array}{l} P_1 = 800\ \text{kPa} \\ v_1 = 0.16667\ \text{m}^3/\text{kg} \end{array}\right\} u_1 = 2004.4\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ T_2 = 250°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.29321\ \text{m}^3/\text{kg} \\ u_2 = 2715.9\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 5\ \text{MPa} \\ T_i = 500°\text{C} \end{array}\right\} h_i = 3434.7\ \text{kJ/kg}$$



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: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_i = m_2 - m_1$

Energy balance:

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

$$Q_{in} - W_{b,out} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

Noting that the pressure remains constant, the boundary work is determined from

$$W_{b,out} = P(V_2 - V_1) = (800\ \text{kPa})(2 \times 0.1 - 0.1)\text{m}^3 = 80\ \text{kJ}$$

The final mass and the mass that has entered are

$$m_2 = \frac{V_2}{v_2} = \frac{0.2\ \text{m}^3}{0.29321\ \text{m}^3/\text{kg}} = 0.682\ \text{kg}$$

$$m_i = m_2 - m_1 = 0.682 - 0.6 = \mathbf{0.082\ kg}$$

(*b*) Finally, substituting into energy balance equation

$$Q_{in} - 80\ \text{kJ} + (0.082\ \text{kg})(3434.7\ \text{kJ/kg}) = (0.682\ \text{kg})(2715.9\ \text{kJ/kg}) - (0.6\ \text{kg})(2004.4\ \text{kJ/kg})$$

$$Q_{in} = \mathbf{447.9\ kJ}$$

Review Problems

5-145 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.

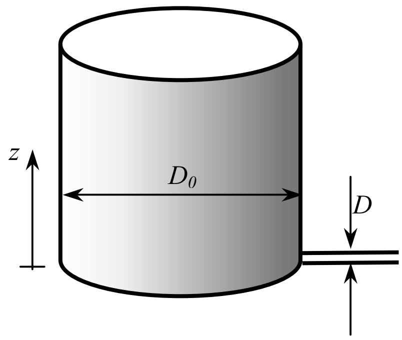
Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The tank is considered to be empty when the water level drops to the center of the valve.

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.

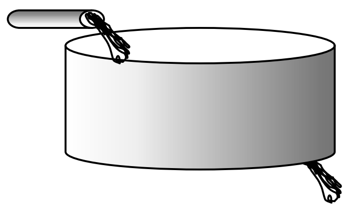
5-146 The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.

Assumptions **1** Water is supplied and discharged steadily. **2** The rate of evaporation of water is negligible. **3** No water is supplied or removed through other means.

Analysis The conservation of mass principle applied to the pool requires that the rate of increase in the amount of water in the pool be equal to the difference between the rate of supply of water and the rate of discharge. That is,

$$\frac{dm_{\text{pool}}}{dt} = \dot{m}_i - \dot{m}_e \quad \rightarrow \quad \dot{m}_i = \frac{dm_{\text{pool}}}{dt} + \dot{m}_e \quad \rightarrow \quad \dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e$$

since the density of water is constant and thus the conservation of mass is equivalent to conservation of volume. The rate of discharge of water is



$$\dot{V}_e = A_e V_{\text{e}} = (\pi D^2/4)V_{\text{e}} = [\pi(0.05\text{ m})^2/4](5\text{ m/s}) = 0.00982\text{ m}^3/\text{s}$$

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,

$$\frac{dV_{\text{pool}}}{dt} = A_{\text{cross-section}}V_{\text{level}} = (3\text{ m}\times 4\text{ m})(0.015\text{ m/min}) = 0.18\text{ m}^3/\text{min} = 0.00300\text{ m}^3/\text{s}$$

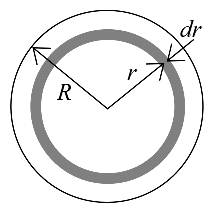
Substituting, the rate at which water is supplied to the pool is determined to be

$$\dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e = 0.003 + 0.00982 = \mathbf{0.01282\ m^3/s}$$

Therefore, water is supplied at a rate of 0.01282 m^3/s = 12.82 L/s.

5-147 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},

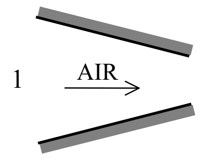
$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R V(r)r\,dr$$

5-148 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,



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

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

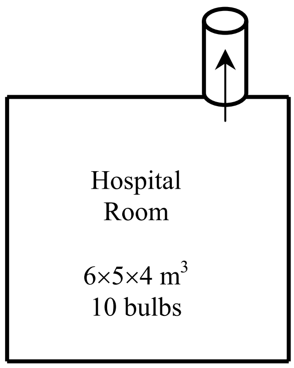
$$\rho_2 = \frac{A_1}{A_2}\frac{V_1}{V_2}\rho_1 = 2\frac{120 \text{ m/s}}{380 \text{ m/s}}(4.18 \text{ kg/m}^3) = \mathbf{2.64\ kg/m^3}$$

Discussion Note that the density of air decreases considerably despite a decrease in the cross-sectional area of the nozzle.

5-149 The air in a hospital room is to be replaced every 15 minutes. The minimum diameter of the duct is to be determined if the air velocity is not to exceed a certain value.

Assumptions **1** The volume occupied by the furniture etc in the room is negligible. **2** The incoming conditioned air does not mix with the air in the room.

Analysis The volume of the room is



$$V = (6 \text{ m})(5 \text{ m})(4 \text{ m}) = 120 \text{ m}^3$$

To empty this air in 20 min, the volume flow rate must be

$$\dot{V} = \frac{V}{\Delta t} = \frac{120 \text{ m}^3}{15 \times 60 \text{ s}} = 0.1333 \text{ m}^3/\text{s}$$

If the mean velocity is 5 m/s, the diameter of the duct is

$$\dot{V} = AV = \frac{\pi D^2}{4}V \quad \rightarrow \quad D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.1333 \text{ m}^3/\text{s})}{\pi(5 \text{ m/s})}} = \mathbf{0.184\ m}$$

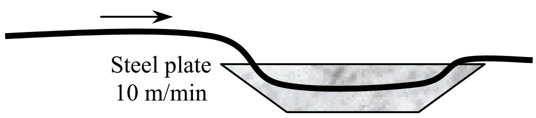
Therefore, the diameter of the duct must be at least 0.184 m to ensure that the air in the room is exchanged completely within 20 min while the mean velocity does not exceed 5 m/s.

Discussion This problem shows that engineering systems are sized to satisfy certain constraints imposed by certain considerations.

5-150 A long roll of large 1-Mn manganese steel plate is to be quenched in an oil bath at a specified rate. The mass flow rate of the plate is to be determined.

Assumptions The plate moves through the bath steadily.

Properties The density of steel plate is given to be ρ = 7854 kg/m^3.



Analysis The mass flow rate of the sheet metal through the oil bath is

$$\dot{m} = \rho\dot{V} = \rho w t V = (7854 \text{ kg/m}^3)(1 \text{ m})(0.005 \text{ m})(10 \text{ m/min}) = 393 \text{ kg/min} = \mathbf{6.55\ kg/s}$$

Therefore, steel plate can be treated conveniently as a "flowing fluid" in calculations.

5-151E A study quantifies the cost and benefits of enhancing IAQ by increasing the building ventilation. The net monetary benefit of installing an enhanced IAQ system to the employer per year is to be determined.

Assumptions The analysis in the report is applicable to this work place.

Analysis The report states that enhancing IAQ increases the productivity of a person by $90 per year, and decreases the cost of the respiratory illnesses by $39 a year while increasing the annual energy consumption by $6 and the equipment cost by about $4 a year. The net monetary benefit of installing an enhanced IAQ system to the employer per year is determined by adding the benefits and subtracting the costs to be

Net benefit = Total benefits – total cost = (90+39) – (6+4) = $119/year (per person)

The total benefit is determined by multiplying the benefit per person by the number of employees,

Total net benefit = No. of employees × Net benefit per person = 120×$119/year = **$14,280/year**

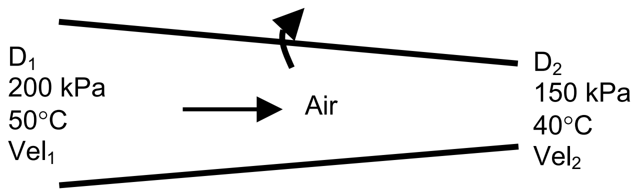
Discussion Note that the unseen savings in productivity and reduced illnesses can be very significant when they are properly quantified.

5-152 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:

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

$$\dot{m}_{in} = \dot{m}_{out} \longrightarrow \rho_1 A_1 V_1 = \rho_2 A_2 V_2 \longrightarrow \frac{P_1}{RT_1}\frac{\pi D_1^2}{4}V_1 = \frac{P_2}{RT_2}\frac{\pi D_2^2}{4}V_2 \longrightarrow \frac{P_1}{T_1}D_1^2 V_1 = \frac{P_2}{T_2}D_2^2 V_2$$

Energy balance:

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

$$\dot{E}_{in} = \dot{E}_{out} \longrightarrow h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{out}$$

or $$c_p T_1 + \frac{V_1^2}{2} = c_p T_2 + \frac{V_2^2}{2} + q_{out}$$

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

$$\left(\frac{200 \text{ kPa}}{323 \text{ K}}\right)(1.8 \text{ m})^2 V_1 = \left(\frac{150 \text{ kPa}}{313 \text{ K}}\right)(1.0 \text{ m})^2 V_2 \quad (1)$$

$$(1.005 \text{ kJ/kg.K})(323 \text{ K}) + \frac{V_1^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = (1.005 \text{ kJ/kg.K})(313 \text{ K}) + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) + 3.3 \text{ kJ/kg} \quad (2)$$

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

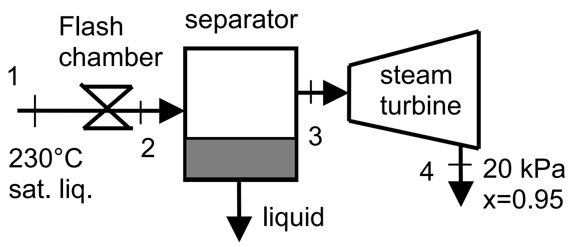
V_1 = **28.6 m/s**

V_2 = **120 m/s**

5-153 Geothermal water flows through a flash chamber, a separator, and a turbine in a geothermal power plant. The temperature of the steam after the flashing process and the power output from the turbine are to be determined for different flash chamber exit pressures.

