Chapter 10

VAPOR AND COMBINED POWER CYCLES

I n Chap. 9 we discussed gas power cycles for which the working fluid remains a gas throughout the entire cycle. In this chapter, we consider *vapor power cycles* in which the working fluid is alternatively vaporized and condensed. We also consider power generation coupled with process heating called *cogeneration.*

The continued quest for higher thermal efficiencies has resulted in some innovative modifications to the basic vapor power cycle. Among these, we discuss the *reheat* and *regenerative cycles,* as well as combined gas–vapor power cycles.

Steam is the most common working fluid used in vapor power cycles because of its many desirable characteristics, such as low cost, availability, and high enthalpy of vaporization. Therefore, this chapter is mostly devoted to the discussion of steam power plants. Steam power plants are commonly referred to as *coal plants, nuclear plants,* or *natural gas plants,* depending on the type of fuel used to supply heat to the steam. However, the steam goes through the same basic cycle in all of them. Therefore, all can be analyzed in the same manner.

Objectives

The objectives of Chapter 10 are to:

- Analyze vapor power cycles in which the working fluid is alternately vaporized and condensed.
- Analyze power generation coupled with process heating called *cogeneration.*
- Investigate ways to modify the basic Rankine vapor power cycle to increase the cycle thermal efficiency.
- Analyze the reheat and regenerative vapor power cycles.
- Analyze power cycles that consist of two separate cycles known as combined cycles and binary cycles.

10–1 ▪ THE CARNOT VAPOR CYCLE

We have mentioned repeatedly that the Carnot cycle is the most efficient cycle operating between two specified temperature limits. Thus it is natural to look at the Carnot cycle first as a prospective ideal cycle for vapor power plants. If we could, we would certainly adopt it as the ideal cycle. As explained below, however, the Carnot cycle is not a suitable model for power cycles. Throughout the discussions, we assume *steam* to be the working fluid since it is the working fluid predominantly used in vapor power cycles.

Consider a steady-flow *Carnot cycle* executed within the saturation dome of a pure substance, as shown in Fig. 10-1*a*. The fluid is heated reversibly and isothermally in a boiler (process 1-2), expanded isentropically in a turbine (process 2-3), condensed reversibly and isothermally in a condenser (process 3-4), and compressed isentropically by a compressor to the initial state (process 4-1).

Several impracticalities are associated with this cycle:

1. Isothermal heat transfer to or from a two-phase system is not difficult to achieve in practice since maintaining a constant pressure in the device automatically fixes the temperature at the saturation value. Therefore, processes 1-2 and 3-4 can be approached closely in actual boilers and condensers. Limiting the heat transfer processes to two-phase systems, however, severely limits the maximum temperature that can be used in the cycle (it has to remain under the critical-point value, which is 374°C for water). Limiting the maximum temperature in the cycle also limits the thermal efficiency. Any attempt to raise the maximum temperature in the cycle involves heat transfer to the working fluid in a single phase, which is not easy to accomplish isothermally.

2. The isentropic expansion process (process 2-3) can be approximated closely by a well-designed turbine. However, the quality of the steam decreases during this process, as shown on the *T-s* diagram in Fig. 10–1*a*. Thus the turbine has to handle steam with low quality, that is, steam with a high moisture content. The impingement of liquid droplets on the turbine blades causes erosion and is a major source of wear. Thus steam with qualities less than about 90 percent cannot be tolerated in the operation of power plants.
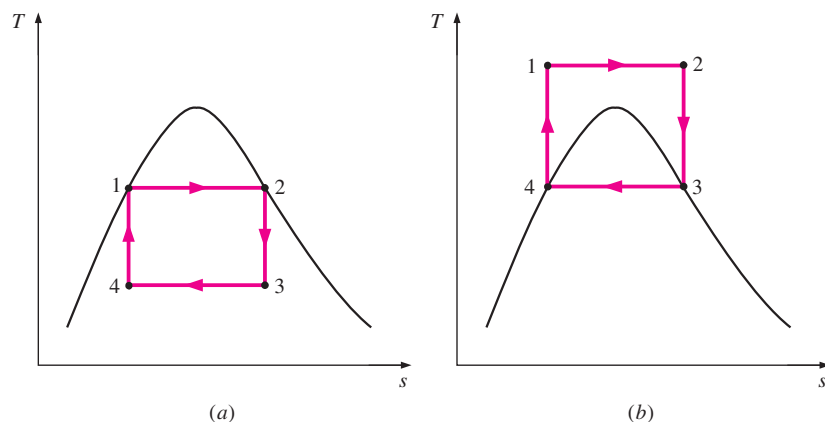

