6.10 be steady with time. This means that application of Eqs. 6.9 and 6.10 to the operation of some device is independent of time.

Many of the applications of the steady-state model are such that there is only one flow stream entering and one leaving the control volume. For this type of process, we can write

Continuity equation: $$\dot{m}_i = \dot{m}_e = \dot{m} \tag{6.11}$$

First law: $$\dot{Q}_{\text{C.V.}} + \dot{m}\left(h_i + \frac{\mathbf{V}_i^2}{2} + gZ_i\right) = \dot{m}\left(h_e + \frac{\mathbf{V}_e^2}{2} + gZ_e\right) + \dot{W}_{\text{C.V.}} \tag{6.12}$$

Rearranging this equation, we have

$$q + h_i + \frac{\mathbf{V}_i^2}{2} + gZ_i = h_e + \frac{\mathbf{V}_e^2}{2} + gZ_e + w \tag{6.13}$$

where, by definition,

$$q = \frac{\dot{Q}_{\text{C.V.}}}{\dot{m}} \quad \text{and} \quad w = \frac{\dot{W}_{\text{C.V.}}}{\dot{m}} \tag{6.14}$$

Note that the units for q and w are kJ/kg. From their definition, q and w can be thought of as the heat transfer and work (other than flow work) per unit mass flowing into and out of the control volume for this particular steady-state process.

The symbols q and w are also used for the heat transfer and work per unit mass of a control mass. However, since it is always evident from the context whether it is a control mass (fixed mass) or control volume (involving a flow of mass) with which we are concerned, the significance of the symbols q and w will also be readily evident in each situation.

The steady-state process is often used in the analysis of reciprocating machines, such as reciprocating compressors or engines. In this case the rate of flow, which may actually be pulsating, is considered to be the average rate of flow for an integral number of cycles. A similar assumption is made regarding the properties of the fluid flowing across the control surface and the heat transfer and work crossing the control surface. It is also assumed that for an integral number of cycles the reciprocating device undergoes, the energy and mass within the control volume do not change.

A number of examples are given in the next section to illustrate the analysis of steady-state processes.

In-Text Concept Questions

b. Can a steady-state device have boundary work?

c. What can you say about changes in $\dot{m}$ and $\dot{V}$ through a steady flow device?

d. In a multiple-device flow system, I want to determine a state property. Where should I look for information—upstream or downstream?

6.4 EXAMPLES OF STEADY-STATE PROCESSES

In this section, we consider a number of examples of steady-state processes in which there is one fluid stream entering and one leaving the control volume, such that the first law can

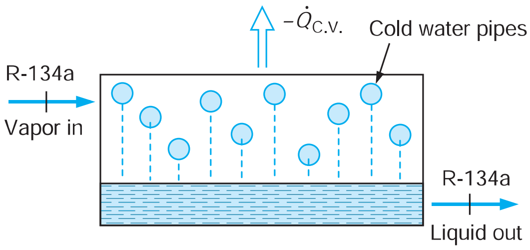

FIGURE 6.5 A refrigeration system condenser.

be written in the form of Eq. 6.13. Some may instead utilize control volumes that include more than one fluid stream, such that it is necessary to write the first law in the more general form of Eq. 6.10.

Heat Exchanger

A steady-state heat exchanger is a simple fluid flow through a pipe or system of pipes, where heat is transferred to or from the fluid. The fluid may be heated or cooled, and may or may not boil, changing from liquid to vapor, or condense, changing from vapor to liquid. One such example is the condenser in an R-134a refrigeration system, as shown in Fig. 6.5. Superheated vapor enters the condenser and liquid exits. The process tends to occur at constant pressure, since a fluid flowing in a pipe usually undergoes only a small pressure drop because of fluid friction at the walls. The pressure drop may or may not be taken into account in a particular analysis. There is no means for doing any work (shaft work, electrical work, etc.), and changes in kinetic and potential energies are commonly negligibly small. (One exception may be a boiler tube in which liquid enters and vapor exits at a much larger specific volume. In such a case, it may be necessary to check the exit velocity using Eq. 6.3.) The heat transfer in most heat exchangers is then found from Eq. 6.13 as the change in enthalpy of the fluid. In the condenser shown in Fig. 6.5, the heat transfer out of the condenser then goes to whatever is receiving it, perhaps a stream of air or of cooling water. It is often simpler to write the first law around the entire heat exchanger, including both flow streams, in which case there is little or no heat transfer with the surroundings. Such a situation is the subject of the following example.

