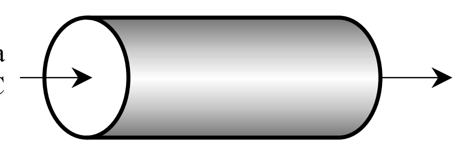
5-176 Saturated refrigerant-134a vapor at a saturation temperature of T_{sat} = 34°C condenses inside a tube. The rate of heat transfer from the refrigerant for the condensate exit temperatures of 34°C and 20°C are to be determined.

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



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

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

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

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

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.

$$\dot{Q}_{out} = (0.1 \text{ kg/min})(268.57 - 99.40) \text{ kJ/kg} = \mathbf{16.9 \text{ kg/min}}$$

Case 2: T_2 = 20°C: $h_2 \cong h_{f@20°C}$ =79.32 kJ/kg.

$$\dot{Q}_{out} = (0.1 \text{ kg/min})(268.57 - 79.32) \text{ kJ/kg} = \mathbf{18.9 \text{ kg/min}}$$

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).

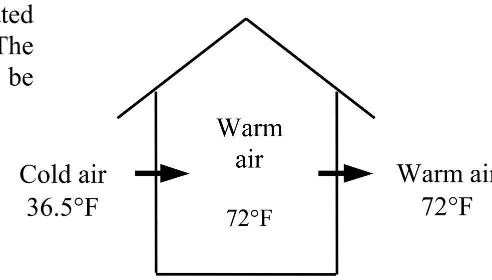
Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5\text{ psia}}{(0.3704\text{ psia.ft}^3/\text{lbm.R})(496.5\text{ R})} = 0.0734\text{ lbm/ft}^3$$

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



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

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

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

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

The reduction in the infiltration rate is 2.2 – 1.1 = 1.1 ACH. The reduction in the sensible infiltration heat load corresponding to it is

$$\begin{aligned}
\dot{Q}_{\text{infiltration, saved}} &= \rho_o c_p (ACH_{\text{saved}})(V_{\text{building}})(T_i - T_o) \\
&= (0.0734\text{ lbm/ft}^3)(0.24\text{ Btu/lbm.°F})(1.1/\text{h})(27{,}000\text{ ft}^3)(72\text{ - }36.5)\text{°F} \\
&= 18{,}573\text{ Btu/h} = 0.18573\text{ therm/h}
\end{aligned}$$

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.

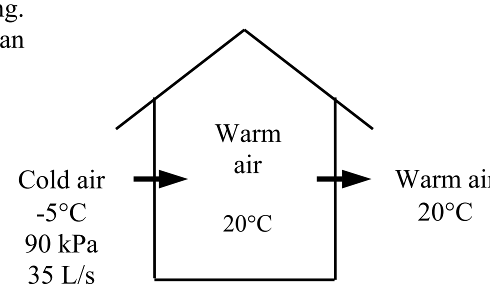
Assumptions **1** The house is maintained at 20°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 20°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies 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

$$\rho_o = \frac{P_o}{RT_o} = \frac{90\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(\text{-}5+273\text{ K})} = 1.17\text{ kg/m}^3$$

We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as



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

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

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

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

Then the sensible infiltration heat load corresponding to an infiltration rate of 35 L/s becomes

$$\begin{aligned}\dot{Q}_{\text{infiltration}} &= \rho_o \dot{\boldsymbol{V}}_{air} c_p (T_i - T_o) \\ &= (1.17\text{ kg/m}^3)(0.035\text{ m}^3/\text{s})(1.005\text{ kJ/kg.°C})[20 - (\text{-}5)]\text{°C} \\ &= \mathbf{1.029\ \ kW}\end{aligned}$$

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:

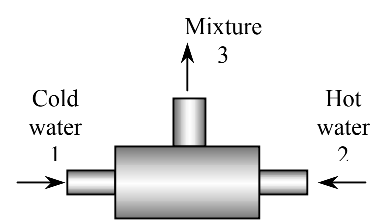
Mass balance:

