Chapter 5

Comments:

5-3C The amount of mass or energy entering a control volume does not have to be equal to the amount of mass or energy leaving during an unsteady-flow process.

5-19C Flow energy or flow work is the energy needed to push a fluid into or out of a control volume. Fluids at rest do not possess any flow energy.

5-27C It is mostly converted to internal energy as shown by a rise in the fluid temperature.

5-45C Yes. For an adiabatic operation of a turbine, \( -\dot{W} = \dot{m}(h_2 - h_1) \)

Problems:

5-6E A garden hose is used to fill a water bucket. The volume and mass flow rates of water, the filling time, and the discharge velocity are to be determined.

Assumptions 1. Water is an incompressible substance. 2. Flow through the hose is steady. 3. There is no waste of water by splashing.

Properties We take the density of water to be 62.4 lbm/ft\(^3\) (Table A-3E).

Analysis (a) The volume and mass flow rates of water are

\[
\dot{V} = \dot{m} = \left(\frac{\pi D^2}{4}\right) \frac{\text{ft}^3}{\text{s}} = 0.04363 \text{ ft}^3/\text{s}
\]

(b) The time it takes to fill a 20-gallon bucket is

\[
\Delta t = \frac{\dot{V}}{\dot{m} \left(\frac{\text{ft}^3}{\text{gal}}\right)} = 61.3 \text{ s}
\]

(c) The average discharge velocity of water at the nozzle exit is

\[
V_o = \sqrt{\frac{2\dot{m}}{\rho D^2}} = \sqrt{\frac{2(62.4 \text{ lbm/ft}^3)(0.04363 \text{ ft}^3/\text{s})}{1 \text{ ft}^2}} = 32 \text{ ft/s}
\]

Discussion Note that for a given flow rate, the average velocity is inversely proportional to the square of the velocity. Therefore, when the diameter is reduced by half, the velocity quadruples.

5-8 Air is expanded and is accelerated as it is heated by a hair dryer of constant diameter. The percent increase in the velocity of air as it flows through the drier is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 1.20 kg/m\(^3\) at the inlet, and 1.05 kg/m\(^3\) at the exit.

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

\[
\rho_1 A V_1 = \rho_2 A V_2
\]

\[
\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20 \text{ kg/m}^3}{1.05 \text{ kg/m}^3} = 1.14 \quad \text{(or, and increase of 14%)}
\]

Therefore, the air velocity increases 14% as it flows through the hair dryer.
Air is accelerated in a nozzle from 30 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle 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 constant 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 gas constant of air is 0.287 kPa·m³/kg·K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is $c_p = 1.02$ kJ/kg·K (Table A-2).

**Analysis** (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(473 \text{ K})}{300 \text{ kPa}} = 0.4525 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_1} A_1' V_1' = \frac{1}{0.4525 \text{ m}^3/\text{kg}} (0.008 \text{ m}^3)(30 \text{ m/s}) = 0.5304 \text{ kg/s}$$

(b) 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

$$\dot{E}_{in} - \dot{E}_{out} = \Delta E_{sys}$$

Rate of energy transfer by loss, work, friction
Rate of change in internal energy, thermal, or sensible energy

$$\dot{E}_{in} = \dot{E}_{out} = \dot{m}(h_1 + V_1'^2/2)$$

Since $Q = W = \Delta p e = 0$

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

Substituting

$$0 = (1.02 \text{ kJ/kg} \cdot \text{K})(T_2 - 200 \text{ }^\circ\text{C}) + (180 \text{ m/s})^2 - (30 \text{ m/s})^2 \left( \frac{1 \text{ kJ/kg}}{1000 \text{ m}^3/\text{s}^2} \right)$$