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 insulated so that there are no heat losses to the surroundings. **4** Properties of steam are used for geothermal water.

Analysis For all components, we take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as



Energy balance:

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

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

For each component, the energy balance reduces to

Flash chamber: $h_1 = h_2$

Separator: $\dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_{liquid} h_{liquid}$

Turbine: $\dot{W}_T = \dot{m}_3 (h_3 - h_4)$

(*a*) For a flash chamber exit pressure of P_2 = 1 MPa

The properties of geothermal water are

$$h_1 = h_{sat\,@\,230°C} = 990.14 \text{ kJ/kg}$$

$$h_2 = h_1$$

$$x_2 = \frac{h_2 - h_{f\,@\,1000\text{ kPa}}}{h_{fg\,@\,1000\text{ kPa}}} = \frac{990.14 - 762.51}{2014.6} = 0.113$$

$$T_2 = T_{sat\,@\,1000\text{ kPa}} = \mathbf{179.9°C}$$

$$\left.\begin{array}{l} P_3 = 1000 \text{ kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2777.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ kPa} \\ x_4 = 0.95 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 251.42 + (0.05)(2357.5 \text{ kJ/kg}) = 2491.1 \text{ kJ/kg}$$

The mass flow rate of vapor after the flashing process is

$$\dot{m}_3 = x_2 \dot{m}_2 = (0.113)(50 \text{ kg/s}) = 5.649 \text{ kg/s}$$

Then, the power output from the turbine becomes

$$\dot{W}_T = (5.649 \text{ kg/s})(2777.1 - 2491.1) = \mathbf{1616\ kW}$$

Repeating the similar calculations for other pressures, we obtain

(*b*) For P_2 = 500 kPa, $T_2 = \mathbf{151.8°C}$, $\dot{W}_T = \mathbf{2134\ kW}$

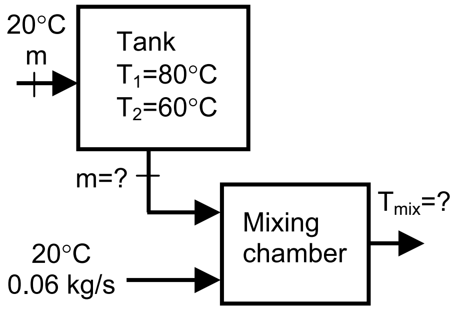
(*c*) For P_2 = 100 kPa, $T_2 = \mathbf{99.6°C}$, $\dot{W}_T = \mathbf{2333\ kW}$

(*d*) For P_2 =50 kPa, $T_2 = \mathbf{81.3°C}$, $\dot{W}_T = \mathbf{2173\ kW}$

5-154 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:

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

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

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

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 $$T_{tank,ave} = \frac{T_{tank,1} + T_{tank,2}}{2} = \frac{80+60}{2} = 70°\text{C}$$

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),

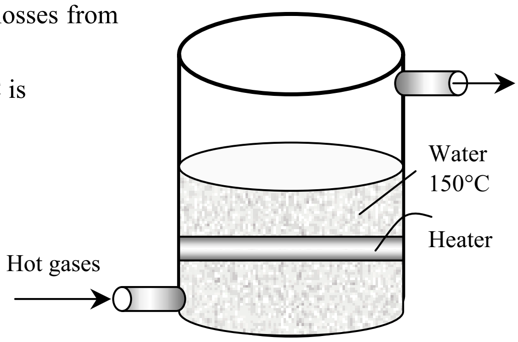
$$(0.0577 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70°\text{C}) + (0.06 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(20°\text{C})$$
$$= [(0.0577 + 0.06)\text{kg/s}](4.18 \text{ kJ/kg.°C})T_{mixture}$$

$$\longrightarrow T_{mixture} = \mathbf{44.5°C}$$

5-155 Water is boiled at a specified temperature by hot gases flowing through a stainless steel pipe submerged in water. The rate of evaporation of is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the boiler are negligible.

Properties The enthalpy of vaporization of water at 150°C is h_{fg} = 2113.8 kJ/kg (Table A-4).



Analysis The rate of heat transfer to water is given to be 74 kJ/s. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{74\ \text{kJ/s}}{2113.8\ \text{kJ/kg}} = \mathbf{0.0350\ kg/s}$$

5-156 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

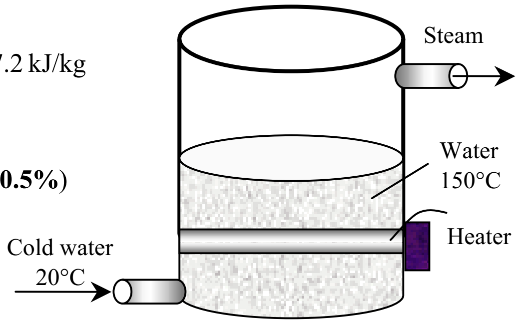
$$q_{\text{preheating}} = c\Delta T = (4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(150-20)^\circ\text{C} = 543.4\ \text{kJ/kg}$$

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

$$\text{Fraction to preheat} = \frac{q_{\text{preheating}}}{q_{\text{total}}} = \frac{543.4}{2657.2} = \mathbf{0.205}\ \text{(or } \mathbf{20.5\%}\text{)}$$



5-157 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.

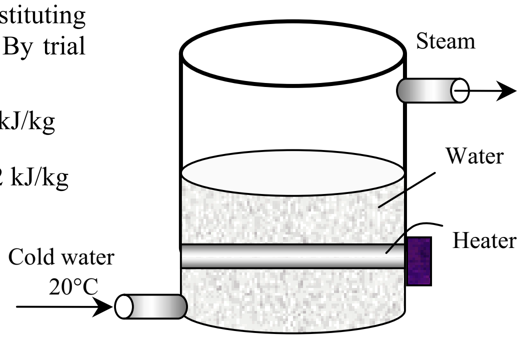
Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature, and Δh represents the amount of heat needed to preheat a unit mass of water from 20°C to the saturation temperature. Therefore,

$$q_{\text{preheating}} = q_{\text{boiling}}$$

$$(h_{f@T_{\text{sat}}} - h_{f@20°\text{C}}) = h_{fg@T_{\text{sat}}}$$

$$h_{f@T_{\text{sat}}} - 83.91\text{ kJ/kg} = h_{fg@T_{\text{sat}}} \rightarrow h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 83.91\text{ kJ/kg}$$

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)



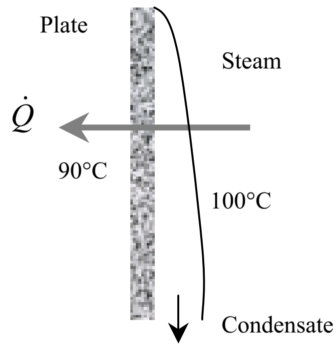
At 310°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}}$ = 1402.0 – 1325.9 = 76.1 kJ/kg

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.**

5-158 Saturated steam at 1 atm pressure and thus at a saturation temperature of T_{sat} = 100°C condenses on a vertical plate maintained at 90°C by circulating cooling water through the other side. The rate of condensation of steam is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The steam condenses and the condensate drips off at 100°C. (In reality, the condensate temperature will be between 90 and 100, and the cooling of the condensate a few °C should be considered if better accuracy is desired).

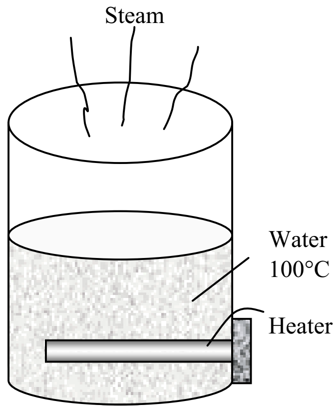


Properties The enthalpy of vaporization of water at 1 atm (101.325 kPa) is h_{fg} = 2256.5 kJ/kg (Table A-5).

Analysis The rate of heat transfer during this condensation process is given to be 180 kJ/s. Noting that the heat of vaporization of water represents the amount of heat released as a unit mass of vapor at a specified temperature condenses, the rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h_{fg}} = \frac{180\text{ kJ/s}}{2256.5\text{ kJ/kg}} = \mathbf{0.0798\ kg/s}$$

5-159 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

$$\dot{m}_{\text{evaporation}} = \frac{\dot{W}_{\text{e,boiling}}}{h_{fg}} = \frac{3\text{ kJ/s}}{2256.4\text{ kJ/kg}} = \mathbf{0.00133\ kg/s = 4.79\ kg/h}$$

5-160 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

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

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

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

$$\dot{m}_1 c_p T_1 + \dot{m}_2 c_p T_2 = \dot{m}_3 c_p T_3$$

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

Solving for final temperature, we find

$$T_3 = \frac{\dot{m}_1}{\dot{m}_3} T_1 + \frac{\dot{m}_2}{\dot{m}_3} T_2$$

5-161 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.

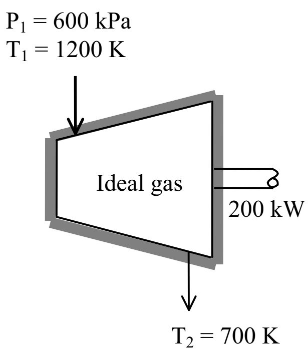
Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

which can be rearranged to solve for mass flow rate

$$\dot{m} = \frac{\dot{W}_{\text{out}}}{h_1 - h_2} = \frac{\dot{W}_{\text{out}}}{c_p(T_1 - T_2)} = \frac{200\ \text{kW}}{(1.13\ \text{kJ/kg.K})(1200-700)\text{K}} = 0.354\ \text{kg/s}$$



The inlet specific volume and the volume flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(1200\ \text{K})}{600\ \text{kPa}} = 0.6\ \text{m}^3/\text{kg}$$

Thus, $\dot{V} = \dot{m}v_1 = (0.354\ \text{kg/s})(0.6\ \text{m}^3/\text{kg}) = \mathbf{0.212\ m^3/s}$

5-162 Two identical buildings in Los Angeles and Denver have the same infiltration rate. The ratio of the heat losses by infiltration at the two cities under identical conditions is to be determined.