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

$$\dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

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

$$\dot{V}_{saved} = [(13.3\text{-}10.5)\text{ L/min}](5\text{ min/person.day})(4\text{ persons})(365\text{ days/yr}) = 20{,}440\text{ L/year}$$

$$\dot{m}_{saved} = \rho \dot{V}_{saved} = (1\text{ kg/L})(20{,}440\text{ L/year}) = 20{,}440\text{ kg/year}$$

Then the energy saved per year becomes

$$\begin{aligned}\text{Energy saved} &= \dot{m}_{saved} c \Delta T = (20{,}440\text{ kg/year})(4.18\text{ kJ/kg.°C})(42\text{-}15)°\text{C} \\ &= 2{,}307{,}000\text{ kJ/year} \\ &= \mathbf{641\ kWh} \quad (\text{since } 1\text{ kWh} = 3600\text{ kJ})\end{aligned}$$

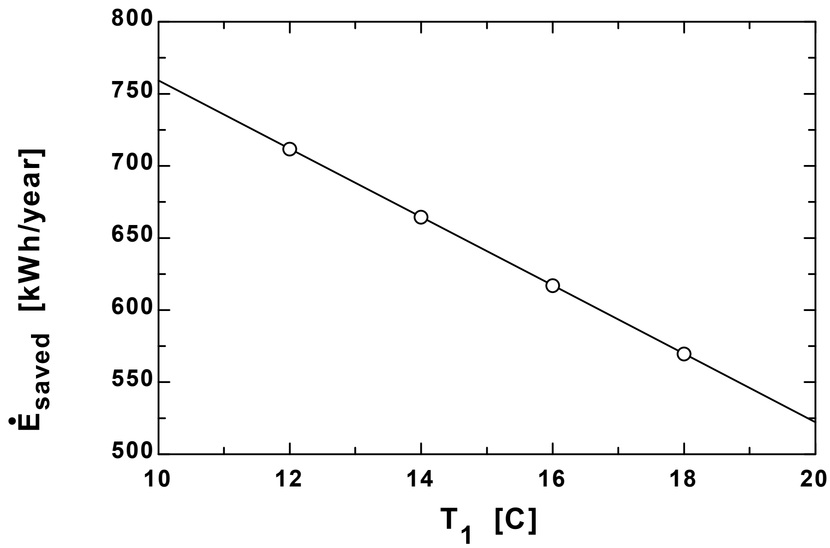
Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per year.

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).

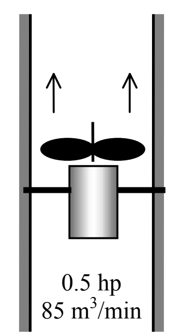
Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

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

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

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

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



Noting that the temperature and thus enthalpy remains constant, the relation above simplifies further to

$$\dot{W}_{e,in} = \dot{m}V_2^2/2$$

where

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1.18 \text{ kg/m}^3)(85 \text{ m}^3/\text{min}) = 100.3 \text{ kg/min} = 1.67 \text{ kg/s}$$

Solving for V_2 and substituting gives

$$V_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} = \sqrt{\frac{2(0.5 \text{ hp})}{1.67 \text{ kg/s}}\left(\frac{745.7 \text{ W}}{1 \text{ hp}}\right)\left(\frac{1 \text{ m}^2/\text{s}^2}{1 \text{ W}}\right)} = \mathbf{21.1 \text{ m/s}}$$

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.

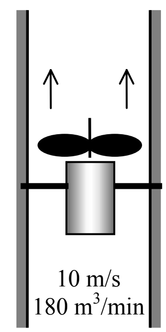
Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity is negligible, $V_1 \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be ρ = 1.20 kg/m³. 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 m³/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).

