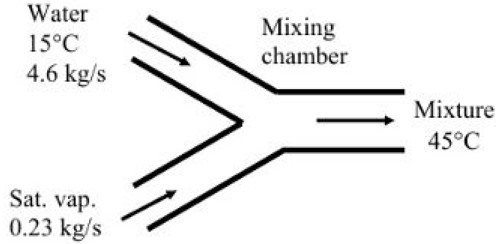
8-92 Water is heated in a chamber by mixing it with saturated steam. The temperature of the steam entering the chamber, the exergy destruction, and the second-law efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the chamber is negligible.

Analysis (*a*) The properties of water are (Tables A-4 through A-6)



$$\left.\begin{array}{l} T_1 = 15°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = h_0 = 62.98 \text{ kJ/kg} \\ s_1 = s_0 = 0.22447 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_3 = 45°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_3 = 188.44 \text{ kJ/kg} \\ s_3 = 0.63862 \text{ kJ/kg.K} \end{array}$$

An energy balance on the chamber gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$(4.6 \text{ kg/s})(62.98 \text{ kJ/kg}) + (0.23 \text{ kg/s}) h_2 = (4.6 + 0.23 \text{ kg/s})(188.44 \text{ kJ/kg})$$

$$h_2 = 2697.5 \text{ kJ/kg}$$

The remaining properties of the saturated steam are

$$\left.\begin{array}{l} h_2 = 2697.5 \text{ kJ/kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{114.3°C} \\ s_2 = 7.1907 \text{ kJ/kg.K} \end{array}$$

(*b*) The specific exergy of each stream is

$$\psi_1 = 0$$

$$\psi_2 = h_2 - h_0 - T_0(s_2 - s_0)$$
$$= (2697.5 - 62.98)\text{kJ/kg} - (15 + 273 \text{ K})(7.1907 - 0.22447)\text{kJ/kg.K} = 628.28 \text{ kJ/kg}$$

$$\psi_3 = h_3 - h_0 - T_0(s_3 - s_0)$$
$$= (188.44 - 62.98)\text{kJ/kg} - (15 + 273 \text{ K})(0.63862 - 0.22447)\text{kJ/kg.K} = 6.18 \text{ kJ/kg}$$

The exergy destruction is determined from an exergy balance on the chamber to be

$$\dot{X}_{\text{dest}} = \dot{m}_1 \psi_1 + \dot{m}_2 \psi_2 - (\dot{m}_1 + \dot{m}_2)\psi_3$$
$$= 0 + (0.23 \text{ kg/s})(628.28 \text{ kJ/kg}) - (4.6 + 0.23 \text{ kg/s})(6.18 \text{ kJ/kg})$$
$$= \mathbf{114.7 \text{ kW}}$$

(*c*) The second-law efficiency for this mixing process may be determined from

$$\eta_{\text{II}} = \frac{(\dot{m}_1 + \dot{m}_2)\psi_3}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} = \frac{(4.6 + 0.23 \text{ kg/s})(6.18 \text{ kJ/kg})}{0 + (0.23 \text{ kg/s})(628.28 \text{ kJ/kg})} = \mathbf{0.207}$$

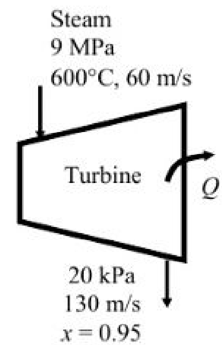
8-90 Steam expands in a turbine, which is not insulated. The reversible power, the exergy destroyed, the second-law efficiency, and the possible increase in the turbine power if the turbine is well insulated are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Potential energy change is negligible.

Analysis (*a*) The properties of the steam at the inlet and exit of the turbine are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 9\text{ MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3634.1\text{ kJ/kg} \\ s_1 = 6.9605\text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 20\text{ kPa} \\ x_2 = 0.95 \end{array}\right\} \begin{array}{l} h_2 = 2491.1\text{ kJ/kg} \\ s_2 = 7.5535\text{ kJ/kg.K} \end{array}$$



The enthalpy at the dead state is

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x = 0 \end{array}\right\} h_0 = 104.83\text{ kJ/kg}$$

The mass flow rate of steam may be determined from an energy balance on the turbine

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

$$\dot{m}\left[3634.1\text{ kJ/kg} + \frac{(60\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right] = \dot{m}\left[2491.1\text{ kJ/kg} + \frac{(130\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right]$$

$$+220\text{ kW} + 4500\text{ kW} \longrightarrow \dot{m} = 4.137\text{ kg/s}$$

The reversible power may be determined from

$$\dot{W}_{\text{rev}} = \dot{m}\left[h_1 - h_2 - T_0(s_1 - s_2) + \frac{V_1^2 - V_2^2}{2}\right]$$

$$= (2.693)\left[(3634.1 - 2491.1) - (298)(6.9605 - 7.5535) + \frac{(60\text{ m/s})^2 - (130\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{5451\text{ kW}}$$