Assumptions **1** Both buildings are identical and both are subjected to the same conditions except the atmospheric conditions. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Steady flow conditions exist.

Analysis We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

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

Los Angeles: 101 kPa
Denver: 83 kPa

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

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

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

Then the sensible infiltration heat loss (heat gain for the infiltrating air) can be expressed

$$\dot{Q}_{\text{infiltration}} = \dot{m}_{\text{air}}c_p(T_i - T_o) = \rho_{o,\text{air}}(\text{ACH})(V_{\text{building}})c_p(T_i - T_o)$$

where *ACH* is the infiltration volume rate in *air changes per hour*. Therefore, the infiltration heat loss is proportional to the density of air, and thus the ratio of infiltration heat losses at the two cities is simply the densities of outdoor air at those cities,

$$\text{Infiltration heat loss ratio} = \frac{\dot{Q}_{\text{infiltration, Los Angeles}}}{\dot{Q}_{\text{infiltration, Denver}}} = \frac{\rho_{o,\text{ air, Los Angeles}}}{\rho_{o,\text{ air, Denver}}}$$

$$= \frac{(P_0/RT_0)_{\text{Los Angeles}}}{(P_0/RT_0)_{\text{Denver}}} = \frac{P_{o,\text{ Los Angeles}}}{P_{0,\text{ Denver}}} = \frac{101\ \text{kPa}}{83\ \text{kPa}} = \mathbf{1.22}$$

Therefore, the infiltration heat loss in Los Angeles will be 22% higher than that in Denver under identical conditions.

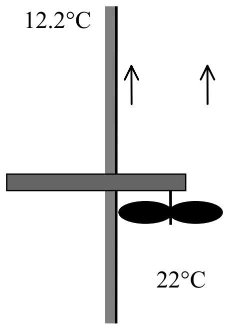
5-163 The ventilating fan of the bathroom of an electrically heated building in San Francisco runs continuously. The amount and cost of the heat "vented out" per month in winter are to be determined.

Assumptions **1** We take the atmospheric pressure to be 1 atm = 101.3 kPa since San Francisco is at sea level. **2** The building is maintained at 22°C at all times. **3** The infiltrating air is heated to 22°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady flow conditions exist.

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

Analysis The density of air at the indoor conditions of 1 atm and 22°C is

$$\rho_o = \frac{P_o}{RT_o} = \frac{(101.3\text{ kPa})}{(0.287\text{ kPa.m}^3/\text{kg.K})(22+273\text{ K})} = 1.20\text{ kg/m}^3$$



Then the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho\dot{V}_{\text{air}} = (1.20\text{ kg/m}^3)(0.030\text{ m}^3/\text{s}) = 0.036\text{ kg/s}$$

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

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

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

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

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

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 12.2°C, the sensible infiltration heat loss (heat gain for the infiltrating air) due to venting by fans can be expressed

$$\dot{Q}_{\text{loss by fan}} = \dot{m}_{\text{air}}c_{\text{p}}(T_{\text{indoors}} - T_{\text{outdoors}})$$
$$= (0.036\text{ kg/s})(1.005\text{ kJ/kg.°C})(22-12.2)\text{°C} = 0.355\text{ kJ/s} = 0.355\text{ kW}$$

Then the amount and cost of the heat "vented out" per month (1 month = 30×24 = 720 h) becomes

Energy loss = $\dot{Q}_{\text{loss by fan}}\Delta t$ = (0.355 kW)(720 h/month) = **256 kWh/month**

Money loss = (Energy loss)(Unit cost of energy) = (256 kWh/month)($0.09/kWh) = **$23.0/month**

Discussion Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used with care.

5-164 Chilled air is to cool a room by removing the heat generated in a large insulated classroom by lights and students. The required flow rate of air that needs to be supplied to the room is to be determined.

Assumptions **1** The moisture produced by the bodies leave the room as vapor without any condensing, and thus the classroom has no latent heat load. **2** Heat gain through the walls and the roof is negligible. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady operating conditions exist.

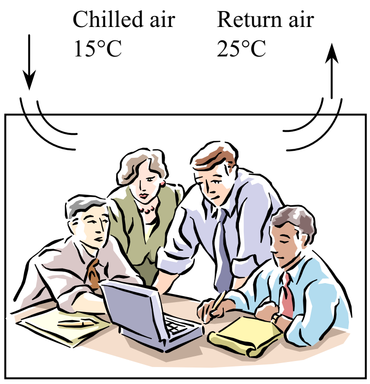
Properties The specific heat of air at room temperature is 1.005 kJ/kg·°C (Table A-2). The average rate of sensible heat generation by a person is given to be 60 W.

Analysis The rate of sensible heat generation by the people in the room and the total rate of sensible internal heat generation are

$$\dot{Q}_{\text{gen, sensible}} = \dot{q}_{\text{gen, sensible}}(\text{No. of people}) = (60\text{ W/person})(150\text{ persons}) = 9000\text{ W}$$

$$\dot{Q}_{\text{total, sensible}} = \dot{Q}_{\text{gen, sensible}} + \dot{Q}_{\text{lighting}} = 9000 + 6000 = 15{,}000\text{ W}$$

Both of these effects can be viewed as heat gain for the chilled air stream, which can be viewed as a steady stream of cool air that is heated as it flows in an imaginary duct passing through the room. The energy balance for this imaginary steady-flow system can be expressed in the rate form as



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

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

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

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

Then the required mass flow rate of chilled air becomes

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{total, sensible}}}{c_p \Delta T} = \frac{15\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot{}^\circ\text{C})(25-15)^\circ\text{C}} = \mathbf{1.49\ kg/s}$$

Discussion The latent heat will be removed by the air-conditioning system as the moisture condenses outside the cooling coils.

5-165 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

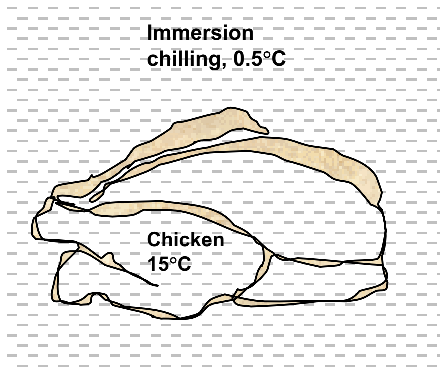
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500\text{ chicken / h})(2.2\text{ kg / chicken}) = 1100\text{ kg / h} = 0.3056\text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as



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

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

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

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

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m} c_p \Delta T)_{\text{chicken}} = (0.3056\text{ kg/s})(3.54\text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0\ kW}$$

The chiller gains heat from the surroundings at a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 0.056 = 13.056\text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p \Delta T)_{\text{water}}} = \frac{13.056\text{ kW}}{(4.18\text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

5-166 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

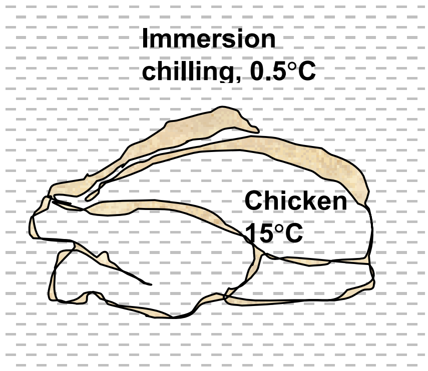
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant. **3** Heat gain of the chiller is negligible.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500 \text{ chicken / h})(2.2 \text{ kg / chicken}) = 1100 \text{ kg / h} = 0.3056 \text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as



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

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

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

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

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m} c_p \Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0 \ kW}$$

Heat gain of the chiller from the surroundings is negligible. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} = 13.0 \text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p \Delta T)_{\text{water}}} = \frac{13.0 \text{ kW}}{(4.18 \text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56 \ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

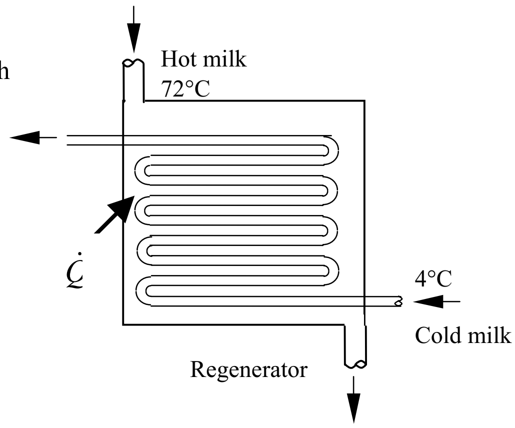
5-167 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year are to be determined.

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

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

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

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

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

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**

5-168E 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.

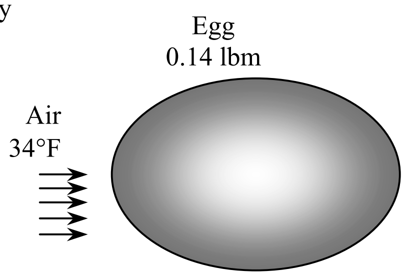
Assumptions **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated. **4** The properties of eggs are constant. **5** The local atmospheric pressure is 1 atm.