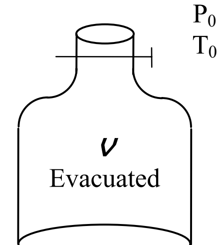
Analysis We take the bottle as the system. It is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be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

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

Energy balance:

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

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



Combining the two balances:

$$Q_{\text{in}} = m_2(u_2 - h_i) = m_2(c_v T_2 - c_p T_i)$$

But $T_i = T_2 = T_0$ and $c_p - c_v = R$. Substituting,

$$Q_{\text{in}} = m_2(c_v - c_p)T_0 = -m_2 R T_0 = -\frac{P_0\boldsymbol{V}}{RT_0}RT_0 = -P_0\boldsymbol{V}$$

Therefore, $\quad Q_{\text{out}} = \boldsymbol{P_0 V}$ (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)

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} h_3 = 3343.6 \text{ kJ/kg}$$

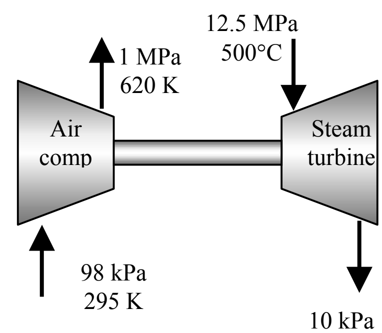
and

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg}$$

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 $\dot{m}_{in} = \dot{m}_{out} = \dot{m}$. We take either the turbine or the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for either steady-flow system can be expressed in the rate form as

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

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

For the turbine and the compressor it becomes

Compressor: $\dot{W}_{comp,\ in} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2 \quad \rightarrow \quad \dot{W}_{comp,\ in} = \dot{m}_{air}(h_2 - h_1)$

Turbine: $\dot{m}_{steam} h_3 = \dot{W}_{turb,\ out} + \dot{m}_{steam} h_4 \quad \rightarrow \quad \dot{W}_{turb,\ out} = \dot{m}_{steam}(h_3 - h_4)$

Substituting,

$$\dot{W}_{comp,in} = (10 \text{ kg/s})(628.07 - 295.17) \text{ kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{turb,out} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{ kJ/kg} = 23{,}777 \text{ kW}$$

Therefore,

$$\dot{W}_{net,out} = \dot{W}_{turb,out} - \dot{W}_{comp,in} = 23{,}777 - 3329 = \mathbf{20{,}448 \text{ kW}}$$

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

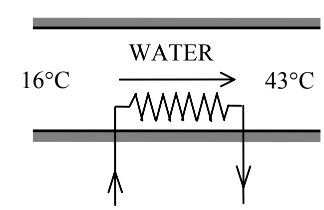
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \rightarrow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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v(P_2 - P_1)^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

where

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1 \text{ kg/L})(10 \text{ L/min}) = 10 \text{ kg/min}$$



Substituting,

$$\dot{W}_{e,in} = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(43 - 16)^\circ\text{C} = \mathbf{18.8\ kW}$$

The energy recovered by the heat exchanger is

$$\begin{aligned}\dot{Q}_{saved} &= \varepsilon \dot{Q}_{max} = \varepsilon \dot{m} C(T_{max} - T_{min}) \\ &= 0.5(10/60 \text{ kg/s})(4.18 \text{ kJ/kg.}^\circ\text{C})(39 - 16)^\circ\text{C} \\ &= 8.0 \text{ kJ/s} = 8.0\text{kW}\end{aligned}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot{W}_{in,new} = \dot{W}_{in,old} - \dot{Q}_{saved} = 18.8 - 8.0 = \mathbf{10.8\ kW}$$

The money saved during a 10-min shower as a result of installing this heat exchanger is