EXAMPLE 6.3 Consider a water-cooled condenser in a large refrigeration system in which R-134a is the refrigerant fluid. The refrigerant enters the condenser at 1.0 MPa and 60°C, at the rate of 0.2 kg/s, and exits as a liquid at 0.95 MPa and 35°C. Cooling water enters the condenser at 10°C and exits at 20°C. Determine the rate at which cooling water flows through the condenser.

Control volume: Condenser.
Sketch: Fig. 6.6
Inlet states: R-134a—fixed; water—fixed.
Exit states: R-134a—fixed; water—fixed.
Process: Steady-state.
Model: R-134a tables; steam tables.

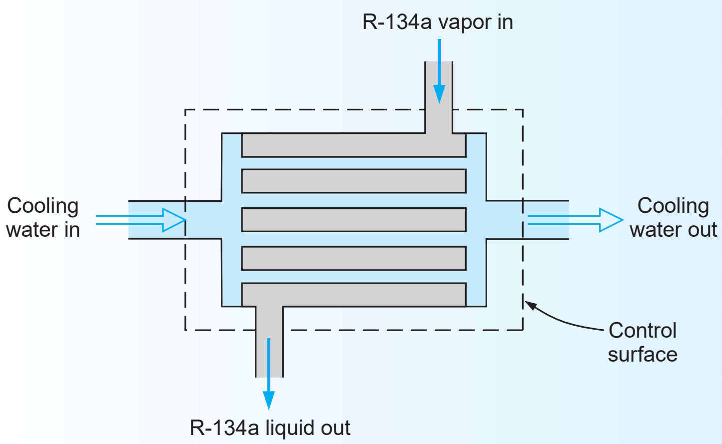

FIGURE 6.6
Schematic diagram of an R-134a condenser.

Analysis

With this control volume we have two fluid streams, the R-134a and the water, entering and leaving the control volume. It is reasonable to assume that both kinetic and potential energy changes are negligible. We note that the work is zero, and we make the other reasonable assumption that there is no heat transfer across the control surface. Therefore, the first law, Eq. 6.10, reduces to

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$

Using the subscript r for refrigerant and w for water, we write

$$\dot{m}_r (h_i)_r + \dot{m}_w (h_i)_w = \dot{m}_r (h_e)_r + \dot{m}_w (h_e)_w$$

Solution

From the R-134a and steam tables, we have

$$(h_i)_r = 441.89 \text{ kJ/kg}, \qquad (h_i)_w = 42.00 \text{ kJ/kg}$$
$$(h_e)_r = 249.10 \text{ kJ/kg}, \qquad (h_e)_w = 83.95 \text{ kJ/kg}$$

Solving the above equation for $\dot{m}_w$, the rate of flow of water, we obtain

$$\dot{m}_w = \dot{m}_r \frac{(h_i - h_e)_r}{(h_e - h_i)_w} = 0.2 \text{ kg/s} \frac{(441.89 - 249.10) \text{ kJ/kg}}{(83.95 - 42.00) \text{ kJ/kg}} = 0.919 \text{ kg/s}$$

This problem can also be solved by considering two separate control volumes, one having the flow of R-134a across its control surface and the other having the flow of water across its control surface. Further, there is heat transfer from one control volume to the other.

The heat transfer for the control volume involving R-134a is calculated first. In this case the steady-state energy equation, Eq. 6.10, reduces to

$$\begin{aligned} \dot{Q}_{\text{C.V.}} &= \dot{m}_r (h_e - h_i)_r \\ &= 0.2 \text{ kg/s} \times (249.10 - 441.89) \text{ kJ/kg} = -38.558 \text{ kW} \end{aligned}$$

This is also the heat transfer to the other control volume, for which $\dot{Q}_{\text{C.V.}} = +38.558\text{ kW}$.

$$\dot{Q}_{\text{C.V.}} = \dot{m}_w (h_e - h_i)_w$$

$$\dot{m}_w = \frac{38.558\text{ kW}}{(83.95 - 42.00)\text{ kJ/kg}} = 0.919\text{ kg/s}$$

Nozzle

A nozzle is a steady-state device whose purpose is to create a high-velocity fluid stream at the expense of the fluid's pressure. It is contoured in an appropriate manner to expand a flowing fluid smoothly to a lower pressure, thereby increasing its velocity. There is no means to do any work—there are no moving parts. There is little or no change in potential energy and usually little or no heat transfer. An exception is the large nozzle on a liquid-propellant rocket, such as was described in Section 1.7, in which the cold propellant is commonly circulated around the outside of the nozzle walls before going to the combustion chamber, in order to keep the nozzle from melting. This case, a nozzle with significant heat transfer, is the exception and would be noted in such an application. In addition, the kinetic energy of the fluid at the nozzle inlet is usually small and would be neglected if its value is not known.