It yields $T_2 = 184.6^\circ\text{C}$

(c) The specific volume of air at the nozzle exit is

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(184.6 + 273 \text{ K})}{100 \text{ kPa}} = 1.313 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_2} A_2 V_2 = 0.5304 \text{ kg/s} = \frac{1}{1.313 \text{ m}^3/\text{kg}} A_2 (180 \text{ m/s}) \rightarrow A_2 = 0.00387 \text{ m}^2 = 38.7 \text{ cm}^2$$
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 0E)

\[
\begin{align*}
& \frac{T_1}{1000 \text{ psia}} = 1448.6 \text{ Btu/lbm} \quad \frac{T_2}{5 \text{ psia}} = 1130.7 \text{ Btu/lbm} \\
& \frac{T_1}{900^\circ F} = \frac{T_2}{\text{sat vapor}} = 1130.7 \text{ Btu/lbm}
\end{align*}
\]

**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

\[
\dot{E}_{in} - \dot{E}_{out} = \frac{\Delta E_{system}}{\text{time}} \stackrel{\text{steady}}{=} 0 = 0
\]

\[
\dot{E}_{in} = \dot{E}_{out} = \dot{m} h_1 + \dot{W}_{in} = \dot{m} h_2 - \dot{W}_{out}
\]

Substituting

\[
\dot{W}_{out} = -(45000 \text{ Btu/s})(1130.7 - 1448.6) \text{Btu/lbm} = -4000 \text{ kJ/s}
\]

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 \( R = 0.1889 \text{ kPa.m}^3/\text{kg.K} \), and its molar mass is \( M = 44 \text{ kg/kmol} \) (Table A-1). The inlet and exit enthalpies of CO\(_2\) are (Table A-20)

\[
\begin{align*}
T_1 &= 300 \text{ K} \quad \rightarrow \quad h_1 = 9.431 \text{ kJ/kg} \\
T_2 &= 450 \text{ K} \quad \rightarrow \quad h_2 = 15.483 \text{ kJ/kg}
\end{align*}
\]

**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

\[
\nu_1 = \frac{R T_1}{P_1} = \frac{(0.1889 \text{ kPa.m}^3/\text{kg.K})(300 \text{ K})}{100 \text{ kPa}} = 0.5667 \text{ m}^3/\text{kg}
\]

\[
\dot{V} = \dot{m} \nu_1 = (0.5 \text{ kg/s})(0.5667 \text{ m}^3/\text{kg}) = 0.283 \text{ m}^3/\text{s}
\]

(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

\[
\dot{E}_{in} - \dot{E}_{out} = \frac{\Delta E_{system}}{\text{time}} \stackrel{\text{steady}}{=} 0
\]

\[
\dot{E}_{in} = \dot{E}_{out} = \dot{m} h_1 + \dot{W}_{in} = \dot{m} h_2 - \dot{W}_{out}
\]

Substituting

\[
\dot{W}_{in} = (0.5 \text{ kg/s})(15.483 - 9.431 \text{ kJ/kg}) = 68.8 \text{ kW}
\]
Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of the air and the rate of heat transfer from the 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. 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. Air is an ideal gas with constant specific heats at room temperature.

**Properties** The gas constant of air is 0.3704 psia ft$^3$/lbm R (Table A-1E). The constant pressure specific heat of air is $c_p = 0.240$ Btu/lbm·°F (Table A-2E). The enthalpies of the R-134a at the inlet and the exit states are (Tables A-11E through A-13E)

- $P_1 = 20$ psia  
  $x_1 = 0.3$  
  $h_1 - h_f + x_1h_{fg} = 11.445 + 0.3 \times 91.282 = 38.83$ Btu/lbm

- $P_2 = 20$ psia  
  sat. vapor  
  $h_2 - h_f = 102.73$ Btu/lbm

**Analysis** (a) The inlet specific volume and the mass flow rate of air are

\[
\frac{V_1}{V_2} = \frac{p_2}{p_1} = 14.7 \text{ psia}
\]

and

\[
\dot{m}_a = \frac{V_1 U}{V_2} = \frac{\frac{200 \text{ ft}^3/\text{min}}{13.86 \text{ ft}^3/\text{lbm}}} = 14.43 \text{ lbm/min}
\]

We take the entire 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):