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

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p \Delta T)_{\text{air}}} = \frac{44{,}800 \text{ Btu/h}}{(0.24 \text{ Btu/lbm.°F})(10\text{°F})} = 18{,}667 \text{ lbm/h}$$

$$\rho_{\text{air}} = \frac{P}{RT} = \frac{14.7 \text{ psia}}{(0.3704 \text{ psia.ft}^3\text{/lbm.R})(34 + 460)\text{R}} = 0.0803 \text{ lbm/ft}^3$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{18{,}667 \text{ lbm/h}}{0.0803 \text{ lbm/ft}^3} = \mathbf{232{,}500 \text{ ft}^3\text{/h}}$$

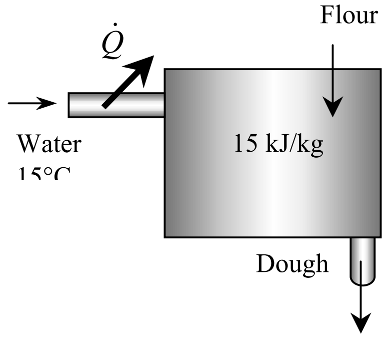
5-169 Dough is made with refrigerated water in order to absorb the heat of hydration and thus to control the temperature rise during kneading. The temperature to which the city water must be cooled before mixing with flour is to be determined to avoid temperature rise during kneading.

Assumptions **1** Steady operating conditions exist. **2** The dough is at uniform temperatures before and after cooling. **3** The kneading section is well-insulated. **4** The properties of water and dough are constant.

Properties The specific heats of the flour and the water are given to be 1.76 and 4.18 kJ/kg.°C, respectively. The heat of hydration of dough is given to be 15 kJ/kg.

Analysis It is stated that 2 kg of flour is mixed with 1 kg of water, and thus 3 kg of dough is obtained from each kg of water. Also, 15 kJ of heat is released for each kg of dough kneaded, and thus 3×15 = 45 kJ of heat is released from the dough made using 1 kg of water.

Taking the cooling section of water 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



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

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

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

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

In order for water to absorb all the heat of hydration and end up at a temperature of 15ºC, its temperature before entering the mixing section must be reduced to

$$Q_{in} = Q_{dough} = mc_p(T_2 - T_1) \rightarrow T_1 = T_2 - \frac{Q}{mc_p} = 15°\text{C} - \frac{45 \text{ kJ}}{(1 \text{ kg})(4.18 \text{ kJ/kg.°C})} = \mathbf{4.2°C}$$

That is, the water must be precooled to 4.2ºC before mixing with the flour in order to absorb the entire heat of hydration.

5-170 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.

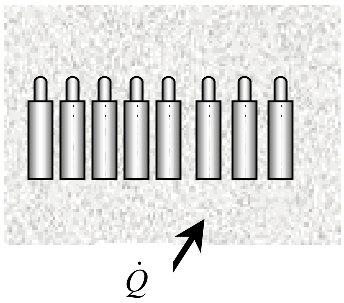
Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 55°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

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



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

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

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

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

Then the rate of heat removal by the bottles as they are heated from 20 to 55°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} c_p \Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg.°C})(55 - 20)\text{°C} = 56{,}000 \text{ W}$$

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

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg·°C})(55 - 15)\text{°C} = 446 \text{ W}$$

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.

5-171 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 50°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

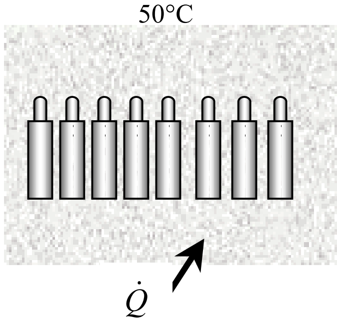
Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

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

Water bath
50°C



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

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

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

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

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.

5-172 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

$$\dot{m} = \rho\dot{\boldsymbol{V}} = \rho(\pi r_0^2)\boldsymbol{V} = (2702\text{ kg/m}^3)\pi(0.0015\text{ m})^2(10\text{ m/min}) = 0.191\text{ kg/min}$$

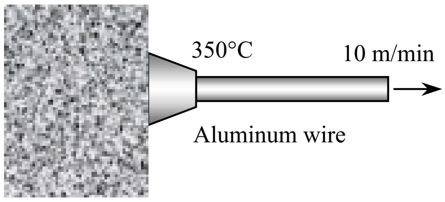
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.191\text{ kg/min})(0.896\text{ kJ/kg.°C})(350 - 50)\text{°C} = 51.3\text{ kJ/min} = \mathbf{0.856\text{ kW}}$$

5-173 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

$$\dot{m} = \rho\dot{\boldsymbol{V}} = \rho(\pi r_0^2)\boldsymbol{V} = (8950\text{ kg/m}^3)\pi(0.0015\text{ m})^2(10\text{ m/min}) = 0.633\text{ kg/min}$$

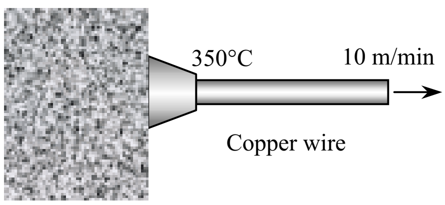
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.633\text{ kg/min})(0.383\text{ kJ/kg.°C})(350 - 50)\text{°C} = 72.7\text{ kJ/min} = \mathbf{1.21\text{ kW}}$$

5-174 Steam at a saturation temperature of T_{sat} = 40°C condenses on the outside of a thin horizontal tube. Heat is transferred to the cooling water that enters the tube at 25°C and exits at 35°C. The rate of condensation of steam is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The changes in kinetic and potential energies are negligible.

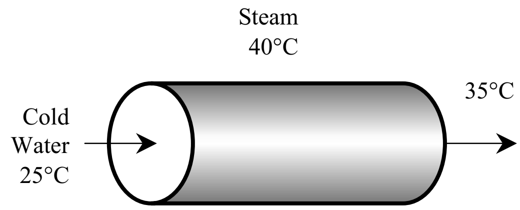
Properties The properties of water at room temperature are ρ = 997 kg/m^3 and c_p = 4.18 kJ/kg.°C (Table A-3). The enthalpy of vaporization of water at 40°C is h_{fg} = 2406.0 kJ/kg (Table A-4).

Analysis The mass flow rate of water through the tube is

$$\dot{m}_{water} = \rho V A_c = (997\ \text{kg/m}^3)(2\ \text{m/s})[\pi(0.03\ \text{m})^2/4] = 1.409\ \text{kg/s}$$

Taking the volume occupied by the cold water in the tube as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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



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

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

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

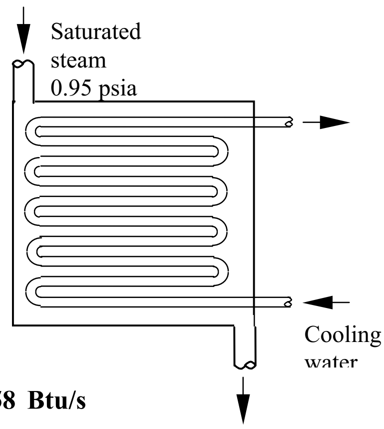
Then the rate of heat transfer to the water and the rate of condensation become

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (1.409\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})(35-25)°\text{C} = 58.9\ \text{kW}$$

$$\dot{Q} = \dot{m}_{cond} h_{fg} \rightarrow \dot{m}_{cond} = \frac{\dot{Q}}{h_{fg}} = \frac{58.9\ \text{kJ/s}}{2406.0\ \text{kJ/kg}} = \mathbf{0.0245\ kg/s}$$

5-175E Saturated steam at a saturation pressure of 0.95 psia and thus at a saturation temperature of T_{sat} = 100°F (Table A-4E) condenses on the outer surfaces of 144 horizontal tubes by circulating cooling water arranged in a 12 × 12 square array. The rate of heat transfer to the cooling water and the average velocity of the cooling water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal. **3** Water is an incompressible substance with constant properties at room temperature. **4** The changes in kinetic and potential energies 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). The enthalpy of vaporization of water at a saturation pressure of 0.95 psia is h_{fg} = 1036.7 Btu/lbm (Table A-4E).

Analysis (*a*) The rate of heat transfer from the steam to the cooling water is equal to the heat of vaporization released as the vapor condenses at the specified temperature,

$$\dot{Q} = \dot{m}h_{fg} = (6800\ \text{lbm/h})(1036.7\ \text{Btu/lbm}) = \mathbf{7{,}049{,}560\ \ Btu/h = 1958\ \ Btu/s}$$

(*b*) All of this energy is transferred to the cold water. Therefore, the mass flow rate of cold water must be

$$\dot{Q} = \dot{m}_{water} c_p \Delta T \rightarrow \dot{m}_{water} = \frac{\dot{Q}}{c_p \Delta T} = \frac{1958\ \text{Btu/s}}{(1.00\ \text{Btu/lbm.°F})(8°\text{F})} = 244.8\ \text{lbm/s}$$

Then the average velocity of the cooling water through the 144 tubes becomes

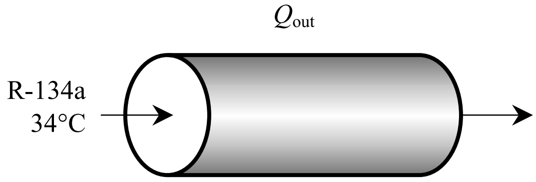
$$\dot{m} = \rho A V \rightarrow V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{\rho(n\pi D^2/4)} = \frac{244.8\ \text{lbm/s}}{(62.1\ \text{lbm/ft}^3)[144\pi(1/12\ \text{ft})^2/4]} = \mathbf{5.02\ ft/s}$$

5-176 Saturated refrigerant-134a vapor at a saturation temperature of T_{sat} = 34°C condenses inside a tube. The rate of heat transfer from the refrigerant for the condensate exit temperatures of 34°C and 20°C are to be determined.

Assumptions **1** Steady flow conditions exist. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions involved.

Properties The properties of saturated refrigerant-134a at 34°C are h_f = 99.40 kJ/kg, are h_g = 268.57 kJ/kg, and are h_{fg} = 169.17 kJ/kg. The enthalpy of saturated liquid refrigerant at 20°C is h_f = 79.32 kJ/kg, (Table A-11).