FIGURE 10–1

T-s diagram of two Carnot vapor cycles.

This problem could be eliminated by using a working fluid with a very steep saturated vapor line.

    3. The isentropic compression process (process 4-1) involves the compression of a liquid–vapor mixture to a saturated liquid. There are two difficulties associated with this process. First, it is not easy to control the condensation process so precisely as to end up with the desired quality at state 4. Second, it is not practical to design a compressor that handles two phases.

    Some of these problems could be eliminated by executing the Carnot cycle in a different way, as shown in Fig. 10–1*b*. This cycle, however, presents other problems such as isentropic compression to extremely high pressures and isothermal heat transfer at variable pressures. Thus we conclude that the Carnot cycle cannot be approximated in actual devices and is not a realistic model for vapor power cycles.

10–2 ▪ RANKINE CYCLE: THE IDEAL CYCLE FOR VAPOR POWER CYCLES

Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser, as shown schematically on a *T-s* diagram in Fig. 10–2. The cycle that results is the **Rankine cycle,** which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:

    1-2   Isentropic compression in a pump

    2-3   Constant pressure heat addition in a boiler

    3-4   Isentropic expansion in a turbine
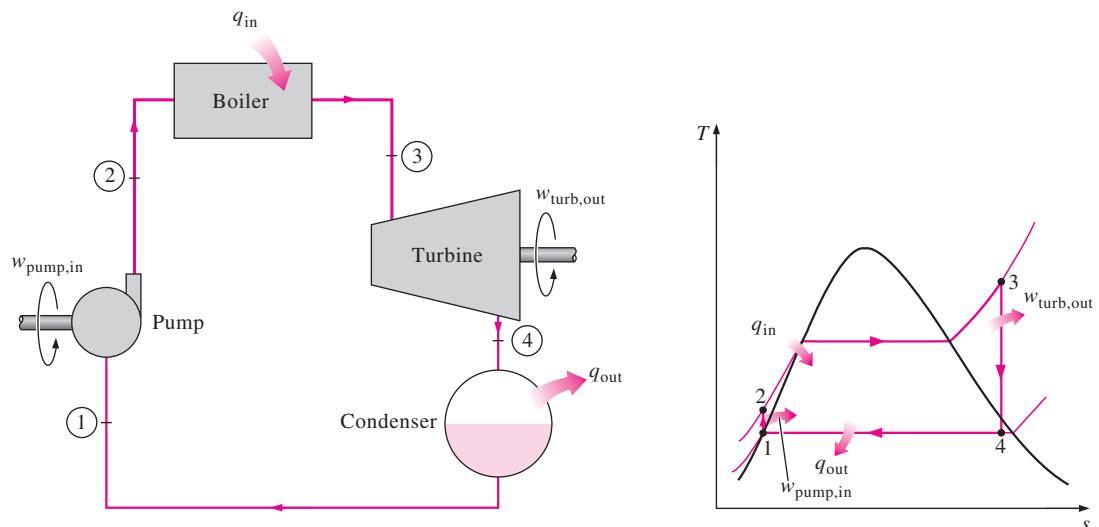
    4-1   Constant pressure heat rejection in a condenser

FIGURE 10–2

The simple ideal Rankine cycle.

Water enters the *pump* at state 1 as saturated liquid and is compressed isentropically to the operating pressure of the boiler. The water temperature increases somewhat during this isentropic compression process due to a slight decrease in the specific volume of water. The vertical distance between states 1 and 2 on the *T-s* diagram is greatly exaggerated for clarity. (If water were truly incompressible, would there be a temperature change at all during this process?)

