8-24 Air is compressed steadily by a compressor. The air temperature is maintained constant by heat rejection to the surroundings. The rate of entropy change of air is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas. **4** The process involves no internal irreversibilities such as friction, and thus it is an isothermal, internally reversible process.

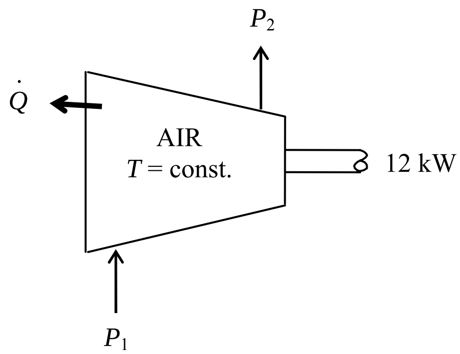
Properties Noting that $h = h(T)$ for ideal gases, we have $h_1 = h_2$ since $T_1 = T_2 = 25°C$.

Analysis We take the compressor as the system. Noting that the enthalpy of air remains constant, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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



Therefore,

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

Noting that the process is assumed to be an isothermal and internally reversible process, the rate of entropy change of air is determined to be

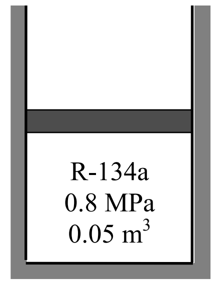
$$\Delta \dot{S}_{air} = -\frac{\dot{Q}_{out,air}}{T_{sys}} = -\frac{12 \text{ kW}}{298 \text{ K}} = \mathbf{-0.0403 \text{ kW/K}}$$

8-44 An insulated cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant expands in a reversible manner until the pressure drops to a specified value. The final temperature in the cylinder and the work done by the refrigerant are to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

Analysis (*a*) This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 0.8 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@0.8 \text{ MPa}} = 0.025621 \text{ m}^3\text{/kg} \\ u_1 = u_{g@0.8 \text{ MPa}} = 246.79 \text{ kJ/kg} \\ s_1 = s_{g@0.8 \text{ MPa}} = 0.91835 \text{ kJ/kg} \cdot \text{K} \end{array}$$



Also,

$$m = \frac{V}{v_1} = \frac{0.05 \text{ m}^3}{0.025621 \text{ m}^3\text{/kg}} = 1.952 \text{ kg}$$

and

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

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

(*b*) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this adiabatic closed system can be expressed as

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

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

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

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

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

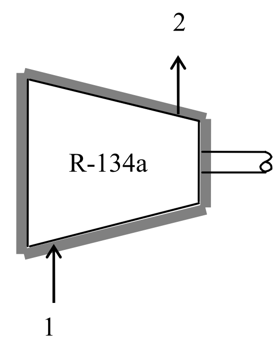
8-46 Saturated Refrigerant-134a vapor at 160 kPa is compressed steadily by an adiabatic compressor. The minimum power input to the compressor is to be determined.

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

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

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

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



Also,

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

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

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

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

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

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

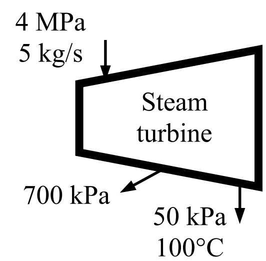
Substituting, the minimum power supplied to the compressor is determined to be

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

8-60 Steam is expanded in an isentropic turbine. The work produced is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic).

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



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

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

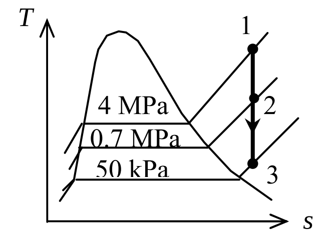
$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{out}$$

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

From a mass balance,

$$\dot{m}_2 = 0.05\dot{m}_1 = (0.05)(5 \text{ kg/s}) = 0.25 \text{ kg/s}$$

$$\dot{m}_3 = 0.95\dot{m}_1 = (0.95)(5 \text{ kg/s}) = 4.75 \text{ kg/s}$$



Noting that the expansion process is isentropic, the enthalpies at three states are determined as follows:

$$\left.\begin{array}{l} P_3 = 50 \text{ kPa} \\ T_3 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 2682.4 \text{ kJ/kg} \\ s_3 = 7.6953 \text{ kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A-6)}$$

$$\left.\begin{array}{l} P_1 = 4 \text{ MPa} \\ s_1 = s_3 = 7.6953 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} h_1 = 3979.3 \text{ kJ/kg} \quad \text{(Table A-6)}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ s_2 = s_3 = 7.6953 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} h_2 = 3309.1 \text{ kJ/kg} \quad \text{(Table A-6)}$$

Substituting,

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

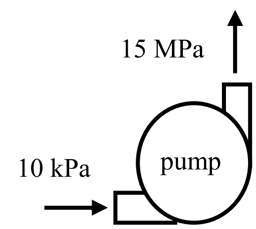
8-73 An adiabatic pump is used to compress saturated liquid water in a reversible manner. The work input is to be determined by different approaches.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible.