Analysis We take the *tube and the refrigerant in it* 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 $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that heat is lost from the system, the energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

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

where at the inlet state $h_1 = h_g$ = 268.57 kJ/kg. Then the rates of heat transfer during this condensation process for both cases become

Case 1: T_2 = 34°C: $h_2 = h_{f@34°C}$ = 99.40 kJ/kg.

$$\dot{Q}_{out} = (0.1\text{ kg/min})(268.57 - 99.40)\text{ kJ/kg} = \mathbf{16.9\ kg/min}$$

Case 2: T_2 = 20°C: $h_2 \cong h_{f@20°C}$ =79.32 kJ/kg.

$$\dot{Q}_{out} = (0.1\text{ kg/min})(268.57 - 79.32)\text{ kJ/kg} = \mathbf{18.9\ kg/min}$$

Discussion Note that the rate of heat removal is greater in the second case since the liquid is subcooled in that case.

5-177E 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).

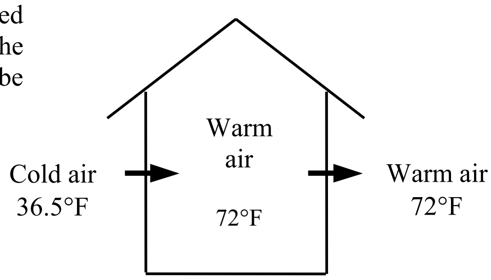
Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5\text{ psia}}{(0.3704\text{ psia.ft}^3/\text{lbm.R})(496.5\text{ R})} = 0.0734\text{ lbm/ft}^3$$

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



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

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

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

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

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.

5-178 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.

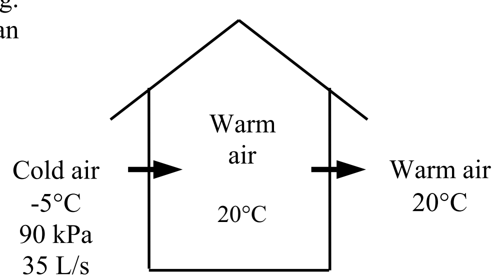
Assumptions **1** The house is maintained at 20°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 20°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies 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

$$\begin{aligned}\dot{Q}_{\text{infiltration}} &= \rho_o \dot{V}_{air} c_p (T_i - T_o)\\ &= (1.17\text{ kg/m}^3)(0.035\text{ m}^3\text{/s})(1.005\text{ kJ/kg.°C})[20-(-5)]\text{°C}\\ &= \mathbf{1.029\ kW}\end{aligned}$$

Therefore, sensible heat will be lost at a rate of 1.029 kJ/s due to infiltration.

5-179 The maximum flow rate of a standard shower head can be reduced from 13.3 to 10.5 L/min by switching to low-flow shower heads. The ratio of the hot-to-cold water flow rates and the amount of electricity saved by a family of four per year by replacing the standard shower heads by the low-flow ones 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 kinetic and potential energies are negligible, $ke \cong pe \cong 0$. **3** Heat losses from the system are negligible and thus $\dot{Q} \cong 0$. **4** There are no work interactions involved. **5** .Showers operate at maximum flow conditions during the entire shower. **6** Each member of the household takes a 5-min shower every day. **7** Water is an incompressible substance with constant properties. **8** The efficiency of the electric water heater is 100%.

Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and c = 4.18 kJ/kg.°C (Table A-3).

Analysis (a) We take the *mixing chamber* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there are two inlets and one exit. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

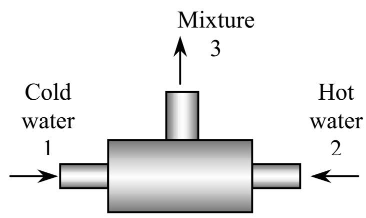
Mass balance:

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

$$\dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

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



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

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

Combining the mass and energy balances and rearranging,

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

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

Then the ratio of the mass flow rates of the hot water to cold water becomes

$$\frac{\dot{m}_2}{\dot{m}_1} = \frac{h_3 - h_1}{h_2 - h_3} = \frac{c(T_3 - T_1)}{c(T_2 - T_3)} = \frac{T_3 - T_1}{T_2 - T_3} = \frac{(42 - 15)°\text{C}}{(55 - 42)°\text{C}} = \mathbf{2.08}$$

(b) The low-flow heads will save water at a rate of

$$\dot{\boldsymbol{V}}_{saved} = [(13.3\text{-}10.5) \text{ L/min}](5 \text{ min/person.day})(4 \text{ persons})(365 \text{ days/yr}) = 20{,}440 \text{ L/year}$$

$$\dot{m}_{saved} = \rho \dot{\boldsymbol{V}}_{saved} = (1 \text{ kg/L})(20{,}440 \text{ L/year}) = 20{,}440 \text{ kg/year}$$

Then the energy saved per year becomes

$$\begin{aligned} \text{Energy saved} &= \dot{m}_{saved} c \Delta T = (20{,}440 \text{ kg/year})(4.18 \text{ kJ/kg.°C})(42\text{ - }15)°\text{C} \\ &= 2{,}307{,}000 \text{ kJ/year} \\ &= \mathbf{641 \text{ kWh}} \quad (\text{since } 1 \text{ kWh} = 3600 \text{ kJ}) \end{aligned}$$

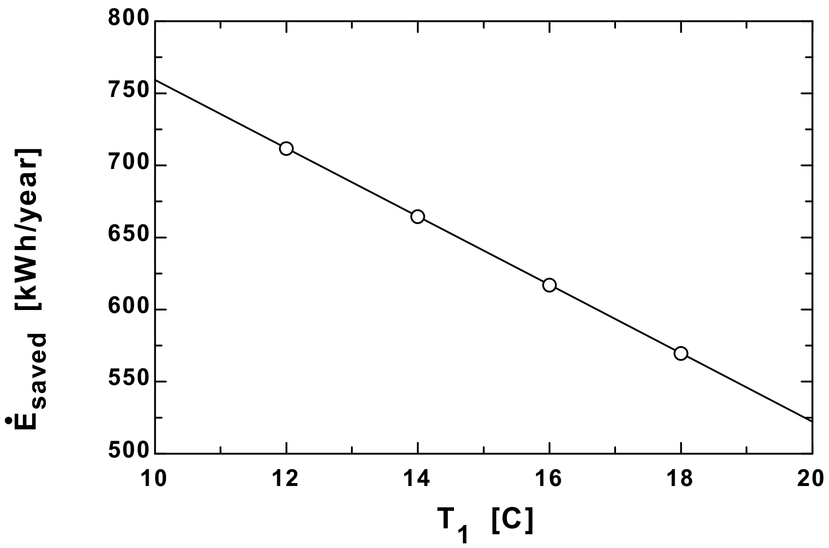
Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per year.

5-180 EES Problem 5-179 is reconsidered. The effect of the inlet temperature of cold water on the energy saved by using the low-flow showerhead as the inlet temperature varies from 10°C to 20°C is to be investigated. The electric energy savings is to be plotted against the water inlet temperature.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
C_P = 4.18 [kJ/kg-K ]
density=1[kg/L]
{T_1 = 15 [C]}
T_2 =  55 [C]
T_3 = 42 [C]
V_dot_old = 13.3 [L/min]
V_dot_new = 10.5 [L/min]
m_dot_1=1[kg/s]  "We can set m_dot_1 = 1 without loss of generality."

"Analysis:"
 "(a) We take the mixing chamber as the system. This is a control volume since mass
crosses the system boundary during the process. We note that there are two inlets
and one exit.  The mass and energy balances for this steady-flow system can be
expressed in the rate form as follows:"

"Mass balance:"
m_dot_in - m_dot_out = DELTAm_dot_sys
DELTAm_dot_sys=0
m_dot_in =m_dot_1 + m_dot_2
m_dot_out = m_dot_3

"The ratio of the mass flow rates of the hot water to cold water is
obtained by setting m_dot_1=1[kg/s].  Then m_dot_2 represents
the ratio of m_dot_2/m_dot_1"

"Energy balance:"
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys=0
E_dot_in = m_dot_1*h_1 + m_dot_2*h_2
E_dot_out = m_dot_3*h_3
h_1 = C_P*T_1
h_2 = C_P*T_2
h_3 = C_P*T_3

"(b) The low-flow heads will save water at a rate of "
V_dot_saved = (V_dot_old  -  V_dot_new)"L/min"*(5"min/person-
day")*(4"persons")*(365"days/year") "[L/year]"
m_dot_saved=density*V_dot_saved "[kg/year]"

"Then the energy saved per year becomes"
E_dot_saved=m_dot_saved*C_P*(T_3 - T_1)"kJ/year"*convert(kJ,kWh) "[kWh/year]"

"Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per year. "