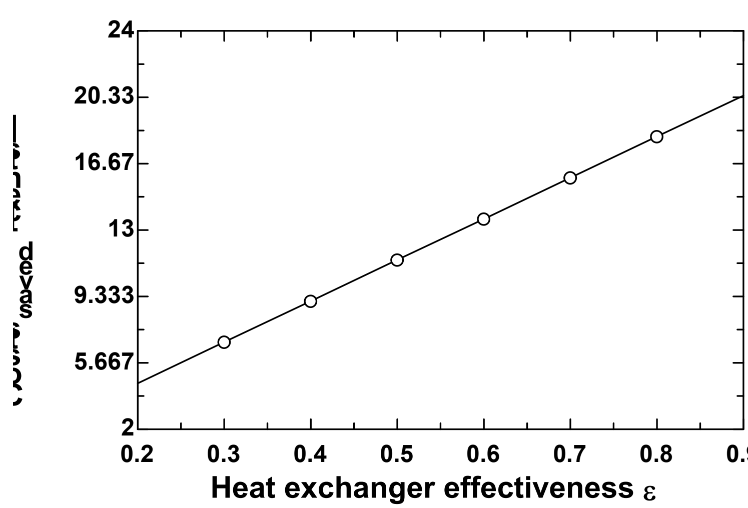
$$(8.0 \text{ kW})(10/60 \text{ h})(8.5 \text{ cents/kWh}) = \mathbf{11.3\ cents}$$

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.

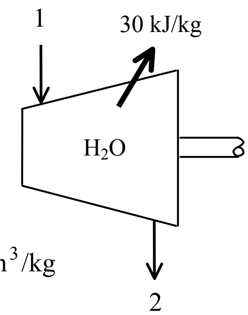
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"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