Analysis The properties of water at the inlet and exit of the pump are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 191.81 \text{ kJ/kg} \\ s_1 = 0.6492 \text{ kJ/kg} \\ \boldsymbol{v}_1 = 0.001010 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 15 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} h_2 = 206.90 \text{ kJ/kg} \\ \boldsymbol{v}_2 = 0.001004 \text{ m}^3\text{/kg} \end{array}$$



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

$$w_\text{P} = h_2 - h_1 = 206.90 - 191.81 = \mathbf{15.10\ kJ/kg}$$

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

$$w_\text{P} = \boldsymbol{v}_1 (P_2 - P_1) = (0.001010 \text{ m}^3\text{/kg})(15{,}000 - 10)\text{kPa} = \mathbf{15.14\ kJ/kg}$$

Error = **0.3%**

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

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

Error = **0%**

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

8-99 Nitrogen is compressed in an adiabatic compressor. The minimum work input is to be determined.

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

Properties The properties of nitrogen at an anticipated average temperature of 400 K are c_p = 1.044 kJ/kg·K and k = 1.397 (Table A-2*b*).

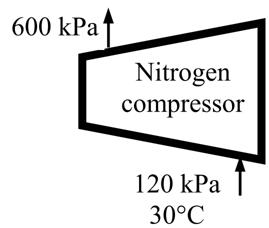
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

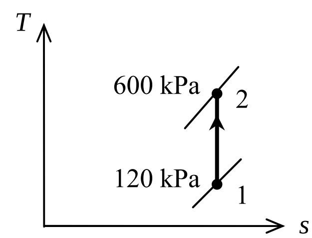
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

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

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

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





For the minimum work input to the compressor, the process must be reversible as well as adiabatic (i.e., isentropic). This being the case, the exit temperature will be

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

Substituting into the energy balance equation gives

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

8-128 Steam is expanded in an adiabatic turbine with an isentropic efficiency of 0.92. The power output of the turbine is to be determined.

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

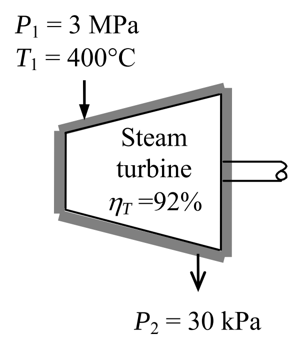
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



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

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

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

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

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

8-132 Steam enters an adiabatic turbine at a specified state, and leaves at a specified state. The mass flow rate of the steam and the isentropic efficiency are to be determined.

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

Analysis (*a*) From the steam tables (Tables A-4 and A-6),

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

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

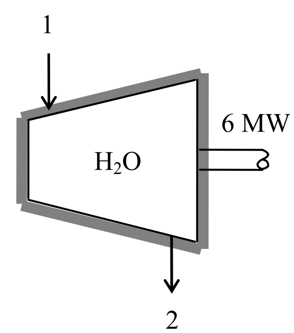
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



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

$$6000 \text{ kJ/s} = -\dot{m}\left(2780.2 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = \mathbf{6.95 \ kg/s}$$

(*b*) The isentropic exit enthalpy of the steam and the power output of the isentropic turbine are

$$\left.\begin{array}{l} P_{2s} = 50 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{7.0910 - 1.0912}{6.5019} = 0.9228 \\ h_{2s} = h_f + x_{2s} h_{fg} = 340.54 + (0.9228)(2304.7) = 2467.3 \text{ kJ/kg} \end{array}$$

and

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

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

$$= 8174 \text{ kW}$$

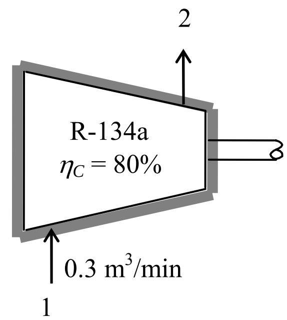
Then the isentropic efficiency of the turbine becomes

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

8-135 CD EES Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The compressor exit temperature and power input to the compressor are to be determined.

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

Analysis (a) From the refrigerant tables (Tables A-11E through A-13E),



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

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

From the isentropic efficiency relation,

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

Thus,

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

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

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

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

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

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

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

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

Substituting, the power input to the compressor becomes,

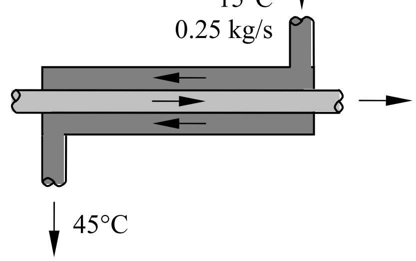
$$\dot{W}_{a,in} = (0.0308 \text{ kg/s})(292.26 - 236.97)\text{kJ/kg} = \mathbf{1.70 \text{ kW}}$$

8-148 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the rate of entropy generation within the heat exchanger are to be determined.