"Ratio of hot-to-cold water flow rates:"
m_ratio = m_dot_2/m_dot_1
```

E_{saved} [kWh/year]	T_1 [C]
759.5	10
712	12
664.5	14
617.1	16
569.6	18
522.1	20



5-181 A fan is powered by a 0.5 hp motor, and delivers air at a rate of 85 m^3/min. The highest possible air velocity at the fan exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity and the change in potential energy are negligible, $V_1 \cong 0$ and $\Delta pe \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** The efficiencies of the motor and the fan are 100% since best possible operation is assumed. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be ρ = 1.18 kg/m^3. The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

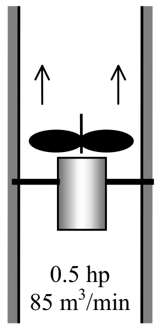
Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

The velocity of air leaving the fan will be highest when all of the entire electrical energy drawn by the motor is converted to kinetic energy, and the friction between the air layers is zero. In this best possible case, no energy will be converted to thermal energy, and thus the temperature change of air will be zero, $T_2 = T_1$. Then the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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



Noting that the temperature and thus enthalpy remains constant, the relation above simplifies further to

$$\dot{W}_{e,in} = \dot{m}V_2^2/2$$

where

$$\dot{m} = \rho\dot{V} = (1.18 \text{ kg/m}^3)(85 \text{ m}^3/\text{min}) = 100.3 \text{ kg/min} = 1.67 \text{ kg/s}$$

Solving for V_2 and substituting gives

$$V_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} = \sqrt{\frac{2(0.5 \text{ hp})}{1.67 \text{ kg/s}}\left(\frac{745.7 \text{ W}}{1 \text{ hp}}\right)\left(\frac{1 \text{ m}^2/\text{s}^2}{1 \text{ W}}\right)} = \mathbf{21.1 \text{ m/s}}$$

Discussion In reality, the velocity will be less because of the inefficiencies of the motor and the fan.

5-182 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.

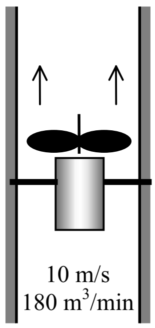
Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\text{CV}} = 0$ and $\Delta E_{\text{CV}} = 0$. **2** The inlet velocity is negligible, $V_1 \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed 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

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1.20\ \text{kg/m}^3)(180\ \text{m}^3/\text{min}) = 216\ \text{kg/min} = 3.6\ \text{kg/s}$$

$$\Delta \dot{\text{KE}} = \dot{m}\frac{V_2^2 - V_1^2}{2} = (3.6\ \text{kg/s})\frac{(10\ \text{m/s})^2 - 0}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 0.18\ \text{kW}$$



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

$$\dot{V} = VA = V(\pi D^2/4) \rightarrow D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(180\ \text{m}^3/\text{min})}{\pi(10\ \text{m/s})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right)} = \mathbf{0.618\ m}$$

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

5-183 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).

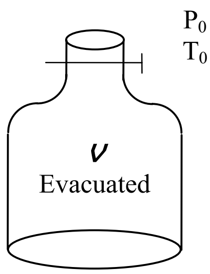
Analysis We take the bottle as the system. It is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can 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,

$$Q_{\text{in}} = m_2(c_v - c_p)T_0 = -m_2 R T_0 = -\frac{P_0 \boldsymbol{V}}{R T_0} R T_0 = -P_0 \boldsymbol{V}$$

Therefore, $\quad Q_{\text{out}} = \boldsymbol{P_0 V} \quad$ (Heat is lost from the tank)

5-184 An adiabatic air compressor is powered by a direct-coupled steam turbine, which is also driving a generator. The net power delivered to the generator is to be determined.

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

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

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} h_3 = 3343.6 \text{ kJ/kg}$$

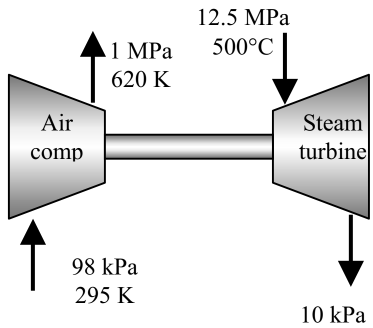
and

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg}$$

From the air table (Table A-17),

$$T_1 = 295 \text{ K} \longrightarrow h_1 = 295.17 \text{ kJ/kg}$$

$$T_2 = 620 \text{ K} \longrightarrow h_2 = 628.07 \text{ kJ/kg}$$



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

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

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

For the turbine and the compressor it becomes

Compressor: $\dot{W}_{\text{comp, in}} + \dot{m}_{\text{air}} h_1 = \dot{m}_{\text{air}} h_2 \quad \rightarrow \quad \dot{W}_{\text{comp, in}} = \dot{m}_{\text{air}} (h_2 - h_1)$

Turbine: $\dot{m}_{\text{steam}} h_3 = \dot{W}_{\text{turb, out}} + \dot{m}_{\text{steam}} h_4 \quad \rightarrow \quad \dot{W}_{\text{turb, out}} = \dot{m}_{\text{steam}} (h_3 - h_4)$

Substituting,

$$\dot{W}_{\text{comp,in}} = (10 \text{ kg/s})(628.07 - 295.17) \text{ kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{\text{turb,out}} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{ kJ/kg} = 23,777 \text{ kW}$$

Therefore,

$$\dot{W}_{\text{net,out}} = \dot{W}_{\text{turb,out}} - \dot{W}_{\text{comp,in}} = 23,777 - 3329 = \mathbf{20,448 \text{ kW}}$$

5-185 Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. The electric power input to the heater, and the money that will be saved during a 10-min shower by installing a heat exchanger with an effectiveness of 0.50 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** Heat losses from the pipe are negligible.

Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and c = 4.18 kJ/kg·°C (Table A-3).

Analysis We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the energy balance for this steady-flow system can be expressed in the rate form as

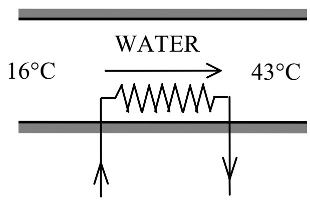
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v(P_2 - P_1)^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

where

$$\dot{m} = \rho\dot{V} = (1 \text{ kg/L})(10 \text{ L/min}) = 10 \text{ kg/min}$$



Substituting,

$$\dot{W}_{e,in} = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(43 - 16)^\circ\text{C} = \mathbf{18.8 \text{ kW}}$$

The energy recovered by the heat exchanger is

$$\dot{Q}_{saved} = \varepsilon\dot{Q}_{max} = \varepsilon\dot{m}C(T_{max} - T_{min})$$
$$= 0.5(10/60 \text{ kg/s})(4.18 \text{ kJ/kg.}^\circ\text{C})(39 - 16)^\circ\text{C}$$
$$= 8.0 \text{ kJ/s} = 8.0\text{kW}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot{W}_{in,new} = \dot{W}_{in,old} - \dot{Q}_{saved} = 18.8 - 8.0 = \mathbf{10.8 \text{ kW}}$$

The money saved during a 10-min shower as a result of installing this heat exchanger is

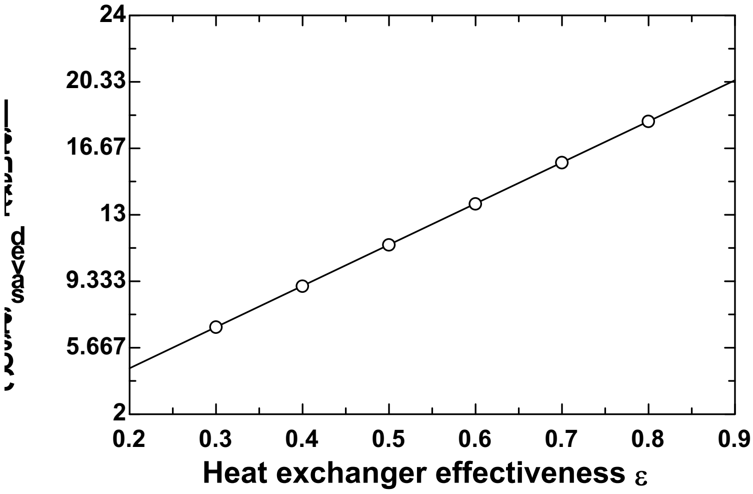
$$(8.0 \text{ kW})(10/60 \text{ h})(8.5 \text{ cents/kWh}) = \mathbf{11.3 \text{ cents}}$$

5-186 EES Problem 5-185 is reconsidered. The effect of the heat exchanger effectiveness on the money saved as the effectiveness ranges from 20 percent to 90 percent is to be investigated, and the money saved is to be plotted against the effectiveness.

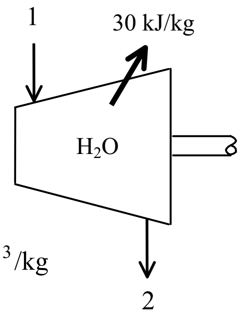
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
density = 1 [kg/L]
V_dot = 10 [L/min]
C = 4.18 [kJ/kg-C]
T_1 = 16 [C]
T_2 = 43 [C]
T_max = 39 [C]
T_min = T_1
epsilon = 0.5  "heat exchanger effectiveness "
EleRate = 8.5 [cents/kWh]

"For entrance, one exit, steady flow m_dot_in = m_dot_out = m_dot_wat er:"
m_dot_water= density*V_dot /convert(min,s)

"Energy balance for the pipe:"
W_dot_ele_in+ m_dot_water*h_1=m_dot_water*h_2 "Neglect ke and pe"

"For incompressible fluid in a constant pressure process, the enthalpy is:"
h_1 = C*T_1
h_2 = C*T_2

"The energy recovered by the heat exchanger is"
Q_dot_saved=epsilon*Q_dot_max
Q_dot_max = m_dot_water*C*(T_max - T_min)

"Therefore, 8.0 kW less energy is needed in this case, and the required
electric power in this case reduces to"
W_dot_ele_new = W_dot_ele_in - Q_dot_saved

"The money saved during a 10-min shower as a result of installing this heat exchanger is"
Costs_saved = Q_dot_saved*time*convert(min,h)*EleRate
time=10 [min]
```

$Costs_{saved}$ [cents]	ε
4.54	0.2
6.81	0.3
9.08	0.4
11.35	0.5
13.62	0.6
15.89	0.7
18.16	0.8
20.43	0.9



5-187 [*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)

