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

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

**Properties** The enthalpy of air at the inlet temperature of 500 K is $h_1 = 503.02 \text{ kJ/kg}$ (Table A-17).

**Analysis**

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

$$\frac{\dot{E}_{in} - \dot{E}_{out}}{\dot{m}} = \Delta \dot{E}_{system} = 0$$

or,

$$\dot{m}(h_1 + \frac{V_1^2}{2}) = \dot{m}(h_2 + \frac{V_2^2}{2}) \quad \text{(since } Q = W = \Delta \text{则)l equals 0)}$$

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

Then from Table A-17 we read $T_2 = 436.5 \text{ K}$

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

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \Rightarrow \frac{1}{RT_2 / P_2} A_2 V_2 = \frac{1}{RT_1 / P_1} A_1 V_1$$

Thus,

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

R-134a is decelerated in a diffuser from a velocity of 140 m/s. The exit velocity of R-134a and the mass flow rate of the R-134a are to be determined.

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

**Properties** From the R-134a tables (Tables A-11 through A-13)

$$P_1 = 700 \text{ kPa} \quad v_1 = 0.0292 \text{ m}^3/\text{kg} \quad h_1 = 261.85 \text{ kJ/kg}$$

and

$$P_2 = 800 \text{ kPa} \quad v_2 = 0.0269 \text{ m}^3/\text{kg} \quad T_2 = 40^\circ \text{C} \quad h_2 = 273.66 \text{ kJ/kg}$$

**Analysis**

(a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity of R-134a is determined from the steady-flow mass balance to be
\[
\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \quad \rightarrow \quad V_2 = \frac{V_1 A_1}{A_2} \quad v_2 = \frac{v_1 A_1}{A_2} \quad 1 \quad 0.0269 \quad 0.0292 \times (140 \text{ m/s}) = 71.7 \text{ m/s}
\]

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

\[
\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system} = 0
\]

\(\dot{Q}_{in} + \dot{m}(h_1 + V_1^2 / 2) = \dot{m}(h_2 + V_2^2 / 2) \quad (\text{since } \dot{W} \approx \Delta p e \approx 0)
\]

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

\[
3 \text{ kJ/s} = \dot{m} \left[ 273.66 - 261.85 + \frac{(71.7 \text{ m/s})^2 - (140 \text{ m/s})^2}{2} \left( \frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) \right] \]

It yields \(\dot{m} = 0.655 \text{ kg/s}\)

**5-79** 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)

\[
P_1 = 10 \text{ MPa} \quad v_1 = 0.02975 \text{ m}^3/\text{kg} \\
T_1 = 450^\circ \text{C} \quad h_1 = 3240.9 \text{ kJ/kg}
\]

and

\[
P_2 = 10 \text{ kPa} \\
x_2 = 0.92
\]

\[
h_2 = h_f + x_2 h_{fg} = 191.83 + 0.92 \times 2392.8 = 2393.2 \text{ kJ/kg}
\]

**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

\[
\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system} (\text{steady}) = 0
\]

\[
\dot{E}_{in} = \dot{E}_{out}
\]
\begin{align*}
\dot{m}(h_1 + \frac{V_1^2}{2}) &= \dot{W}_{\text{out}} + \dot{m}(h_2 + \frac{V_2^2}{2}) \quad \text{(since } \dot{Q} \approx \Delta p \approx 0) \\
\dot{W}_{\text{out}} &= -\dot{m} \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \right)
\end{align*}

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

\[\dot{W}_{\text{out}} = -(12 \text{ kg/s})(2393.2 - 3240.9 - 1.95) \text{kJ/kg} = 10.2 \text{ MW}\]

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

\[m = \frac{1}{v_1} A_1 V_1 \Rightarrow A_1 = \frac{\dot{m} V_1}{V_1} = \frac{(12 \text{ kg/s})(0.02975 \text{ m}^3/\text{kg})}{80 \text{ m/s}} = 0.00446 \text{ m}^2\]

5-93C Yes. (Would have to be a non-ideal substance like R-134a or steam.)