(b) The exergy destroyed in the turbine is

$$\dot{X}_{\text{dest}} = \dot{W}_{\text{rev}} - \dot{W}_{\text{a}} = 5451 - 4500 = \mathbf{951\text{ kW}}$$

(*c*) The second-law efficiency is

$$\eta_{II} = \frac{\dot{W}_{\text{a}}}{\dot{W}_{\text{rev}}} = \frac{4500\text{ kW}}{5451\text{ kW}} = \mathbf{0.826}$$

(*d*) The energy of the steam at the turbine inlet in the given dead state is

$$\dot{Q} = \dot{m}(h_1 - h_0) = (4.137\text{ kg/s})(3634.1 - 104.83)\text{kJ/kg} = 14{,}602\text{ kW}$$

The fraction of energy at the turbine inlet that is converted to power is

$$f = \frac{\dot{W}_{\text{a}}}{\dot{Q}} = \frac{4500\text{ kW}}{14{,}602\text{ kW}} = 0.3082$$

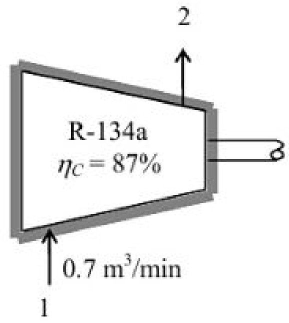
Assuming that the same fraction of heat loss from the turbine could have been converted to work, the possible increase in the power if the turbine is to be well-insulated becomes

$$\dot{W}_{\text{increase}} = f\dot{Q}_{\text{out}} = (0.3082)(220\text{ kW}) = \mathbf{67.8\text{ kW}}$$

7-131 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),



$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@100 \text{ kPa}} = 234.44 \text{ kJ/kg} \\ s_1 = s_{g@100 \text{ kPa}} = 0.95183 \text{ kJ/kg} \cdot \text{K} \\ v_1 = v_{g@100 \text{ kPa}} = 0.19254 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 282.51 \text{ kJ/kg}$$

From the isentropic efficiency relation,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_C = 234.44 + (282.51 - 234.44)/0.87 = 289.69 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 1 \text{ MPa} \\ h_{2a} = 289.69 \text{ kJ/kg} \end{array}\right\} T_{2a} = \mathbf{56.5^\circ C}$$

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

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.7/60 \text{ m}^3\text{/s}}{0.19254 \text{ m}^3\text{/kg}} = 0.06059 \text{ kg/s}$$

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

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

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

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

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

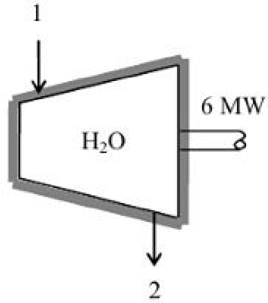
Substituting, the power input to the compressor becomes,

$$\dot{W}_{a,in} = (0.06059 \text{ kg/s})(289.69 - 234.44)\text{kJ/kg} = \mathbf{3.35 \text{ kW}}$$

7-125 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

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

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

(*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

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

7-128 Steam is expanded in an adiabatic turbine. The isentropic efficiency 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.

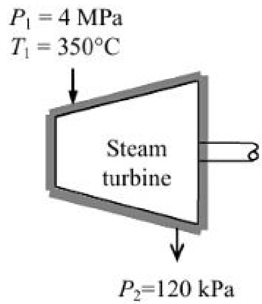
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

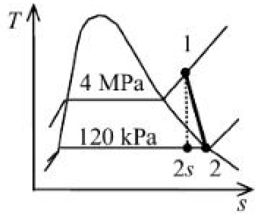


From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 4 \text{ MPa} \\ T_1 = 350°C \end{array}\right\} \begin{array}{l} h_1 = 3093.3 \text{ kJ/kg} \\ s_1 = 6.5843 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 120 \text{ kPa} \\ x_2 = 1 \end{array}\right\} h_2 = 2683.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{2s} = 120 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = 0.8798 \\ h_{2s} = 2413.4 \text{ kJ/kg} \end{array}$$



From the definition of the isentropic efficiency,

$$\eta_T = \frac{\dot{W}_{a,out}}{\dot{W}_{s,out}} = \frac{\dot{m}(h_1 - h_2)}{\dot{m}(h_1 - h_{2s})} = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{3093.3 - 2683.1}{3093.3 - 2413.4} = \mathbf{0.603} = \mathbf{60.3\%}$$