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

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

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



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

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

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

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

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

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

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

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

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

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

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

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

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

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

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

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

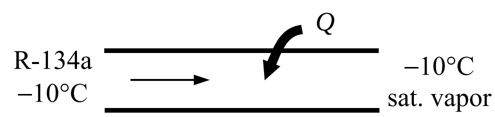
$$= \mathbf{0.0190 \ kW/K}$$

8-152 In an ice-making plant, water is frozen by evaporating saturated R-134a liquid. The rate of entropy generation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis We take the control volume formed by the R-134a evaporator with a single inlet and single exit as the system. The rate of entropy generation within this evaporator during this process can be determined by applying the rate form of the entropy balance on the system. The entropy balance for this steady-flow system can be expressed as

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



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

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

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

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

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

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

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

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

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

Substituting,

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

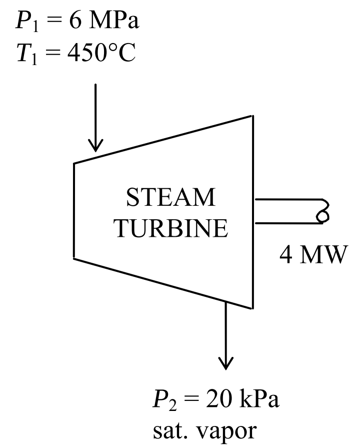
8-166 Steam expands in a turbine from a specified state to another specified state. The rate of entropy generation during this process is to be determined.

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

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

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

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



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

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

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

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

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

Substituting,

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

The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the turbine and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

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

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

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

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

8-168 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of entropy generation are to be determined.

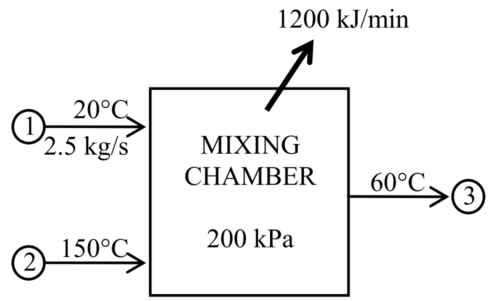
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Properties Noting that $T < T_{sat\,@\,200\,kPa} = 120.21°C$, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From Tables A-4 through A-6,

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

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

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



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

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

Energy balance:

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

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

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

Combining the two relations gives $\dot{Q}_{out} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3 = \dot{m}_1(h_1 - h_3) + \dot{m}_2(h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

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

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 2.5 + 0.166 = 2.666\text{ kg/s}$

(*b*) The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

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

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

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

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

8-189 An insulated rigid tank is connected to a piston-cylinder device with zero clearance that is maintained at constant pressure. A valve is opened, and some steam in the tank is allowed to flow into the cylinder. The final temperatures in the tank and the cylinder are to be determined.

Assumptions **1** Both the tank and cylinder are well-insulated and thus heat transfer is negligible. **2** The water that remains in the tank underwent a reversible adiabatic process. **3** The thermal energy stored in the tank and cylinder themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible.

Analysis (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 500 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = v_{g@500 \text{ kPa}} = 0.37483 \text{ m}^3/\text{kg} \\ u_1 = u_{g@500 \text{ kPa}} = 2560.7 \text{ kJ/kg} \\ s_1 = s_{g@500 \text{ kPa}} = 6.8207 \text{ kJ/kg}\cdot\text{K} \end{array}$$

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

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

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

The initial and the final masses in tank A are

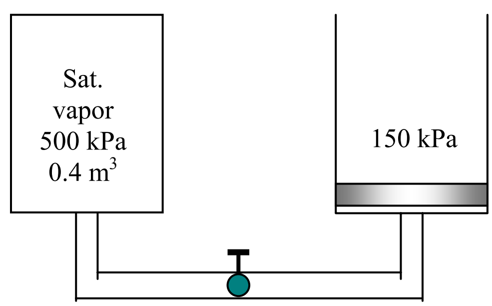
$$m_{1,A} = \frac{V_A}{v_{1,A}} = \frac{0.4 \text{ m}^3}{0.37483 \text{ m}^3/\text{kg}} = 1.067 \text{ kg} \quad \text{and} \quad m_{2,A} = \frac{V_A}{v_{2,A}} = \frac{0.4 \text{ m}^3}{1.0789 \text{ m}^3/\text{kg}} = 0.371 \text{ kg}$$

Thus,

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

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

$$W_{b,out} = \int_1^2 P\,dV = P_B(V_{2,B} - 0) = P_B m_{2,B} v_{2,B}$$



Taking the contents of both the tank and the cylinder to be the system, the energy balance for this closed system can be expressed as

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

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

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

or,

$$P_B m_{2,B} v_{2,B} + (m_2 u_2 - m_1 u_1)_A + (m_2 u_2)_B = 0$$

$$m_{2,B} h_{2,B} + (m_2 u_2 - m_1 u_1)_A = 0$$

Thus,

$$h_{2,B} = \frac{(m_1 u_1 - m_2 u_2)_A}{m_{2,B}} = \frac{(1.067)(2560.7) - (0.371)(2376.6)}{0.696} = 2658.8 \text{ kJ/kg}$$

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

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