5-108 Two streams of refrigerant-134a are mixed in a chamber. If the cold stream enters at twice the rate of the hot stream, the temperature and quality (if saturated) of the exit stream are to be determined.

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

**Properties**
From R-134a tables (Tables A-11 through A-13),

\[h_1 \approx h_f \text{ @ } 12^\circ \text{C} = 66.18 \text{ kJ/kg}\]
\[h_2 = h_{fg} \text{ @ } 1 \text{ MPa, } 60^\circ \text{C} = 291.36 \text{ kJ/kg}\]

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

Mass balance: \[\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}} \Rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \Rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2 \text{ since } \dot{m}_1 = 2\dot{m}_2\]

Energy balance:

\[\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \dot{E}_{\text{system}} \Rightarrow \Delta \dot{E}_{\text{system}} (\text{steady}) = 0\]

\[\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad \text{(since } \dot{Q} \approx \dot{W} \approx \Delta p \approx 0)\]

Combining the two gives \[2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3 \text{ or } h_3 = \frac{(2h_1 + h_2)}{3}\]

Substituting,

\[h_3 = (2 \times 66.18 + 291.36)/3 = 141.24 \text{ kJ/kg}\]

At 1 MPa, \(h_f = 105.29 \text{ kJ/kg} \text{ and } h_g = 267.97 \text{ kJ/kg}. \) Thus the exit stream is a saturated mixture since \(h_f < h_3 < h_g. \) Therefore,

\[T_3 = T_{\text{sat}} \text{ @ } 1 \text{ MPa} = 39.39^\circ \text{C}\]

and

\[x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{141.24 - 105.29}{162.68} = 0.221\]
Steam is condensed by cooling water in the condenser of a power plant. If the temperature rise of the cooling water is not to exceed 10°C, the minimum mass flow rate of the cooling water required is to be determined.

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

**Properties**
The cooling water exists as compressed liquid at both states, and its specific heat at room temperature is $C_W = 4.18 \, \text{kJ/kg} \cdot ^\circ \text{C}$ (Table A-3). The enthalpies of the steam at the inlet and the exit states are (Tables A-5 and A-6)

$$
\begin{align*}
P_3 &= 20 \, \text{kPa} \\
x_3 &= 0.95 \\
\left. h_3\right|_{sat\text{liquid}} &= 251.40 + 0.95 \times 2358.3 = 2491.8\, \text{kJ/kg} \\
P_4 &= 20 \, \text{kPa} \\
\left. h_4\right|_{20\, \text{kPa}} &= 251.40 \, \text{kJ/kg}
\end{align*}
$$

**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):

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

**Energy balance** (for the heat exchanger):

$$
\dot{E}_\text{in} - \dot{E}_\text{out} = \Delta \dot{E}_\text{system} \overset{(steady)}{=} 0
$$

Where

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

Combining the two,

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

Solving for $\dot{m}_w$:

$$
\dot{m}_w = \frac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \equiv \frac{h_1 - h_4}{C_p(T_2 - T_1)} \dot{m}_s
$$

Substituting,

$$
\dot{m}_w = \frac{(2491.8 - 251.4) \, \text{kJ/kg}}{(4.18 \, \text{kJ/kg} \cdot ^\circ \text{C})(10 \, ^\circ \text{C})} \approx 298 \, \text{kg/s} = 17.866 \, \text{kg/min}
$$
Two identical buildings in Los Angeles and Denver have the same infiltration rate. The ratio of the heat losses by infiltration at the two cities under identical conditions is to be determined.

**Assumptions** 1 Both buildings are identical and both are subjected to the same conditions except the atmospheric conditions. 2 Air is an ideal gas with constant specific heats at room temperature. 3 Steady flow conditions exist.

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

\[
\dot{E}_{in} - \dot{E}_{out} = \dot{\Delta E}_{system} \equiv 0
\]

Los Angeles: 101 kPa
Denver: 83 kPa

\[
\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad \text{(since } \Delta ke \equiv \Delta pe \equiv 0)\\
\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1) = \rho V C_p(T_2 - T_1)
\]

Then the sensible infiltration heat loss (heat gain for the infiltrating air) can be expressed

\[
\dot{Q}_{infiltration} = \dot{m}_{air} C_p(T_1 - T_0) = \dot{m}_{air} (ACH)(V_{building})C_p(T_1 - T_0)
\]

where \(ACH\) is the infiltration volume rate in air changes per hour.