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

and

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

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

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

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity is determined from

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

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

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

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

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

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

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

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

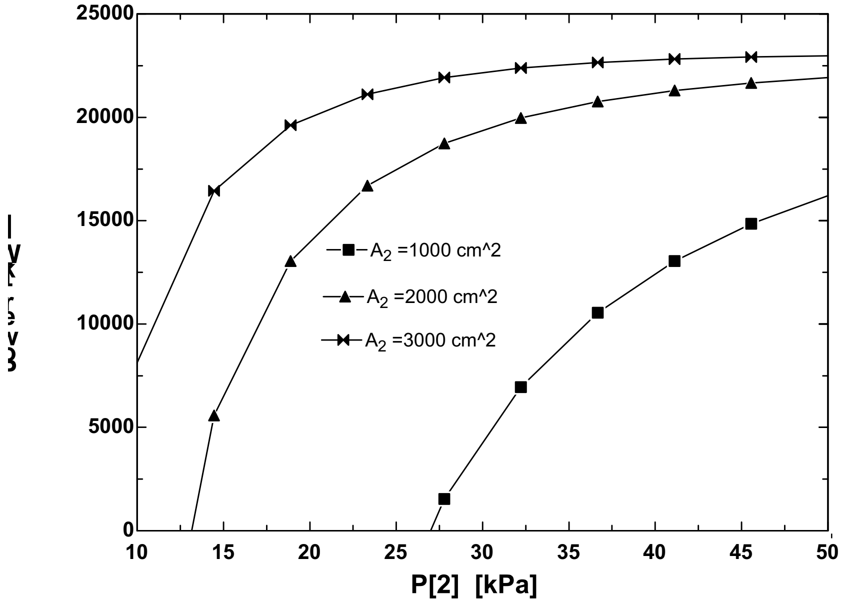
$$= \mathbf{10{,}330 \text{ kW}}$$

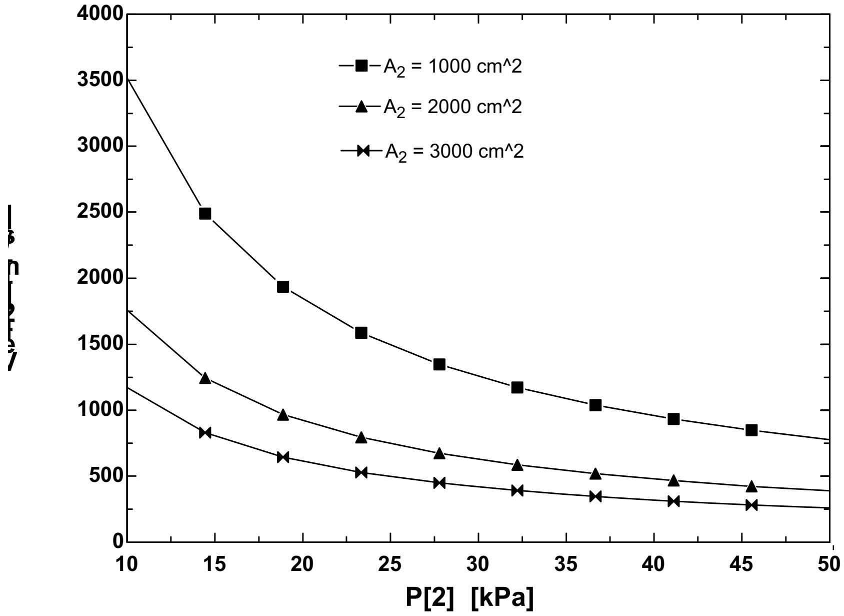
5-188 EES Problem 5-187 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 cm^2 to 3000 cm^2 is to be investigated. The exit velocity and the power output are to be plotted against the exit pressure for the exit areas of 1000, 2000, and 3000 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
P[2] [kPa]
A2 =1000 cm^2
A2 =2000 cm^2
A2 =3000 cm^2

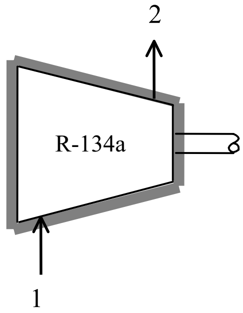

4000
3500
3000
2500
2000
1500
1000
500
0
10
15
20
25
30
35
40
45
50
P[2] [kPa]
A2 = 1000 cm^2
A2 = 2000 cm^2
A2 = 3000 cm^2

5-189E 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

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{10 \text{ ft}^3/\text{s}}{3.2551 \text{ ft}^3/\text{lbm}} = \mathbf{3.072 \text{ lbm/s}}$$

(*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

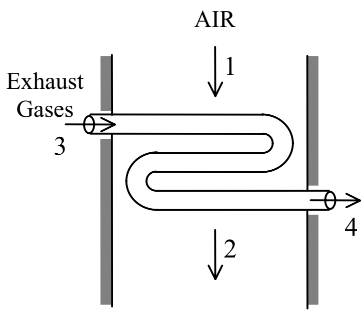
$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ h_2 = 117.87 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{95.7°F}$$

5-190 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

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

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

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

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

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,

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

$$\dot{m}_{exhaust} h_3 = \dot{Q}_{out} + \dot{m}_{exhaust} h_4 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}_{exhaust}(h_3 - h_4)$$

$$3200\text{ kJ/s} = \dot{m}_{exhaust}(821.95 - 607.02)\text{ kJ/kg}$$

It yields

$$\dot{m}_{exhaust} = \mathbf{14.9\ kg/s}$$

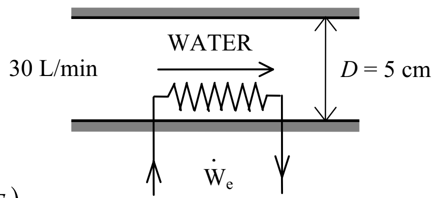
5-191 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

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

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



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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

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,

$$\dot{W}_{e,in} = \dot{m}c(T_2 - T_1) = (30/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(55 - 20)^\circ\text{C} = \mathbf{73.2 \text{ kW}}$$

(*b*) The average velocity of water through the pipe is determined from

$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}}{\pi r^2} = \frac{0.030 \text{ m}^3/\text{min}}{\pi(0.025 \text{ m})^2} = \mathbf{15.3 \text{ m/min}}$$

5-192 The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted seam the feedwater are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** 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 steam and feedwater at are (Tables A-4 through A-6)

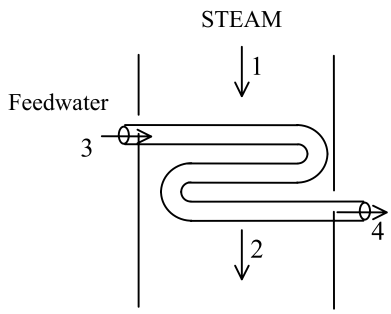
$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 200°\text{C} \end{array}\right\} h_1 = 2828.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_2 = h_{f@1\text{MPa}} = 762.51 \text{ kJ/kg} \\ T_2 = 179.9°\text{C} \end{array}$$

and

$$\left.\begin{array}{l} P_3 = 2.5 \text{ MPa} \\ T_3 = 50°\text{C} \end{array}\right\} h_3 \cong h_{f@50°\text{C}} = 209.34 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 2.5 \text{ MPa} \\ T_4 = T_2 - 10 \cong 170°\text{C} \end{array}\right\} h_4 \cong h_{f@170°\text{C}} = 718.55 \text{ kJ/kg}$$



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):

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

Energy balance (for the heat exchanger):

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

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

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

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

Dividing by $\dot{m}_{fw}$ and substituting,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_3 - h_4}{h_2 - h_1} = \frac{(718.55 - 209.34) \text{kJ/kg}}{(2828.3 - 762.51) \text{kJ/kg}} = \mathbf{0.246}$$

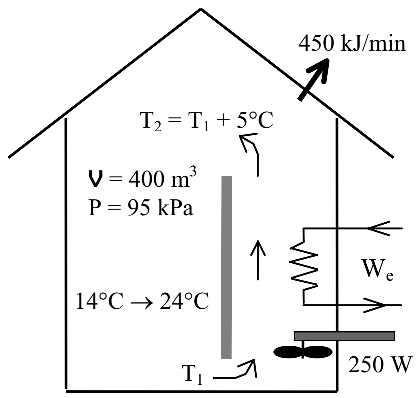
5-193 A building is to be heated by a 30-kW electric resistance heater placed in a duct inside. The time it takes to raise the interior temperature from 14°C to 24°C, and the average mass flow rate of air as it passes through the heater in the duct are to be determined.

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 building.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heats of air at room temperature are $c_p = 1.005$ and $c_v = 0.718$ kJ/kg·K (Table A-2).

Analysis (*a*) The total mass of air in the building is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95 \text{ kPa})(400 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(287 \text{ K})} = 461.3 \text{ kg} .$$



We first take the *entire building* as our system, which is a closed system since no mass leaks in or out. The time required to raise the air temperature to 24°C is determined by applying the energy balance to this constant volume closed system:

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

$$W_{\text{e,in}} + W_{\text{fan,in}} - Q_{\text{out}} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}\right) = mc_{v,\text{avg}}(T_2 - T_1)$$

Solving for Δt gives

$$\Delta t = \frac{mc_{v,\text{avg}}(T_2 - T_1)}{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}} = \frac{(461.3 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(24-14)^\circ\text{C}}{(30 \text{ kJ/s}) + (0.25 \text{ kJ/s}) - (450/60 \text{ kJ/s})} = \mathbf{146 \text{ s}}$$

(*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

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

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

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

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

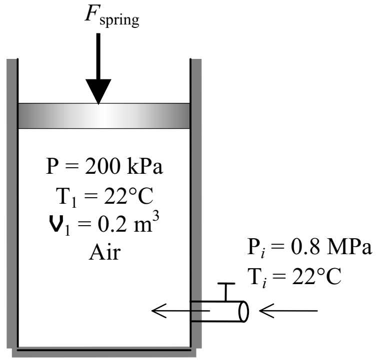
Thus,

$$\dot{m} = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{c_p \Delta T} = \frac{(30 + 0.25) \text{ kJ/s}}{(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(5^\circ\text{C})} = \mathbf{6.02 \text{ kg/s}}$$

5-194 [*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, $u = c_v T$ and $h = c_p T$.



Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $\quad m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

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

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

Combining the two relations, $\quad (m_2 - m_1)h_i = W_{b,out} + m_2 u_2 - m_1 u_1$

or, $\quad (m_2 - m_1)c_p T_i = W_{b,out} + m_2 c_v T_2 - m_1 c_v T_1$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(200 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 0.472 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(600 \text{ kPa})(0.4 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{836.2}{T_2}$$

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 $\quad T_2$ = **344.1 K**

Thus, $\quad m_2 = \dfrac{836.2}{\text{T}_2} = \dfrac{836.2}{344.1} = 2.430 \text{ kg}$

and $\quad m_i = m_2 - m_1 = 2.430 - 0.472 =$ **1.96 kg**

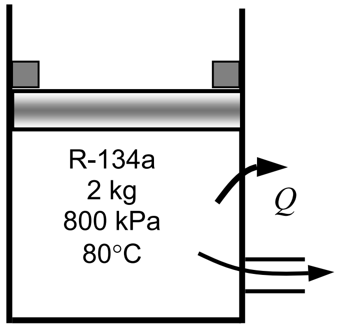
5-195 R-134a is allowed to leave a piston-cylinder device with a pair of stops. The work done and the heat transfer are to be determined.