EXAMPLE 6.4 Steam at 0.6 MPa and 200°C enters an insulated nozzle with a velocity of 50 m/s. It leaves at a pressure of 0.15 MPa and a velocity of 600 m/s. Determine the final temperature if the steam is superheated in the final state and the quality if it is saturated.

Control volume: Nozzle.
Inlet state: Fixed (see Fig. 6.7).
Exit state: P_e known.
Process: Steady-state.
Model: Steam tables.

Analysis

We have

$$\dot{Q}_{\text{C.V.}} = 0 \quad \text{(nozzle insulated)}$$
$$\dot{W}_{\text{C.V.}} = 0$$
$$\text{PE}_i \approx \text{PE}_e$$

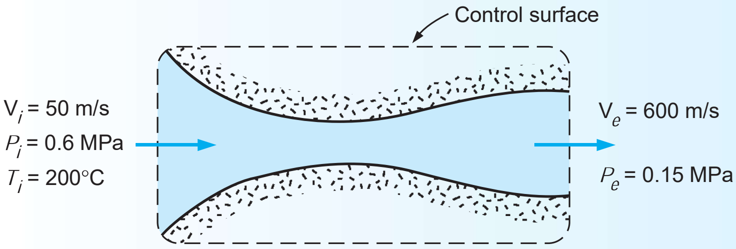

FIGURE 6.7 Illustration for Example 6.4.

The first law (Eq. 6.13) yields

$$h_i + \frac{\mathbf{V}_i^2}{2} = h_e + \frac{\mathbf{V}_e^2}{2}$$

Solution

Solving for h_e we obtain

$$h_e = 2850.1 + \left[\frac{(50)^2}{2 \times 1000} - \frac{(600)^2}{2 \times 1000}\right]\frac{\text{m}^2/\text{s}^2}{\text{J/kJ}} = 2671.4 \text{ kJ/kg}$$

The two properties of the fluid leaving that we now know are pressure and enthalpy, and therefore the state of this fluid is determined. Since h_e is less than h_g at 0.15 MPa, the quality is calculated.

$$h = h_f + xh_{fg}$$

$$2671.4 = 467.1 + x_e 2226.5$$

$$x_e = 0.99$$

EXAMPLE 6.4E

Steam at 100 lbf/in.^{2}, 400 F, enters an insulated nozzle with a velocity of 200 ft/s. It leaves at a pressure of 20.8 lbf/in.^{2} and a velocity of 2000 ft/s. Determine the final temperature if the steam is superheated in the final state and the quality if it is saturated.

Control volume: Nozzle.
Inlet state: Fixed (see Fig. 6.7E).
Exit state: P_e known.
Process: Steady-state.
Model: Steam tables.

Analysis

$$\dot{Q}_{\text{C.V.}} = 0 \quad \text{(nozzle insulated)}$$
$$\dot{W}_{\text{C.V.}} = 0, \qquad \text{PE}_i = \text{PE}_e$$

First law (Eq. 6.13):

$$h_i + \frac{\mathbf{V}_i^2}{2} = h_e + \frac{\mathbf{V}_e^2}{2}$$

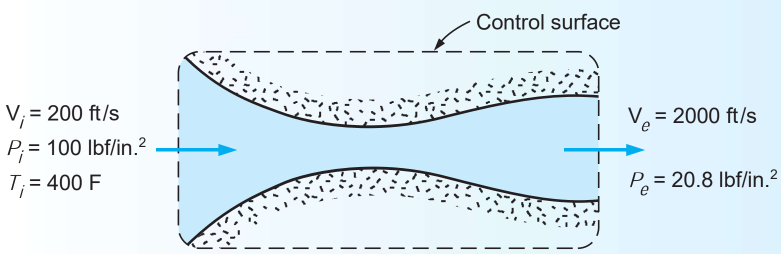

FIGURE 6.7E Illustration for Example 6.4E.

Solution

$$h_e = 1227.5 + \frac{(200)^2}{2 \times 32.17 \times 778} - \frac{(2000)^2}{2 \times 32.17 \times 778} = 1148.3 \text{ Btu/lbm}$$

The two properties of the fluid leaving that we now know are pressure and enthalpy, and therefore the state of this fluid is determined. Since h_e is less than h_g at 20.8 lbf/in.^{2}, the quality is calculated.

$$h = h_f + x h_{fg}$$

$$1148.3 = 198.31 + x_e 958.81$$

$$x_e = 0.99$$

Diffuser

A steady-state diffuser is a device constructed to decelerate a high-velocity fluid in a manner that results in an increase in pressure of the fluid. In essence, it is the exact opposite of a nozzle, and it may be thought of as a fluid flowing in the opposite direction through a nozzle, with the opposite effects. The assumptions are similar to those for a nozzle, with a large kinetic energy at the diffuser inlet and a small, but usually not negligible, kinetic energy at the exit being the only terms besides the enthalpies remaining in the first law, Eq. 6.13.

