Turbines and Compressors

5-45C Yes.

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

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

5-48C No.

5-49 Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

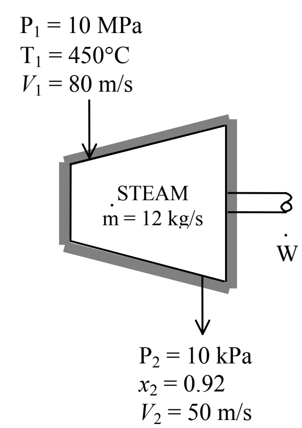
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

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

and

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



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

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

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

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

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

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

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

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

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

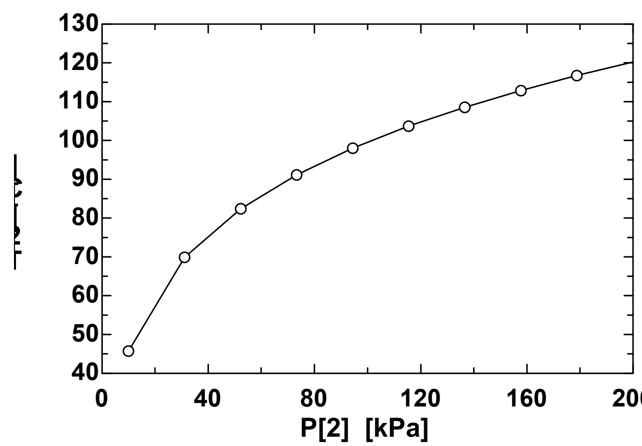
(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{\boldsymbol{v}_1}A_1V_1 \longrightarrow A_1 = \frac{\dot{m}\boldsymbol{v}_1}{V_1} = \frac{(12\text{ kg/s})(0.029782\text{ m}^3/\text{kg})}{80\text{ m/s}} = \mathbf{0.00447\ m^2}$$

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

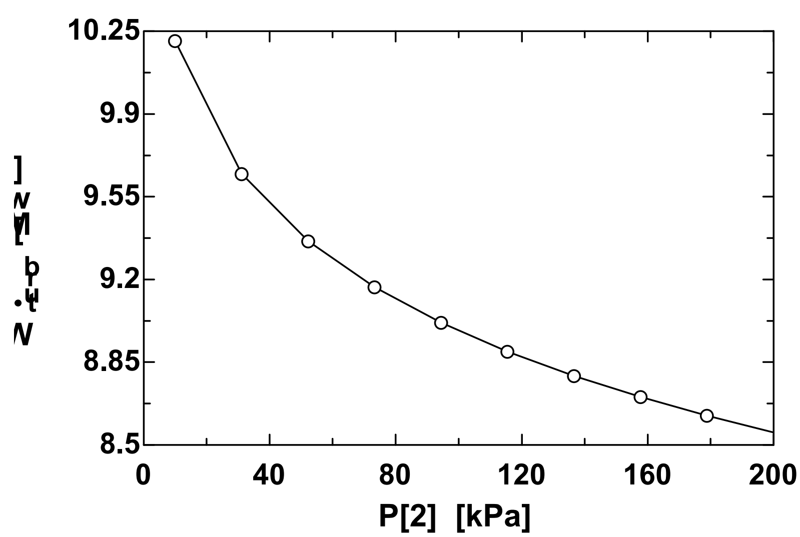
Analysis The problem is solved using EES, and the solution is given below.