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

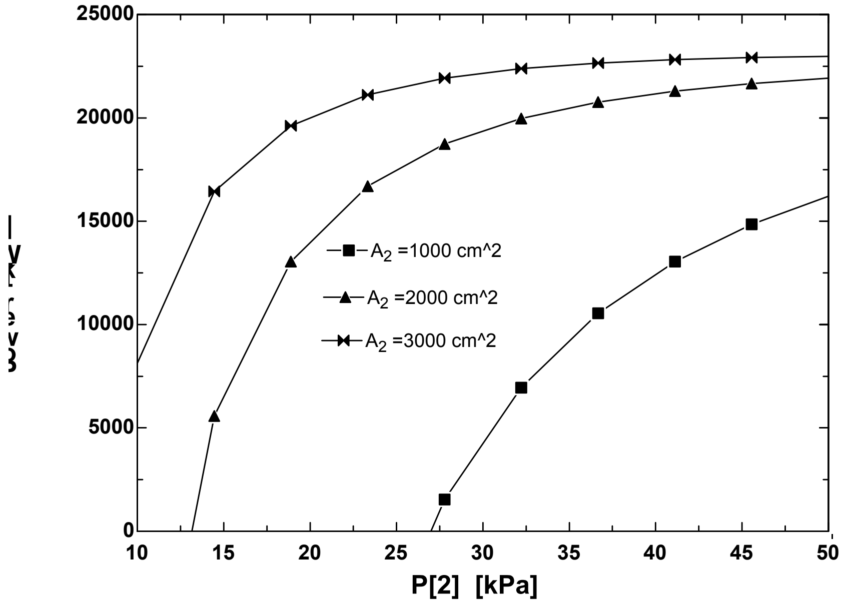
$$= \mathbf{10{,}330 \text{ kW}}$$

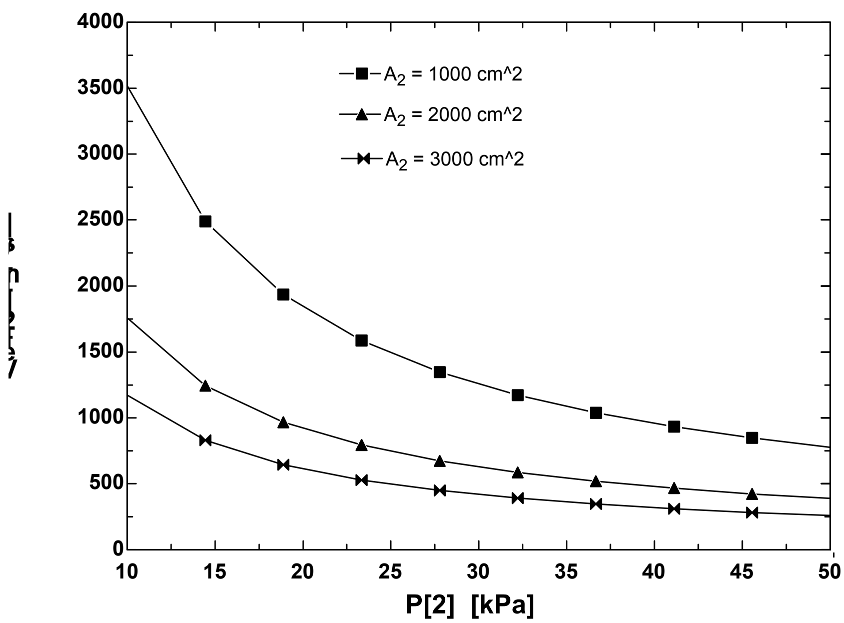
5-188 EES Problem 5-187 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 cm^2 to 3000 cm^2 is to be investigated. 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

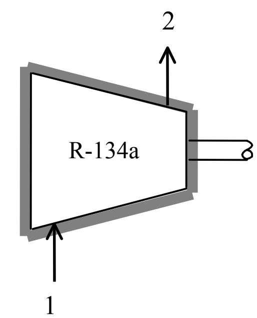

4000
3500
3000
2500
2000
1500
1000
500
0
10
15
20
25
30
35
40
45
50
P[2] [kPa]
A2 = 1000 cm^2
A2 = 2000 cm^2
A2 = 3000 cm^2

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

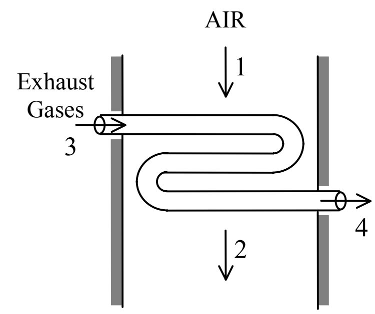
$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ h_2 = 117.87 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{95.7°F}$$

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

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

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

Substituting,

$$3200\text{ kJ/s} = (800/60\text{ kg/s})(h_2 - 554.71\text{ kJ/kg}) \rightarrow h_2 = 794.71\text{ kJ/kg}$$

Then from Table A-17 we read $\quad T_2 = \mathbf{775.1\ K}$

(*b*) Treating the exhaust gases as an ideal gas, the mass flow rate of the exhaust gases is determined from the steady-flow energy relation applied only to the exhaust gases,

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}_{\text{exhaust}} h_3 = \dot{Q}_{\text{out}} + \dot{m}_{\text{exhaust}} h_4 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{Q}_{\text{out}} = \dot{m}_{\text{exhaust}}(h_3 - h_4)$$
$$3200\text{ kJ/s} = \dot{m}_{\text{exhaust}}(821.95 - 607.02)\text{ kJ/kg}$$

It yields

$$\dot{m}_{\text{exhaust}} = \mathbf{14.9\ kg/s}$$

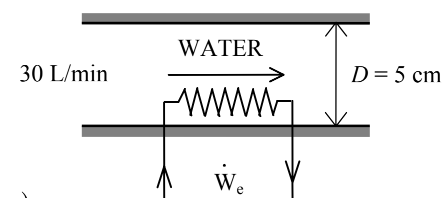
5-191 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

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/m^3 and c = 4.18 kJ/kg·°C (Table A-3).