Throttle

A throttling process occurs when a fluid flowing in a line suddenly encounters a restriction in the flow passage. This may be a plate with a small hole in it, as shown in Fig. 6.8, it may be a partially closed valve protruding into the flow passage, or it may be a change to a tube of much smaller diameter, called a capillary tube, which is normally found on a refrigerator. The result of this restriction is an abrupt pressure drop in the fluid, as it is forced to find its way through a suddenly smaller passageway. This process is drastically different from the smoothly contoured nozzle expansion and area change, which results in a significant velocity increase. There is typically some increase in velocity in a throttle, but both inlet and exit kinetic energies are usually small enough to be neglected. There is no means for doing work and little or no change in potential energy. Usually, there is neither time nor opportunity for appreciable heat transfer, such that the only terms left in the first law, Eq. 6.13, are the inlet and exit enthalpies. We conclude that a steady-state throttling process is approximately a pressure drop at constant enthalpy, and we will assume this to be the case unless otherwise noted.

Frequently, a throttling process involves a change in the phase of the fluid. A typical example is the flow through the expansion valve of a vapor-compression refrigeration system. The following example deals with this problem.

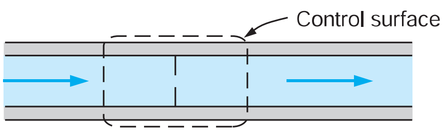

FIGURE 6.8 The throttling process.

EXAMPLE 6.5

Consider the throttling process across the expansion valve or through the capillary tube in a vapor-compression refrigeration cycle. In this process the pressure of the refrigerant drops from the high pressure in the condenser to the low pressure in the evaporator, and during this process some of the liquid flashes into vapor. If we consider this process to be adiabatic, the quality of the refrigerant entering the evaporator can be calculated.

Consider the following process, in which ammonia is the refrigerant. The ammonia enters the expansion valve at a pressure of 1.50 MPa and a temperature of 35°C. Its pressure on leaving the expansion valve is 291 kPa. Calculate the quality of the ammonia leaving the expansion valve.

Control volume: Expansion valve or capillary tube.
Inlet state: P_i, T_i known; state fixed.
Exit state: P_e known.
Process: Steady-state.
Model: Ammonia tables.

Analysis

We can use standard throttling process analysis and assumptions. The first law reduces to

$$h_i = h_e$$

Solution

From the ammonia tables we get

$$h_i = 346.8 \text{ kJ/kg}$$

(The enthalpy of a slightly compressed liquid is essentially equal to the enthalpy of saturated liquid at the same temperature.)

$$h_e = h_i = 346.8 = 134.4 + x_e(1296.4)$$
$$x_e = 0.1638 = 16.38\%$$

Turbine

A turbine is a rotary steady-state machine whose purpose is to produce shaft work (power, on a rate basis) at the expense of the pressure of the working fluid. Two general classes of turbines are steam (or other working fluid) turbines, in which the steam exiting the turbine passes to a condenser, where it is condensed to liquid, and gas turbines, in which the gas usually exhausts to the atmosphere from the turbine. In either type, the turbine exit pressure is fixed by the environment into which the working fluid exhausts, and the turbine inlet pressure has been reached by previously pumping or compressing the working fluid in another process. Inside the turbine, there are two distinct processes. In the first, the working fluid passes through a set of nozzles, or the equivalent—fixed blade passages contoured to expand the fluid to a lower pressure and to a high velocity. In the second process inside the turbine, this high-velocity fluid stream is directed onto a set of moving (rotating) blades, in which the velocity is reduced before being discharged from the passage. This directed velocity decrease produces a torque on the rotating shaft, resulting in shaft work output. The low-velocity, low-pressure fluid then exhausts from the turbine.

The first law for this process is either Eq. 6.10 or 6.13. Usually, changes in potential energy are negligible, as is the inlet kinetic energy. Often, the exit kinetic energy is neglected, and any heat rejection from the turbine is undesirable and is commonly small. We therefore normally assume that a turbine process is adiabatic, and the work output in this case reduces to the decrease in enthalpy from the inlet to exit states. In the following example, however, we include all the terms in the first law and study their relative importance.

EXAMPLE 6.6

The mass rate of flow into a steam turbine is 1.5 kg/s, and the heat transfer from the turbine is 8.5 kW. The following data are known for the steam entering and leaving the turbine.