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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

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

$$W_{\text{b,in}} - Q_{\text{out}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since ke} \cong \text{pe} \cong 0)$$

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

$$W_{\text{b,in}} = P_2(V_1 - V_2) = (500\text{ kPa})(0.06532 - 0.04212)\text{m}^3 = \mathbf{11.6\text{ kJ}}$$

(*b*) Substituting,

$$11.6\text{ kJ} - Q_{\text{out}} - (1\text{ kg})(290.21\text{ kJ/kg}) = (1\text{ kg})(242.40\text{ kJ/kg}) - (2\text{ kg})(290.84\text{ kJ/kg})$$

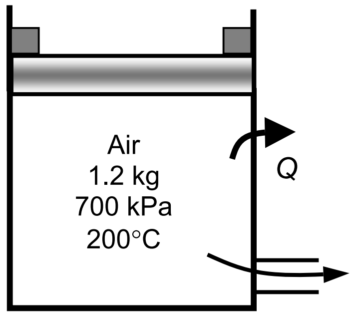
$$Q_{\text{out}} = \mathbf{60.7\text{ kJ}}$$

5-196 Air is allowed to leave a piston-cylinder device with a pair of stops. Heat is lost from the cylinder. The amount of mass that has escaped and the work done 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. 3 Air is an ideal gas with constant specific heats at the average temperature.

Properties The properties of air are R = 0.287 kPa.m^3/kg.K (Table A-1), c_v = 0.733 kJ/kg.K, c_p = 1.020 kJ/kg.K at the anticipated average temperature of 450 K (Table A-2b).

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: $m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:

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

$$W_{\text{b,in}} - Q_{\text{out}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since ke} \cong \text{pe} \cong 0)$$

or

$$W_{\text{b,in}} - Q_{\text{out}} - m_e C_p T_e = m_2 c_v T_2 - m_1 c_v T_1$$

The temperature of the air withdrawn from the cylinder is assumed to be the average of initial and final temperatures of the air in the cylinder. That is,

$$T_e = (1/2)(T_1 + T_2) = (1/2)(473 + T_2)$$

The volumes and the masses at the initial and final states and the mass that has escaped from the cylinder are given by

$$V_1 = \frac{m_1 R T_1}{P_1} = \frac{(1.2\text{ kg})(0.287\text{ kPa.m}^3\text{/kg.K})(200+273\text{ K})}{(700\text{ kPa})} = 0.2327\text{ m}^3$$

$$V_2 = 0.80 V_1 = (0.80)(0.2327) = 0.1862\text{ m}^3$$

$$m_2 = \frac{P_2 V_2}{R T_2} = \frac{(600\text{ kPa})(0.1862\text{ m}^3)}{(0.287\text{ kPa.m}^3\text{/kg.K})T_2} = \frac{389.18}{T_2}\text{ kg}$$

$$m_e = m_1 - m_2 = \left(1.2 - \frac{389.18}{T_2}\right)\text{kg}$$

Noting that the pressure remains constant after the piston starts moving, the boundary work is determined from

$$W_{\text{b,in}} = P_2(V_1 - V_2) = (600\text{ kPa})(0.2327 - 0.1862)\text{m}^3 = \mathbf{27.9\text{ kJ}}$$

Substituting,

$$27.9\text{ kJ} - 40\text{ kJ} - \left(1.2 - \frac{389.18}{T_2}\right)(1.020\text{ kJ/kg.K})(1/2)(473 + T_2)$$
$$= \left(\frac{389.18}{T_2}\right)(0.733\text{ kJ/kg.K})T_2 - (1.2\text{ kg})(0.733\text{ kJ/kg.K})(473\text{ K})$$

The final temperature may be obtained from this equation by a trial-error approach or using EES to be

$$T_2 = \mathbf{415.0\text{ K}}$$

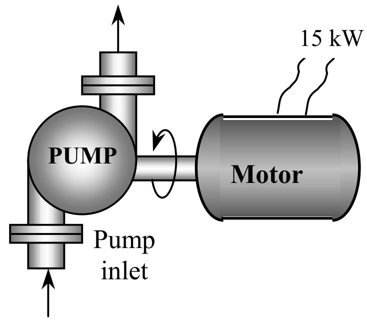
Then, the amount of mass that has escaped becomes

$$m_e = 1.2 - \frac{389.18}{415.0\text{ K}} = \mathbf{0.262\text{ kg}}$$

5-197 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,

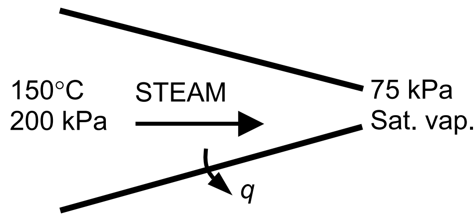
$$\Delta T = \frac{\dot{E}_{\text{mech,loss}}}{\dot{m}c} = \frac{3.5\text{ kW}}{(50\text{ kg/s})(4.18\text{ kJ/kg}\cdot\text{K})} = \mathbf{0.017°C}$$

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- 198 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:

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

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

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

or $$V_2 = \sqrt{2(h_1 - h_2 - q_{out})}$$

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

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} h_1 = 2769.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 75 \text{ kPa} \\ \text{sat. vap.} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 2.2172 \text{ m}^3\text{/kg} \\ h_2 = 2662.4 \text{ kJ/kg} \end{array}$$

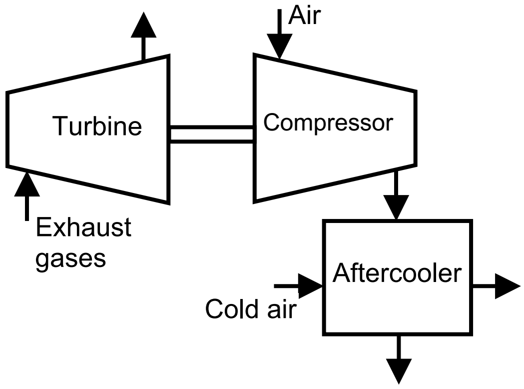
Substituting,

$$V_2 = \sqrt{2(h_1 - h_2 - q_{out})} = \sqrt{2(2769.1 - 2662.4 - 26)\text{kJ/kg}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)} = \mathbf{401.7 \text{ m/s}}$$

(*b*) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{2.2172 \text{ m}^3\text{/kg}}(0.001 \text{ m}^2)(401.7 \text{ m/s}) = \mathbf{0.181 \text{ kg/s}}$$

5-199 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

$$\dot{W}_{\mathrm{T}} = \dot{m}_{\mathrm{exh}} c_{p,\mathrm{exh}} \left(T_{\mathrm{exh},1} - T_{\mathrm{exh},2}\right) = (0.02\ \mathrm{kg/s})(1.063\ \mathrm{kJ/kg\cdot K})(400-350)\mathrm{K} = 1.063\ \mathrm{kW}$$

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

$$\dot{W}_{\mathrm{C}} = \dot{m}_{\mathrm{a}} c_{p,\mathrm{a}} (T_{\mathrm{a},2} - T_{\mathrm{a},1})$$

$$1.063\ \mathrm{kW} = (0.018\ \mathrm{kg/s})(1.008\ \mathrm{kJ/kg\cdot K})(T_{\mathrm{a},2} - 50)\mathrm{K} \longrightarrow T_{\mathrm{a},2} = \mathbf{108.6\ ^\circ C}$$

(*b*) An energy balance on the aftercooler gives the mass flow rate of cold ambient air

$$\dot{m}_{\mathrm{a}} c_{p,\mathrm{a}} (T_{\mathrm{a},2} - T_{\mathrm{a},3}) = \dot{m}_{\mathrm{ca}} c_{p,\mathrm{ca}} (T_{\mathrm{ca},2} - T_{\mathrm{ca},1})$$

$$(0.018\ \mathrm{kg/s})(1.008\ \mathrm{kJ/kg\cdot ^\circ C})(108.6 - 80)^\circ\mathrm{C} = \dot{m}_{\mathrm{ca}} (1.005\ \mathrm{kJ/kg\cdot ^\circ C})(40-30)^\circ\mathrm{C}$$

$$\dot{m}_{\mathrm{ca}} = 0.05161\ \mathrm{kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of aftercooler. That is,

$$v_{\mathrm{ca}} = \frac{RT}{P} = \frac{(0.287\ \mathrm{kJ/kg\cdot K})(30+273\ \mathrm{K})}{100\ \mathrm{kPa}} = 0.8696\ \mathrm{m^3/kg}$$

$$\dot{V}_{\mathrm{ca}} = \dot{m} v_{\mathrm{ca}} = (0.05161\ \mathrm{kg/s})(0.8696\ \mathrm{m^3/kg}) = \mathbf{0.0449\ m^3/s} = \mathbf{44.9\ L/s}$$

Fundamentals of Engineering (FE) Exam Problems

5-200 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"
```

5-201 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 (b) 50 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"
```

5-202 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"
```

5-203 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"
```

5-204 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"
```

5-205 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"
```

5-206 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"
```

5-207 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"
```

5-208 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"
```

5-209 Steam is compressed by an adiabatic compressor from 0.2 MPa and 150°C to 2500 MPa and 250°C at a rate of 1.30 kg/s. The power input to the compressor is
(a) 144 kW (b) 234 kW (c) 438 kW (d) 717 kW (e) 901 kW

Answer (a) 144 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).

```
"Note: This compressor violates the 2nd law. Changing State 2 to 800 kPa and 350C will correct
this problem (it would give 511 kW)"

P1=200 "kPa"
T1=150 "C"
P2=2500 "kPa"
T2=250 "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"
```

5-210 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"
```

5-211 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"
```

5-212 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"
```

5-213 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"
```

5-214 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"
```

5-215 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"
```

5-216 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"
```

5-217, 5-218 Design and Essay Problems