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

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



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

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

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

The mass flow rate of water through the pipe is

$$\dot{m} = \rho \dot{\boldsymbol{V}}_1 = (1000 \text{ kg/m}^3)(0.030 \text{ m}^3/\text{min}) = 30 \text{ kg/min}$$

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

$$V_1 = \frac{\dot{\boldsymbol{V}}_1}{A_1} = \frac{\dot{\boldsymbol{V}}}{\pi r^2} = \frac{0.030 \text{ m}^3/\text{min}}{\pi(0.025 \text{ m})^2} = \mathbf{15.3 \text{ m/min}}$$

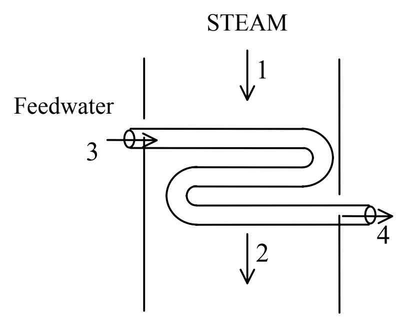
5-192 The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted seam the feedwater are to be determined.

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,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_3 - h_4}{h_2 - h_1} = \frac{(718.55 - 209.34)\text{kJ/kg}}{(2828.3 - 762.51)\text{kJ/kg}} = \mathbf{0.246}$$

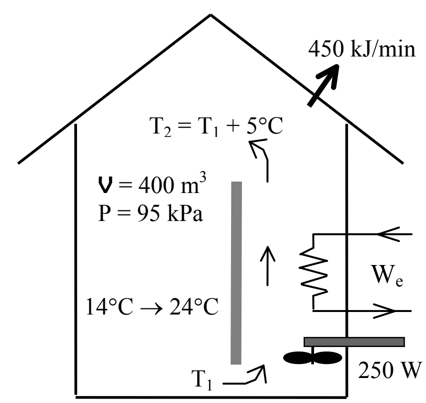
5-193 A building is to be heated by a 30-kW electric resistance heater placed in a duct inside. The time it takes to raise the interior temperature from 14°C to 24°C, and the average mass flow rate of air as it passes through the heater in the duct are to be determined.

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

$$\Delta t = \frac{mc_{v,\text{avg}}(T_2 - T_1)}{\dot W_{\text{e,in}} + \dot W_{\text{fan,in}} - \dot Q_{\text{out}}} = \frac{(461.3\text{ kg})(0.718\text{ kJ/kg}\cdot^\circ\text{C})(24-14)^\circ\text{C}}{(30\text{ kJ/s}) + (0.25\text{ kJ/s}) - (450/60\text{ kJ/s})} = \mathbf{146\ \ s}$$

(*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

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

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

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

$$\dot W_{\text{e,in}} + \dot W_{\text{fan,in}} = \dot m(h_2 - h_1) = \dot m c_p (T_2 - T_1)$$

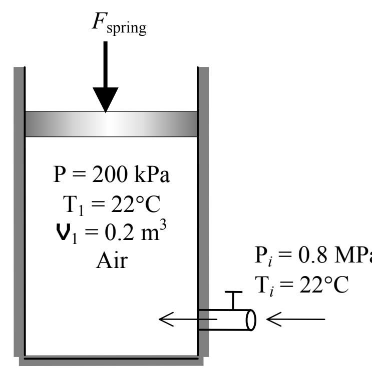
Thus,

$$\dot m = \frac{\dot W_{\text{e,in}} + \dot W_{\text{fan,in}}}{c_p \Delta T} = \frac{(30 + 0.25)\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot^\circ\text{C})(5^\circ\text{C})} = \mathbf{6.02\ kg/s}$$

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

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

or, $$(m_2 - m_1)c_p T_i = W_{b,out} + m_2 c_v T_2 - m_1 c_v T_1$$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(200 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 0.472 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(600 \text{ kPa})(0.4 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{836.2}{T_2}$$