	Inlet Conditions	**Exit Conditions**
Pressure	2.0 MPa	0.1 MPa
Temperature	350°C	
Quality		100%
Velocity	50 m/s	100 m/s
Elevation above reference plane	6 m	3 m
$g = 9.8066$ m/s^2		

Determine the power output of the turbine.

Control volume: Turbine (Fig. 6.9).

Inlet state: Fixed (above).

Exit state: Fixed (above).

Process: Steady-state.

Model: Steam tables.

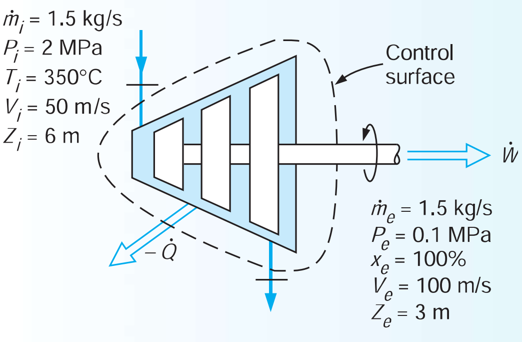

FIGURE 6.9 Illustration for Example 6.6.

Analysis

From the first law (Eq. 6.12) we have

$$\dot{Q}_{\text{C.V.}} + \dot{m}\left(h_i + \frac{\mathbf{V}_i^2}{2} + gZ_i\right) = \dot{m}\left(h_e + \frac{\mathbf{V}_e^2}{2} + gZ_e\right) + \dot{W}_{\text{C.V.}}$$

with

$$\dot{Q}_{\text{C.V.}} = -8.5\ \text{kW}$$

Solution

From the steam tables, $h_i = 3137.0$ kJ/kg. Substituting inlet conditions gives

$$\frac{\mathbf{V}_i^2}{2} = \frac{50 \times 50}{2 \times 1000} = 1.25\ \text{kJ/kg}$$

$$gZ_i = \frac{6 \times 9.8066}{1000} = 0.059\ \text{kJ/kg}$$

Similarly, for the exit $h_e = 2675.5$ kJ/kg and

$$\frac{\mathbf{V}_e^2}{2} = \frac{100 \times 100}{2 \times 1000} = 5.0\ \text{kJ/kg}$$

$$gZ_e = \frac{3 \times 9.8066}{1000} = 0.029\ \text{kJ/kg}$$

Therefore, substituting into Eq. 6.12, we obtain

$$-8.5 + 1.5(3137 + 1.25 + 0.059) = 1.5(2675.5 + 5.0 + 0.029) + \dot{W}_{\text{C.V.}}$$

$$\dot{W}_{\text{C.V.}} = -8.5 + 4707.5 - 4020.8 = 678.2\ \text{kW}$$

If Eq. 6.13 is used, the work per kilogram of fluid flowing is found first.

$$q + h_i + \frac{\mathbf{V}_i^2}{2} + gZ_i = h_e + \frac{\mathbf{V}_e^2}{2} + gZ_e + w$$

$$q = \frac{-8.5}{1.5} = -5.667\ \text{kJ/kg}$$

Therefore, substituting into Eq. 6.13, we get

$$-5.667 + 3137 + 1.25 + 0.059 = 2675.5 + 5.0 + 0.029 + w$$

$$w = 452.11\ \text{kJ/kg}$$

$$\dot{W}_{\text{C.V.}} = 1.5\ \text{kg/s} \times 452.11\ \text{kJ/kg} = 678.2\ \text{kW}$$

Two further observations can be made by referring to this example. First, in many engineering problems, potential energy changes are insignificant when compared with the other energy quantities. In the above example the potential energy change did not affect any of the significant figures. In most problems where the change in elevation is small, the potential energy terms may be neglected.

Second, if velocities are small—say, under 20 m/s—in many cases the kinetic energy is insignificant compared with other energy quantities. Furthermore, when the velocities entering and leaving the system are essentially the same, the change in kinetic energy is small. Since it is the change in kinetic energy that is important in the steady-state energy equation, the kinetic energy terms can usually be neglected when there is no significant difference between the velocity of the fluid entering and that leaving the control volume. Thus, in many thermodynamic problems, one must make judgments as to which quantities may be negligible for a given analysis.

The preceding discussion and example concerned the turbine, which is a rotary work-producing device. There are other nonrotary devices that produce work, which can be called *expanders* as a general name. In such devices, the first-law analysis and assumptions are generally the same as for turbines, except that in a piston/cylinder-type expander, there would in most cases be a larger heat loss or rejection during the process.