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

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

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

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




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

DELTAke=Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)-Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)
```

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



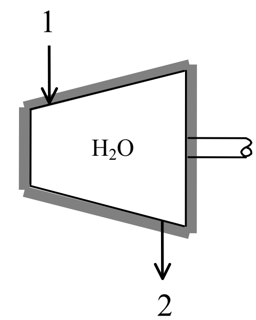
5-51 Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

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

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

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

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



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

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

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

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

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

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

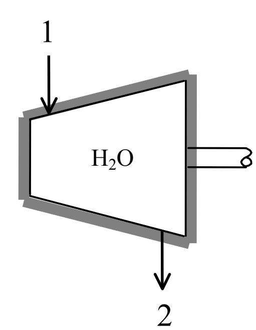
$$5000 \text{ kJ/s} = -\dot{m}(2344.7 - 3375.1) \text{ kJ/kg} \longrightarrow \dot{m} = \mathbf{4.852 \text{ kg/s}}$$

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

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

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

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



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

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

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

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

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

Substituting,

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

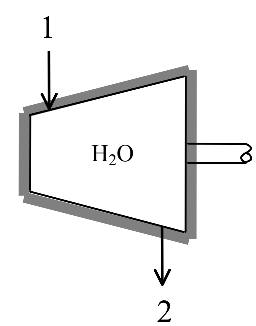
5-53 Steam expands in a turbine. The exit temperature of the steam for a power output of 2 MW is to be determined.

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

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

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

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



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

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

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

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

Substituting,

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

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

Then the exit temperature becomes

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

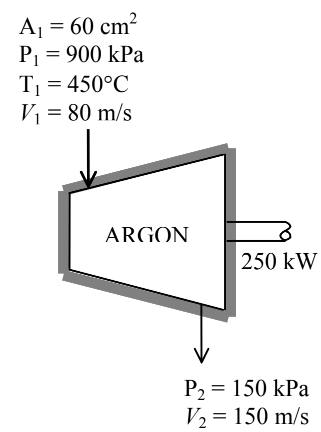
5-54 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is to be determined.

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

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

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

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



Thus,

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

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

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

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

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

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

Substituting,

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

It yields T_2 = **267.3°C**

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

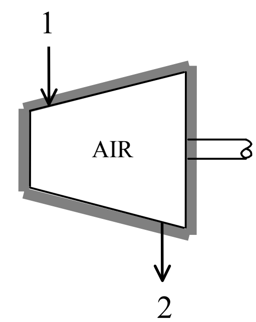
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

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

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1360 \text{ R})}{150 \text{ psia}} = 3.358 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{3.358 \text{ ft}^3/\text{lbm}}(0.1 \text{ ft}^2)(350 \text{ ft/s}) = \mathbf{10.42\ lbm/s}$$



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

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

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

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

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

Substituting,

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

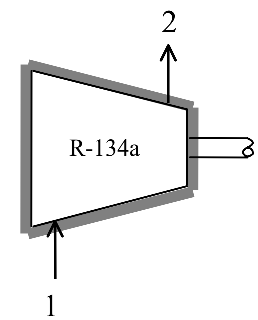
$$= 1486.5 \text{ Btu/s} = \mathbf{1568\ \ kW}$$

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

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

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

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



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

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

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

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

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

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

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

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

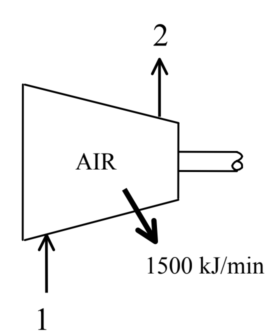
5-57 Air is compressed by a compressor. The mass flow rate of air through the compressor is to be determined.

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

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

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

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



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

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

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

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

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

Substituting, the mass flow rate is determined to be

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

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

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

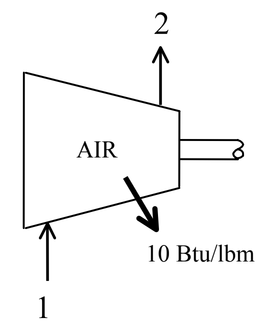
Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1E). The inlet enthalpy of air is (Table A-17E)

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

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

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

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



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

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

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

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

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

Substituting,

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

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