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

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

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

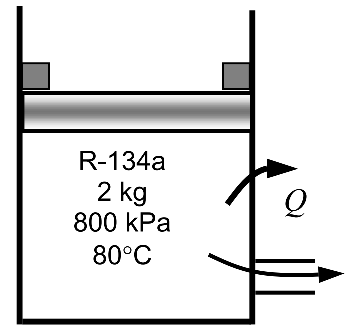
It yields T_2 = **344.1 K**

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

and $m_i = m_2 - m_1 = 2.430 - 0.472 =$ **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)

$$\left.\begin{array}{l} P_1 = 800\text{ kPa} \\ T_1 = 80°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.032659\text{ m}^3/\text{kg} \\ u_1 = 290.84\text{ kJ/kg} \\ h_1 = 316.97\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 500\text{ kPa} \\ T_2 = 20°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.042115\text{ m}^3/\text{kg} \\ u_2 = 242.40\text{ kJ/kg} \\ h_2 = 263.46\text{ kJ/kg} \end{array}$$

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

$$\boldsymbol{V}_1 = m_1 \boldsymbol{v}_1 = (2\text{ kg})(0.032659\text{ m}^3/\text{kg}) = 0.06532\text{ m}^3$$

$$\boldsymbol{V}_2 = m_2 \boldsymbol{v}_2 = (1/2) m_1 \boldsymbol{v}_2 = (1/2)(2\text{ kg})(0.042115\text{ m}^3/\text{kg}) = 0.04212\text{ m}^3$$

$$m_e = m_1 - m_2 = 2 - 1 = 1\text{ kg}$$

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

$$W_{\text{b,in}} = P_2(\boldsymbol{V}_1 - \boldsymbol{V}_2) = (500\text{ kPa})(0.06532 - 0.04212)\text{m}^3 = \mathbf{11.6\text{ kJ}}$$

(*b*) Substituting,

$$11.6\text{ kJ} - Q_{\text{out}} - (1\text{ kg})(290.21\text{ kJ/kg}) = (1\text{ kg})(242.40\text{ kJ/kg}) - (2\text{ kg})(290.84\text{ kJ/kg})$$

$$Q_{\text{out}} = \mathbf{60.7\text{ kJ}}$$

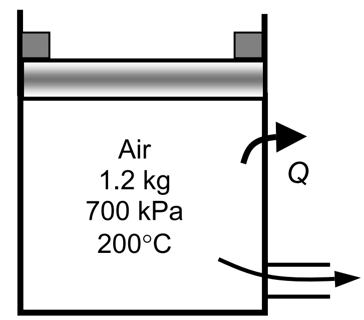
5-196 Air is allowed to leave a piston-cylinder device with a pair of stops. Heat is lost from the cylinder. The amount of mass that has escaped and the work done are to be determined.

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

$$W_{\text{b,in}} = P_2(V_1 - V_2) = (600\text{ kPa})(0.2327 - 0.1862)\text{m}^3 = \mathbf{27.9\ kJ}$$

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

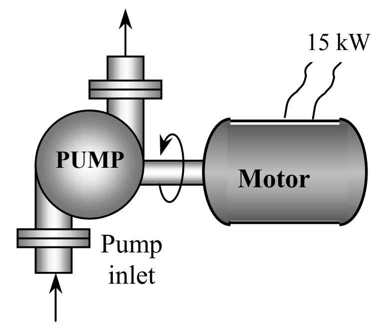
$T_2 =$ **415.0 K**

Then, the amount of mass that has escaped becomes

$$m_e = 1.2 - \frac{389.18}{415.0\text{ K}} = \mathbf{0.262\ kg}$$

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

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1\,\text{kg/L})(50\,\text{L/s}) = 50\,\text{kg/s}$$

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,

$$\dot{E}_{\text{mech,loss}} = \dot{m}(u_2 - u_1) = \dot{m}c\Delta T$$

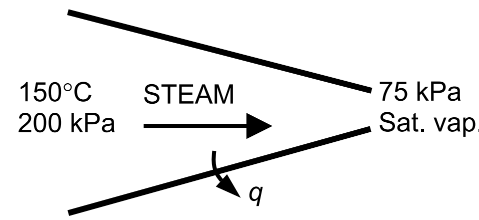
Solving for ΔT,

$$\Delta T = \frac{\dot{E}_{\text{mech,loss}}}{\dot{m}c} = \frac{3.5\,\text{kW}}{(50\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot\text{K})} = \mathbf{0.017^\circ C}$$