Compressor and Pump

The purpose of a steady-state compressor (gas) or pump (liquid) is the same: to increase the pressure of a fluid by putting in shaft work (power, on a rate basis). There are two fundamentally different classes of compressors. The most common is a rotary-type compressor (either axial flow or radial/centrifugal flow), in which the internal processes are essentially the opposite of the two processes occurring inside a turbine. The working fluid enters the compressor at low pressure, moving into a set of rotating blades, from which it exits at high velocity, a result of the shaft work input to the fluid. The fluid then passes through a diffuser section, in which it is decelerated in a manner that results in a pressure increase. The fluid then exits the compressor at high pressure.

The first law for the compressor is either Eq. 6.10 or 6.13. Usually, changes in potential energy are negligible, as is the inlet kinetic energy. Often the exit kinetic energy is neglected as well. Heat rejection from the working fluid during compression would be desirable, but it is usually small in a rotary compressor, which is a high-volume flow-rate machine, and there is not sufficient time to transfer much heat from the working fluid. We therefore normally assume that a rotary compressor process is adiabatic, and the work input in this case reduces to the change in enthalpy from the inlet to exit states.

In a piston/cylinder-type compressor, the cylinder usually contains fins to promote heat rejection during compression (or the cylinder may be water-jacketed in a large compressor for even greater cooling rates). In this type of compressor, the heat transfer from the working fluid is significant and is not neglected in the first law. As a general rule, in any example or problem in this book, we will assume that a compressor is adiabatic unless otherwise noted.

EXAMPLE 6.7

The compressor in a plant (see Fig. 6.10) receives carbon dioxide at 100 kPa, 280 K, with a low velocity. At the compressor discharge, the carbon dioxide exits at 1100 kPa, 500 K, with velocity of 25 m/s and then flows into a constant-pressure aftercooler (heat exchanger) where it is cooled down to 350 K. The power input to the compressor is 50 kW. Determine the heat transfer rate in the aftercooler.

Solution

C.V. compressor, steady state, single inlet and exit flow.

Energy Eq. 6.13: $$q + h_1 + \frac{1}{2}\mathbf{V}_1^2 = h_2 + \frac{1}{2}\mathbf{V}_2^2 + w$$

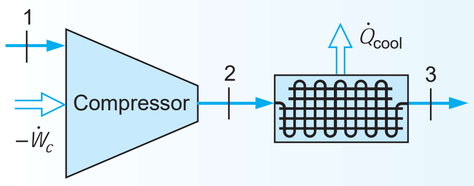

FIGURE 6.10 Sketch for Example 6.7.

In this solution, let us assume that the carbon dioxide behaves as an ideal gas with variable specific heat (Appendix A.8). It would be more accurate to use Table B.3 to find the enthalpies, but the difference is fairly small in this case.

We also assume that $q \cong 0$ and $\mathbf{V}_1 \cong 0$, so, getting h from Table A. 8,

$$-w = h_2 - h_1 + \frac{1}{2}\mathbf{V}_2^2 = 401.52 - 198 + \frac{(25)^2}{2 \times 1000} = 203.5 + 0.3 = 203.8 \text{ kJ/kg}$$

Remember here to convert kinetic energy J/kg to kJ/kg by division by 1000.

$$\dot{m} = \frac{\dot{W}_c}{w} = \frac{-50}{-203.8} = 0.245 \text{ kg/s}$$

C.V. aftercooler, steady state, single inlet and exit flow, and no work.

Energy Eq. 6.13: $$q + h_2 + \frac{1}{2}\mathbf{V}_2^2 = h_3 + \frac{1}{2}\mathbf{V}_3^2$$

Here we assume no significant change in kinetic energy (notice how unimportant it was) and again we look for h in Table A.8:

$$q = h_3 - h_2 = 257.9 - 401.5 = -143.6 \text{ kJ/kg}$$
$$\dot{Q}_{cool} = -\dot{Q}_{C.V.} = -\dot{m}q = 0.245 \text{ kg/s} \times 143.6 \text{ kJ/kg} = 35.2 \text{ kW}$$

EXAMPLE 6.8

A small liquid water pump is located 15 m down in a well (see Fig. 6.11), taking water in at 10°C, 90 kPa at a rate of 1.5 kg/s. The exit line is a pipe of diameter 0.04 m that goes up to a receiver tank maintaining a gauge pressure of 400 kPa. Assume the process is adiabatic with the same inlet and exit velocities and the water stays at 10°C. Find the required pump work.

C.V. pump + pipe. Steady state, one inlet, one exit flow. Assume same velocity in and out and no heat transfer.