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

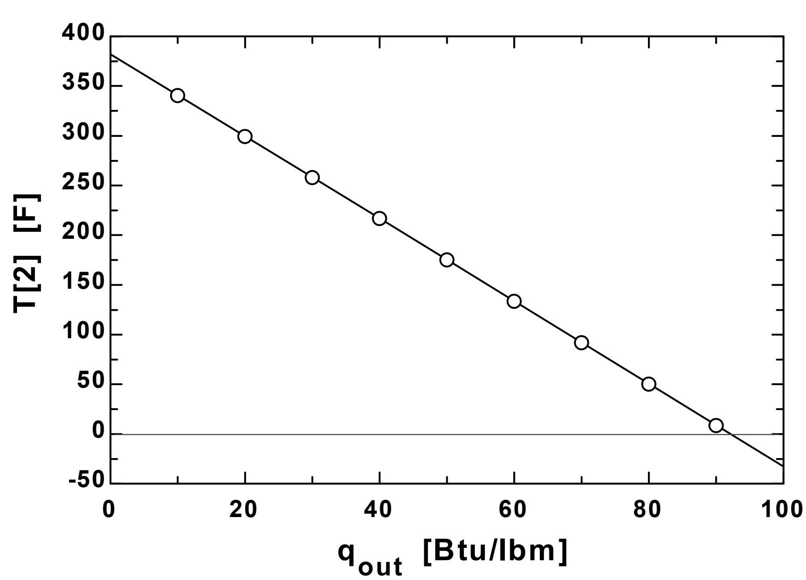
$$T_2 = 801\text{ R} = \mathbf{341°F}$$

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

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

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

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

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

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

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

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



5-60 Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required is to be determined.

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

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

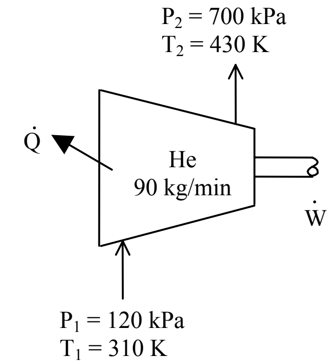
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Thus,

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

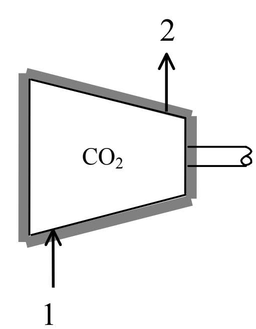
5-61 CO_2 is compressed by a compressor. The volume flow rate of CO_2 at the compressor inlet and the power input to the compressor are to be determined.

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

Properties The gas constant of CO_2 is R = 0.1889 kPa.m³/kg.K, and its molar mass is M = 44 kg/kmol (Table A-1). The inlet and exit enthalpies of CO_2 are (Table A-20)

$$T_1 = 300\text{ K} \rightarrow \bar{h}_1 = 9{,}431\text{ kJ/kmol}$$

$$T_2 = 450\text{ K} \rightarrow \bar{h}_2 = 15{,}483\text{ kJ/kmol}$$



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

The inlet specific volume of air and its volume flow rate are

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

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

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

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

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

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

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

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

Throttling Valves

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

5-63C Yes.

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

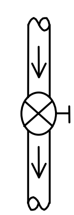
5-65C If it remains in the liquid phase, no. But if some of the liquid vaporizes during throttling, then yes.

5-66 Refrigerant-134a is throttled by a valve. The temperature drop of the refrigerant and specific volume after expansion are to be determined.

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

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

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



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

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

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

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

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

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

The quality at this state is determined from

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

Thus,

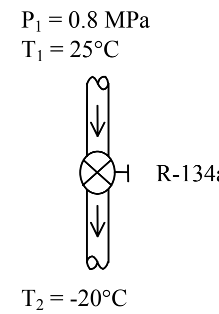
$$v_2 = v_f + x_2 v_{fg} = 0.0007437 + 0.2745 \times (0.12348 - 0.0007437) = \mathbf{0.0344\ m^3/kg}$$

5-67 [*Also solved by EES on enclosed CD*] Refrigerant-134a is throttled by a valve. The pressure and internal energy after expansion are to be determined.

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

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

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



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

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

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

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

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

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

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

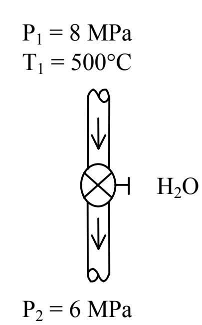
Thus, $$u_2 = u_f + x_2 u_{fg} = 25.39 + 0.2861 \times 193.45 = \mathbf{80.74 \text{ kJ/kg}}$$

5-68 Steam is throttled by a well-insulated valve. The temperature drop of the steam after the expansion is to be determined.

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

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

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



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

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

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

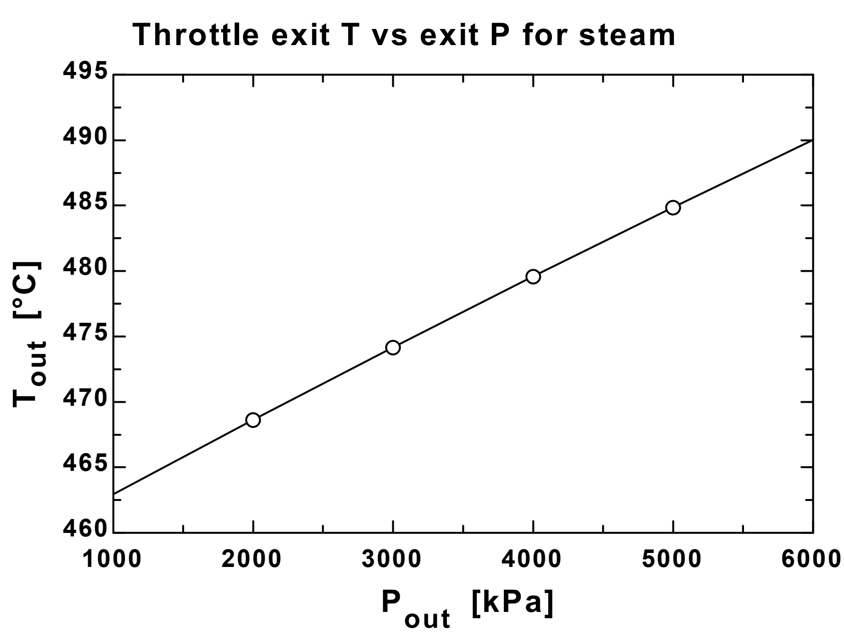
$$\left.\begin{array}{l} P_2 = 6 \text{ MPa} \\ (h_2 = h_1) \end{array}\right\} T_2 = \mathbf{490.1°C}$$

5-69 EES Problem 5-68 is reconsidered. The effect of the exit pressure of steam on the exit temperature after throttling as the exit pressure varies from 6 MPa to 1 MPa is to be investigated. The exit temperature of steam is to be plotted against the exit pressure.

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

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

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

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

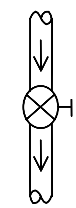
P_{out} [kPa]	T_{out} [C]
1000	463.1
2000	468.8
3000	474.3
4000	479.7
5000	484.9
6000	490.1



5-70E High-pressure air is throttled to atmospheric pressure. The temperature of air after the expansion is to be determined.

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

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



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

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

Therefore,

$$T_2 = T_1 = \mathbf{90°F}$$

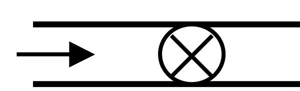
5-71 Carbon dioxide flows through a throttling valve. The temperature change of CO_2 is to be determined if CO_2 is assumed an ideal gas and a real gas.

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

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

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

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



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

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

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

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

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

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

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

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