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:

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

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

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

or $\quad V_2 = \sqrt{2(h_1 - h_2 - q_{\text{out}})}$

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

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} h_1 = 2769.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 75 \text{ kPa} \\ \text{sat. vap.} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 2.2172 \text{ m}^3/\text{kg} \\ h_2 = 2662.4 \text{ kJ/kg} \end{array}$$

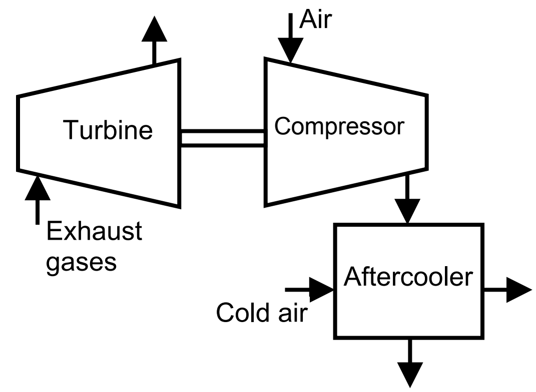
Substituting,

$$V_2 = \sqrt{2(h_1 - h_2 - q_{\text{out}})} = \sqrt{2(2769.1 - 2662.4 - 26)\text{kJ/kg}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)} = \mathbf{401.7 \text{ m/s}}$$

(*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

$$\dot{W}_{\rm T} = \dot{m}_{\rm exh} c_{p,\rm exh}\left(T_{\rm exh,1} - T_{\rm exh,2}\right) = (0.02 \text{ kg/s})(1.063 \text{ kJ/kg}\cdot\text{K})(400-350)\text{K} = 1.063 \text{ kW}$$

This is also the power input to the compressor since the mechanical efficiency between the turbine and the compressor is assumed to be 100%. An energy balance on the compressor gives the air temperature at the compressor outlet

$$\dot{W}_{\rm C} = \dot{m}_{\rm a} c_{p,\rm a}(T_{\rm a,2} - T_{\rm a,1})$$

$$1.063 \text{ kW} = (0.018 \text{ kg/s})(1.008 \text{ kJ/kg}\cdot\text{K})(T_{\rm a,2} - 50)\text{K} \longrightarrow T_{\rm a,2} = \mathbf{108.6\ °C}$$

(*b*) An energy balance on the aftercooler gives the mass flow rate of cold ambient air

$$\dot{m}_{\rm a} c_{p,\rm a}(T_{\rm a,2} - T_{\rm a,3}) = \dot{m}_{\rm ca} c_{p,\rm ca}(T_{\rm ca,2} - T_{\rm ca,1})$$

$$(0.018 \text{ kg/s})(1.008 \text{ kJ/kg}\cdot°\text{C})(108.6-80)°\text{C} = \dot{m}_{\rm ca}(1.005 \text{ kJ/kg}\cdot°\text{C})(40-30)°\text{C}$$

$$\dot{m}_{\rm ca} = 0.05161 \text{ kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of aftercooler. That is,

$$\boldsymbol{v}_{\rm ca} = \frac{RT}{P} = \frac{(0.287 \text{ kJ/kg}\cdot\text{K})(30+273\text{ K})}{100 \text{ kPa}} = 0.8696 \text{ m}^3/\text{kg}$$

$$\dot{\boldsymbol{V}}_{\rm ca} = \dot{m}\boldsymbol{v}_{\rm ca} = (0.05161 \text{ kg/s})(0.8696 \text{ m}^3/\text{kg}) = \mathbf{0.0449\ m^3/s} = \mathbf{44.9\ L/s}$$

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