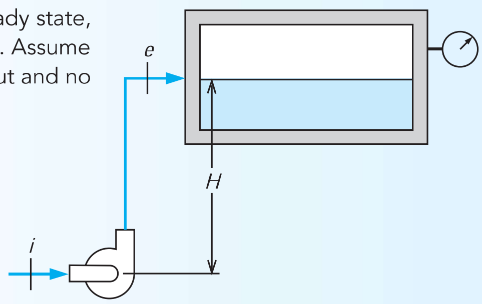

FIGURE 6.11 Sketch for Example 6.8.

Solution

Continuity equation: $\dot{m}_{in} = \dot{m}_{ex} = \dot{m}$

Energy Eq. 6.12: $\dot{m}\left(h_{in} + \frac{1}{2}\mathbf{V}_{in}^2 + gZ_{in}\right) = \dot{m}\left(h_{ex} + \frac{1}{2}\mathbf{V}_{ex}^2 + gZ_{ex}\right) + \dot{W}$

States: $h_{ex} = h_{in} + (P_{ex} - P_{in})v$ (v is constant and u is constant.)

From the energy equation

$$\dot{W} = \dot{m}(h_{in} + gZ_{in} - h_{ex} - gZ_{ex}) = \dot{m}[g(Z_{in} - Z_{ex}) - (P_{ex} - P_{in})v]$$

$$= 1.5\,\frac{\text{kg}}{\text{s}} \times \left[9.807\,\frac{\text{m}}{\text{s}^2} \times \frac{-15 - 0}{1000}\,\text{m} - (400 + 101.3 - 90)\text{ kPa} \times 0.001\,001\,\frac{\text{m}^3}{\text{kg}}\right]$$

$$= 1.5 \times (-0.147 - 0.412) = -0.84\text{ kW}$$

That is, the pump requires a power input of 840 W.

Power Plant and Refrigerator

The following examples illustrate the incorporation of several of the devices and machines already discussed in this section into a complete thermodynamic system, which is built for a specific purpose.

EXAMPLE 6.9

Consider the simple steam power plant, as shown in Fig. 6.12. The following data are for such a power plant.

Location	Pressure	Temperature or Quality
Leaving boiler	2.0 MPa	300°C
Entering turbine	1.9 MPa	290°C
Leaving turbine, entering condenser	15 kPa	90%
Leaving condenser, entering pump	14 kPa	45°C
Pump work = 4 kJ/kg		

Determine the following quantities per kilogram flowing through the unit:

a. Heat transfer in the line between the boiler and turbine.
b. Turbine work.
c. Heat transfer in the condenser.
d. Heat transfer in the boiler.

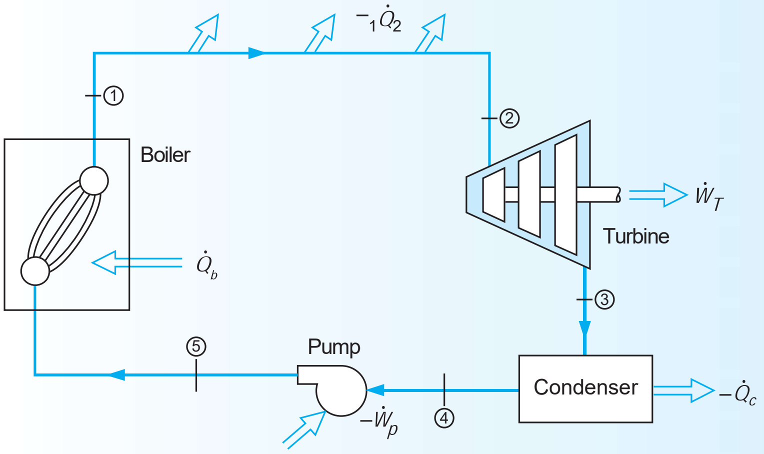

FIGURE 6.12 Simple steam power plant.

There is a certain advantage in assigning a number to various points in the cycle. For this reason, the subscripts i and e in the steady-state energy equation are often replaced by appropriate numbers.

Since there are several control volumes to be considered in the solution to this problem, let us consolidate our solution procedure somewhat in this example. Using the notation of Fig. 6.12, we have:

All processes: Steady-state.

Model: Steam tables.

From the steam tables:

$$h_1 = 3023.5 \text{ kJ/kg}$$
$$h_2 = 3002.5 \text{ kJ/kg}$$
$$h_3 = 226.0 + 0.9(2373.1) = 2361.8 \text{ kJ/kg}$$
$$h_4 = 188.5 \text{ kJ/kg}$$

All analyses: No changes in kinetic or potential energy will be considered in the solution. In each case, the first law is given by Eq. 6.13.