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

\[
0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}}, \quad \dot{m}_1 = \dot{m}_2 = \dot{m}_a \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_R
\]

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

\[
\begin{align*}
\dot{E}_{\text{in}} - \dot{E}_{\text{out}} & = \Delta \dot{E}_{\text{system}} \quad \text{(steady)} \\
& = 0
\end{align*}
\]

\[
\dot{m}_1 h_1 + \dot{m}_2 h_2 - \dot{m}_3 h_3 - \dot{m}_4 h_4 \quad \text{(since} \quad Q - W - \Delta \dot{E}_e = \Delta \dot{E}_p \approx 0) \]

Combining the two,

\[
\dot{m}_R (h_3 - h_4) = \dot{m}_a (h_2 - h_1) = \dot{m}_a c_p(T_2 - T_1)
\]

Solving for $T_2$:

\[
T_2 = T_1 + \frac{\dot{m}_R (h_3 - h_4)}{\dot{m}_a c_p}
\]

Substituting,

\[
T_2 = 90^\circ F + \frac{(4 \text{ lbm/min})(38.83 - 102.73) \text{Btu/lbm}}{(14.43 \text{ Btu/min}) (0.24 \text{ Btu/lbm} \cdot ^\circ F)} = 16.2^\circ F
\]

(b) The rate of heat transfer from the air to the refrigerant is determined from the steady-flow energy balance applied to the air only. It yields

\[
-Q_{\text{air, net}} = \dot{m}_a (h_1 - h_4) - \dot{m}_a c_p (T_2 - T_1)
\]

\[
-Q_{\text{air, net}} = -[(14.43 \text{ lbm/min})(0.24 \text{ Btu/lbm} \cdot ^\circ F)(16.2 - 90)^\circ F] = 255.6 \text{ Btu/min}
\]
An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

**Assumptions**

1. This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant.
2. Air is an ideal gas with variable specific heats.
3. Kinetic and potential energies are negligible.
4. There are no work interactions involved.
5. The direction of heat transfer is to the air in the bottle (will be verified).

**Properties**

The gas constant of air is 0.287 kPa·m³/kg·K (Table A-1).

**Analysis**

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

**Mass balance:**

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

**Energy balance:**

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

Net energy transfer by heat, work, and mass
Change in internal, kinetic, potential, etc. energies

$$Q_{\text{in}} = m_i h_i = m_f h_f \quad \text{(since } W = E_{\text{out}} = E_{\text{initial}} = 0, \text{ke} = p a = 0)$$

**Combining the two balances:**

$$Q_{\text{in}} = m_i (u_f - h_i)$$

where

$$m_i = \frac{PV}{RT_1} = \frac{(100 \text{ kPa})(0.008 \text{ m}^3)}{(0.287 \text{ kPa·m}^3/\text{kg·K})(290 \text{ K})} = 0.0096 \text{ kg}$$

$$T_1 = T_2 = 290 \text{ K (Table A-17)}$$

$$h_f = 290.16 \text{ kJ/kg}$$

$$u_f = 269.1 \text{ kJ/kg}$$

Substituting,

$$Q_{\text{in}} = (0.0096 \text{ kg})(290.16 - 269.1) \text{ kJ/kg} - 0.6 \text{ kJ}$$

Therefore, we reverse the direction.
A thermal-energy reservoir is a body that can supply or absorb finite quantities of heat isothermally.

**Assumptions**
1. This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant.
2. Kinetic and potential energies are negligible.
3. There are no work interactions involved.
4. The direction of heat transfer is to the tank (will be verified).

**Properties**

\[
\begin{align*}
T_1 &= 200^\circ C & \nu_1 &= \nu_{f@200^\circ C} = 0.001157 \text{ m}^3/\text{kg} \\
& \text{sat. liquid} & u_1 &= u_{f@200^\circ C} = 850.46 \text{ kJ/kg} \\
T_2 &= 200^\circ C & \rho_2 &= \rho_{f@200^\circ C} = 852.26 \text{ kJ/kg} \\
& \text{sat. liquid} &
\end{align*}
\]

**Analysis**

We take the tank as the system, which is a control volume since mass crosses the boundary.

Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy \( h \) and internal energy \( u \), respectively, the mass and energy balances for this uniform-flow system can be expressed as

**Mass balance:**

\[
m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_\rho = m_1 - m_2
\]

**Energy balance:**

\[
E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}
\]

Net energy transfer:

\[
\text{by heat, work, and mass}
\]

Change in internal, kinetic, potential, etc. energies:

\[
Q_\text{in} = m_1 h_2 + m_2 u_2 - m_1 u_1 \quad (\text{since } W = 0, \text{pe} = 0)
\]

The initial and final masses in the tank are:

\[
m_1 = \frac{V_1}{\nu_1} = \frac{0.3 \text{ m}^3}{0.001157 \text{ m}^3/\text{kg}} = 259.4 \text{ kg}
\]

\[
m_2 = \frac{1}{2} m_1 = \frac{1}{2} (259.4 \text{ kg}) = 129.7 \text{ kg}
\]

Then from the mass balance,

\[
m_\rho = m_1 - m_2 = 259.4 - 129.7 = 129.7 \text{ kg}
\]

Now we determine the final internal energy,

\[
\nu_\rho = \frac{m_2}{m_1} = \frac{0.3 \text{ m}^3}{129.7 \text{ kg}} - 0.002313 \text{ m}^3/\text{kg}
\]

\[
x_\rho = \frac{\nu_2 - \nu_\rho}{\nu_f} = \frac{0.002313 - 0.001157}{0.1721 - 0.001157} = 0.009171
\]

\[
T_3 = 200^\circ C \\
x_2 = 0.009171
\]

\[
\frac{u_2 - u_f}{u_f + x_2 u_f} = 850.46 + (0.009171)(1743.7) = 866.46 \text{ kJ/kg}
\]

Then the heat transfer during this process is determined from the energy balance by substitution to be

\[
Q = (129.7 \text{ kg})(852.26 \text{ kJ/kg}) - (129.7 \text{ kg})(866.46 \text{ kJ/kg}) - (259.4 \text{ kg})(850.46 \text{ kJ/kg})
\]

\[
= 2308 \text{ kJ}
\]

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**Chapter 6**

**Comments:**

6-6C A thermal-energy reservoir is a body that can supply or absorb finite quantities of heat isothermally.
Some examples are the oceans, the lakes, and the atmosphere.

6-29C The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a cold medium whereas the purpose of a heat pump is to supply heat to a warm medium.

6-32C No. Because the heat pump consumes work to accomplish this task.

Problems:

6-18 The rates of heat supply and heat rejection of a power plant are given. The power output and the thermal efficiency of this power plant are to be determined.

Assumptions 1 The plant operates steadily. 2 Heat losses from the working fluid at the pipes and other components are taken into consideration.

Analysis (a) The total heat rejected by this power plant is

\[ \dot{Q}_L = 145 + 8 = 153 \text{ GJ/h} \]

Then the net power output of the plant becomes

\[ \dot{W}_{\text{net, out}} = \dot{Q}_H - \dot{Q}_L = 280 - 153 = 127 \text{ GJ/h} = 35.3 \text{ MW} \]

(b) The thermal efficiency of the plant is determined from its definition,

\[ \eta_H = \frac{\dot{W}_{\text{net, out}}}{\dot{Q}_H} = \frac{127 \text{ GJ/h}}{280 \text{ GJ/h}} = 0.454 = 45.4\% \]

6-51 The rate of heat loss, the rate of internal heat gain, and the COP of a heat pump are given. The power input to the heat pump is to be determined.

Assumptions The heat pump operates steadily.