Therefore, the infiltration heat loss is proportional to the density of air, and thus the ratio of infiltration heat losses at the two cities is simply the densities of outdoor air at those cities,

\[
\frac{\dot{Q}_{infiltration, Los Angeles}}{\dot{Q}_{infiltration, Denver}} = \frac{\rho_{o, air, Los Angeles}}{\rho_{o, air, Denver}}\\
= \frac{(P_0 / RT_0)_{Los Angeles}}{(P_0 / RT_0)_{Denver}}\\
= \frac{101 \text{ kPa}}{83 \text{ kPa}} = 1.22
\]

Therefore, the infiltration heat loss in Los Angeles will be 22% higher than that in Denver under identical conditions.

Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

**Assumptions** 1 Steady operating conditions exist. 2 The thermal properties of chickens and water are constant.

**Properties** The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

**Analysis** (a) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

\[
\dot{m}_{chicken} = (500 \text{ chicken} / \text{h})(2.2 \text{ kg / chicken}) = 1100 \text{ kg / h} = 0.3056 \text{ kg / s}
\]

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as

\[
\dot{E}_{in} - \dot{E}_{out} = \dot{\Delta E}_{system} \equiv 0
\]
\[ \dot{E}_{in} - \dot{E}_{out} = \frac{\Delta E_{system}}{\text{(steady)}} = 0 \]

Rate of net energy transfer by heat, work, and mass

\[ \dot{E}_{in} = \dot{E}_{out} \]

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

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

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3°C becomes

\[ \dot{Q}_{\text{chicken}} = (\dot{m}c_p \Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg} \cdot \text{°C})(15 - 3) \text{°C} = 13.0 \text{ kW} \]

The chiller gains heat from the surroundings at a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

\[ \dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 0.056 = 13.056 \text{ kW} \]

Noting that the temperature rise of water is not to exceed 2°C as it flows through the chiller, the mass flow rate of water must be at least

\[ \dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(C_p \Delta T)_{\text{water}}} = \frac{13.056 \text{ kW}}{(4.18 \text{ kJ/kg} \cdot \text{°C})(2 \text{°C})} = 1.56 \text{ kg/s} \]

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.
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)

\[
\begin{align*}
P_3 &= 12.5 \text{ MPa} \\
T_3 &= 500^\circ\text{C} \\
\end{align*}
\]

and

\[
\begin{align*}
P_4 &= 10 \text{ kPa} \\
x_4 &= 0.92 \\
\end{align*}
\]

From the air table (Table A-17),

\[
\begin{align*}
T_1 &= 295 \text{ K} \quad \Rightarrow \quad h_1 = 295.17 \text{ kJ/kg} \\
T_2 &= 620 \text{ K} \quad \Rightarrow \quad h_2 = 628.07 \text{ kJ/kg} \\
\end{align*}
\]

**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

\[
\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system} = 0
\]

For the turbine and the compressor it becomes

**Compressor:**

\[
\dot{W}_{comp, in} + \dot{m}_{air} h_1 - \dot{m}_{air} h_2 = \dot{W}_{comp, in} (h_2 - h_1)
\]

**Turbine:**

\[
\dot{m}_{steam} h_3 + \dot{m}_{steam} h_4 = \dot{W}_{turb, out} (h_3 - h_4)
\]

Substituting,

\[
\begin{align*}
\dot{W}_{comp, in} &= (10 \text{ kg/s})(628.07 - 295.17)\text{ kJ/kg} = 3329 \text{ kW} \\
\dot{W}_{turb, out} &= (25 \text{ kg/s})(3341.8 - 2393.2)\text{ kJ/kg} = 23,715 \text{ kW}
\end{align*}
\]

Therefore,

\[
\dot{W}_{net, out} = \dot{W}_{turb, out} - \dot{W}_{comp, in} = 23,715 - 3329 = 20,386 \text{ kW}
\]