Water enters the *boiler* as a compressed liquid at state 2 and leaves as a superheated vapor at state 3. The boiler is basically a large heat exchanger where the heat originating from combustion gases, nuclear reactors, or other sources is transferred to the water essentially at constant pressure. The boiler, together with the section where the steam is superheated (the super-heater), is often called the *steam generator*.

The superheated vapor at state 3 enters the *turbine,* where it expands isen-tropically and produces work by rotating the shaft connected to an electric generator. The pressure and the temperature of steam drop during this process to the values at state 4, where steam enters the *condenser.* At this state, steam is usually a saturated liquid–vapor mixture with a high quality. Steam is condensed at constant pressure in the condenser, which is basically a large heat exchanger, by rejecting heat to a cooling medium such as a lake, a river, or the atmosphere. Steam leaves the condenser as saturated liq-uid and enters the pump, completing the cycle. In areas where water is pre-cious, the power plants are cooled by air instead of water. This method of cooling, which is also used in car engines, is called *dry cooling.* Several power plants in the world, including some in the United States, use dry cooling to conserve water.

Remembering that the area under the process curve on a *T-s* diagram represents the heat transfer for internally reversible processes, we see that the area under process curve 2-3 represents the heat transferred to the water in the boiler and the area under the process curve 4-1 represents the heat rejected in the condenser. The difference between these two (the area enclosed by the cycle curve) is the net work produced during the cycle.

Energy Analysis of the Ideal Rankine Cycle

All four components associated with the Rankine cycle (the pump, boiler, turbine, and condenser) are steady-flow devices, and thus all four processes that make up the Rankine cycle can be analyzed as steady-flow processes. The kinetic and potential energy changes of the steam are usually small rel-ative to the work and heat transfer terms and are therefore usually neglected. Then the *steady-flow energy equation* per unit mass of steam reduces to

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i \qquad \text{(kJ/kg)} \qquad \textbf{(10–1)}$$

The boiler and the condenser do not involve any work, and the pump and the turbine are assumed to be isentropic. Then the conservation of energy relation for each device can be expressed as follows:

Pump ($q = 0$): $\qquad\qquad\qquad w_{pump,in} = h_2 - h_1$ $\qquad\qquad\qquad$ **(10–2)**

or,

$$w_{\text{pump,in}} = v(P_2 - P_1) \tag{10–3}$$

where

$$h_1 = h_{f\ @\ P_1} \quad \text{and} \quad v \cong v_1 = v_{f\ @\ P_1} \tag{10–4}$$

Boiler ($w = 0$):
$$q_{\text{in}} = h_3 - h_2 \tag{10–5}$$

Turbine ($q = 0$):
$$w_{\text{turb,out}} = h_3 - h_4 \tag{10–6}$$

Condenser ($w = 0$):
$$q_{\text{out}} = h_4 - h_1 \tag{10–7}$$

The *thermal efficiency* of the Rankine cycle is determined from

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} \tag{10–8}$$

where

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = w_{\text{turb,out}} - w_{\text{pump,in}}$$

The conversion efficiency of power plants in the United States is often expressed in terms of **heat rate,** which is the amount of heat supplied, in Btu's, to generate 1 kWh of electricity. The smaller the heat rate, the greater the efficiency. Considering that 1 kWh = 3412 Btu and disregarding the losses associated with the conversion of shaft power to electric power, the relation between the heat rate and the thermal efficiency can be expressed as

$$\eta_{\text{th}} = \frac{3412\ (\text{Btu/kWh})}{\text{Heat rate}\ (\text{Btu/kWh})} \tag{10–9}$$

For example, a heat rate of 11,363 Btu/kWh is equivalent to 30 percent efficiency.

The thermal efficiency can also be interpreted as the ratio of the area enclosed by the cycle on a *T-s* diagram to the area under the heat-addition process. The use of these relations is illustrated in the following example.