Analysis The heating load of this heat pump system is the difference between the heat lost to the outdoors and the heat generated in the house from the people, lights, and appliances.

\[ \dot{Q}_H = 60,000 - 4,000 = 56,000 \text{ kJ/h} \]

Using the definition of COP, the power input to the heat pump is determined to be

\[ \dot{W}_{\text{act.in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{56,000 \text{ kJ/h}}{2.5} \left( \frac{1 \text{ kW}}{3600 \text{ kJ/h}} \right) = 0.22 \text{ kW} \]
6-24 The projected power needs of the United States is to be met by building inexpensive but inefficient coal plants or by building expensive but efficient IGCC plants. The price of coal that will enable the IGCC plants to recover their cost difference from fuel savings in 5 years is to be determined.

**Assumptions**
1. Power is generated continuously by either plant at full capacity.
2. The time value of money (interest, inflation, etc.) is not considered.

**Properties**
The heating value of the coal is given to be $28 \times 10^8$ kJ/ton.

**Analysis**
For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are:

\[
\text{Construction cost (coal)} = (150,000,000 \text{ kW})(1300 \text{ kW}) = 195 \times 10^9
\]

\[
\text{Construction cost (IGCC)} = (150,000,000 \text{ kW})(1500 \text{ kW}) = 225 \times 10^9
\]

\[
\text{Construction cost difference} = 225 \times 10^9 - 195 \times 10^9 = 30 \times 10^9
\]

The amount of electricity produced by either plant in 5 years is:

\[
W_e = \frac{E_d}{\eta} = \frac{150,000,000 \text{ kW}}{\eta} = 6.570 \times 10^{12} \text{ kWh}
\]

The amount of fuel needed to generate a specified amount of power can be determined from:

\[
\eta = \frac{W_e}{Q_a} \Rightarrow Q_a = \frac{W_e}{\eta}
\]

or

\[
\frac{m_{\text{fuel}}}{\eta(\text{Heating value})} = \frac{Q_a}{\eta(\text{Heating value})} = \frac{W_e}{\eta(\text{Heating value})}
\]

Then the amount of coal needed to generate this much electricity by each plant and their difference are:

\[
\begin{align*}
\dot{m}_{\text{coal, coal plant}} &= \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570 \times 10^{12} \text{ kWh}}{3000 \text{ kJ}} \left( \frac{1 \text{ kWh}}{1 \text{ kJ}} \right) = 2.484 \times 10^8 \text{ tons} \\
\dot{m}_{\text{coal, IGCC plant}} &= \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570 \times 10^{12} \text{ kWh}}{5600 \text{ kJ}} \left( \frac{1 \text{ kWh}}{1 \text{ kJ}} \right) = 1.877 \times 10^9 \text{ tons}
\end{align*}
\]

\[
\Delta \dot{m}_{\text{coal}} = \dot{m}_{\text{coal, coal plant}} - \dot{m}_{\text{coal, IGCC plant}} = 2.484 \times 10^8 - 1.877 \times 10^9 = 0.607 \times 10^8 \text{ tons}
\]

For $\Delta \dot{m}_{\text{coal}}$ to pay for the construction cost difference of $30 billion, the price of coal should be:

\[
\text{Unit cost of coal} = \frac{\text{Construction cost difference}}{\Delta \dot{m}_{\text{coal}}} = \frac{30 \times 10^9}{0.607 \times 10^8 \text{ tons}} = 49.4 \text{ /ton}
\]

Therefore, the IGCC plant becomes attractive when the price of coal is above $49.4 per ton.

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6-42E The COP and the refrigeration rate of an ice machine are given. The power consumption is to be determined.

**Assumptions**
The ice machine operates steadily.

**Analysis**
The cooling load of this ice machine is:

\[
\dot{Q}_L = \dot{m}_L (28 \text{ lbm/h})(169 \text{ Btu/lbm}) = 4732 \text{ Btu/h}
\]

Using the definition of the coefficient of performance, the power input to the ice machine system is determined to be:

\[
\frac{W_{\text{in,im}}}{\text{COP}} = \frac{\dot{Q}_L}{2.4} = \frac{4732 \text{ Btu/h}}{2.4} \left( \frac{1 \text{ hp}}{2545 \text{ Btu/h}} \right) = 0.775 \text{ hp}
\]