Now, we proceed to answer the specific questions raised in the problem statement.

a. For the control volume for the pipeline between the boiler and the turbine, the first law and solution are

$$_1q_2 + h_1 = h_2$$
$$_1q_2 = h_2 - h_1 = 3002.5 - 3023.5 = -21.0 \text{ kJ/kg}$$

b. A turbine is essentially an adiabatic machine. Therefore, it is reasonable to neglect heat transfer in the first law, so that

$$h_2 = h_3 + {}_2w_3$$
$${}_2w_3 = 3002.5 - 2361.8 = 640.7 \text{ kJ/kg}$$

c. There is no work for the control volume enclosing the condenser. Therefore, the first law and solution are

$${}_3q_4 + h_3 = h_4$$
$${}_3q_4 = 188.5 - 2361.8 = -2173.3 \text{ kJ/kg}$$

d. If we consider a control volume enclosing the boiler, the work is equal to zero, so that the first law becomes

$${}_5q_1 + h_5 = h_1$$

A solution requires a value for h_5, which can be found by taking a control volume around the pump:

$$h_4 = h_5 + {}_4w_5$$
$$h_5 = 188.5 - (-4) = 192.5 \text{ kJ/kg}$$

Therefore, for the boiler,

$${}_5q_1 + h_5 = h_1$$
$${}_5q_1 = 3023.5 - 192.5 = 2831 \text{ kJ/kg}$$

EXAMPLE 6.10

The refrigerator shown in Fig. 6.13 uses R-134a as the working fluid. The mass flow rate through each component is 0.1 kg/s, and the power input to the compressor is 5.0 kW. The following state data are known, using the state notation of Fig. 6.13:

$$P_1 = 100 \text{ kPa}, \qquad T_1 = -20^\circ\text{C}$$
$$P_2 = 800 \text{ kPa}, \qquad T_2 = 50^\circ\text{C}$$
$$T_3 = 30^\circ\text{C}, \qquad x_3 = 0.0$$
$$T_4 = -25^\circ\text{C}$$

Determine the following:

a. The quality at the evaporator inlet.
b. The rate of heat transfer to the evaporator.
c. The rate of heat transfer from the compressor.

All processes: Steady-state.
Model: R-134a tables.
All analyses: No changes in kinetic or potential energy. The first law in each case is given by Eq. 6.10.

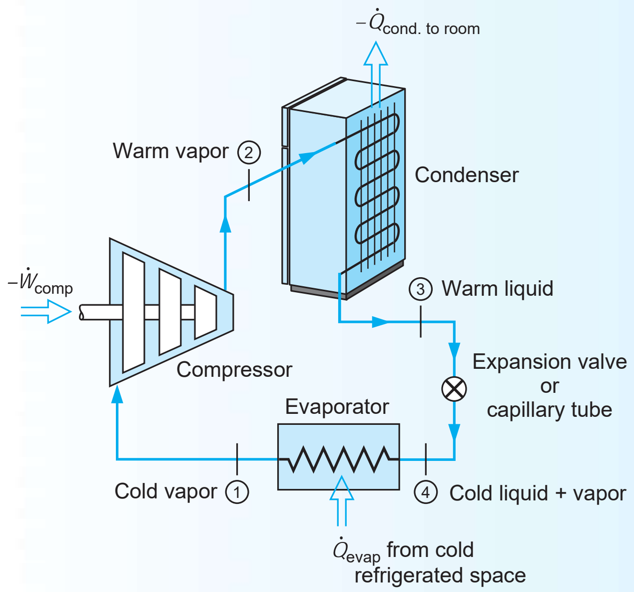

FIGURE 6.13
Refrigerator.

Solution

a. For a control volume enclosing the throttle, the first law gives

$$h_4 = h_3 = 241.8 \text{ kJ/kg}$$
$$h_4 = 241.8 = h_{f4} + x_4 h_{fg4} = 167.4 + x_4 \times 215.6$$
$$x_4 = 0.345$$

b. For a control volume enclosing the evaporator, the first law gives

$$\dot{Q}_{\text{evap}} = \dot{m}(h_1 - h_4)$$
$$= 0.1(387.2 - 241.8) = 14.54 \text{ kW}$$

c. And for the compressor, the first law gives

$$\dot{Q}_{\text{comp}} = \dot{m}(h_2 - h_1) + \dot{W}_{\text{comp}}$$
$$= 0.1(435.1 - 387.2) - 5.0 = -0.21 \text{ kW}$$

In-Text Concept Questions

e. How does a nozzle or sprayhead generate kinetic energy?

f. What is the difference between a nozzle flow and a throttle process?

g. If you throttle a saturated liquid, what happens to the fluid state? What happens if this is done to an ideal gas?

h. A turbine at the bottom of a dam has a flow of liquid water through it. How does that produce power? Which terms in the energy equation are important if the *CV* is the