EXAMPLE 10–1   The Simple Ideal Rankine Cycle

Consider a steam power plant operating on the simple ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350°C and is condensed in the condenser at a pressure of 75 kPa. Determine the thermal efficiency of this cycle.

Solution   A steam power plant operating on the simple ideal Rankine cycle is considered. The thermal efficiency of the cycle is to be determined.
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.
Analysis   The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–3. We note that the power plant operates on the ideal Rankine cycle. Therefore, the pump and the turbine are isentropic, there are no pressure drops in the boiler and condenser, and steam leaves the condenser and enters the pump as saturated liquid at the condenser pressure.
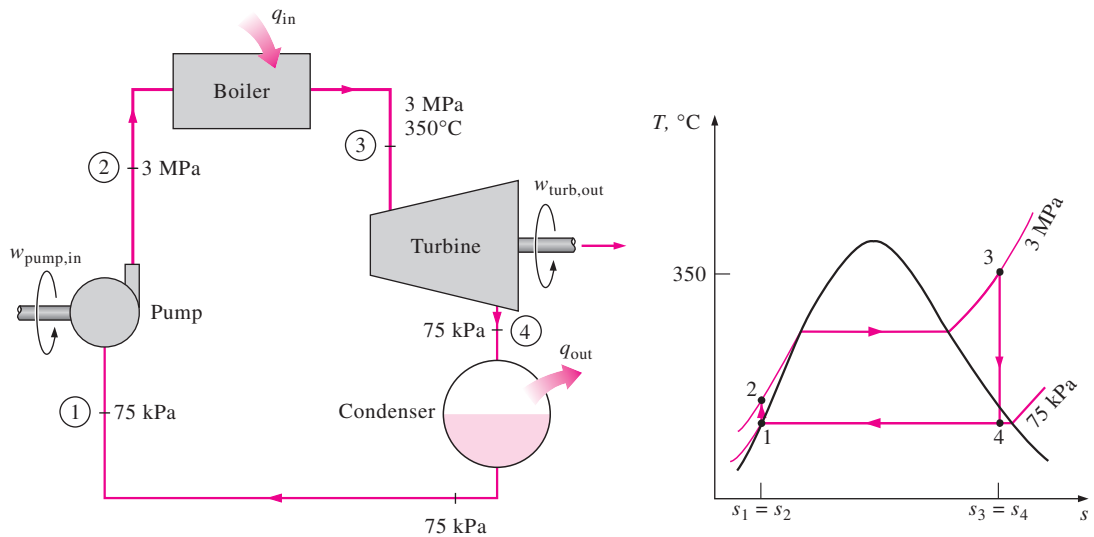
FIGURE 10–3
Schematic and *T-s* diagram for Example 10–1.

It is also interesting to note the thermal efficiency of a Carnot cycle operating between the same temperature limits

$$\eta_{\text{th,Carnot}} = 1 - \frac{T_{\min}}{T_{\max}} = 1 - \frac{(91.76 + 273)\ \text{K}}{(350 + 273)\ \text{K}} = 0.415$$

The difference between the two efficiencies is due to the large external irreversibility in Rankine cycle caused by the large temperature difference between steam and combustion gases in the furnace.

10–3 ▪ DEVIATION OF ACTUAL VAPOR POWER CYCLES FROM IDEALIZED ONES

The actual vapor power cycle differs from the ideal Rankine cycle, as illustrated in Fig. 10–4*a*, as a result of irreversibilities in various components. Fluid friction and heat loss to the surroundings are the two common sources of irreversibilities.

Fluid friction causes pressure drops in the boiler, the condenser, and the piping between various components. As a result, steam leaves the boiler at a somewhat lower pressure. Also, the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to the pressure drop in the connecting pipes. The pressure drop in the condenser is usually very small. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle calls for. This requires a larger pump and larger work input to the pump.

The other major source of irreversibility is the *heat loss* from the steam to the surroundings as the steam flows through various components. To maintain the same level of net work output, more heat needs to be transferred to

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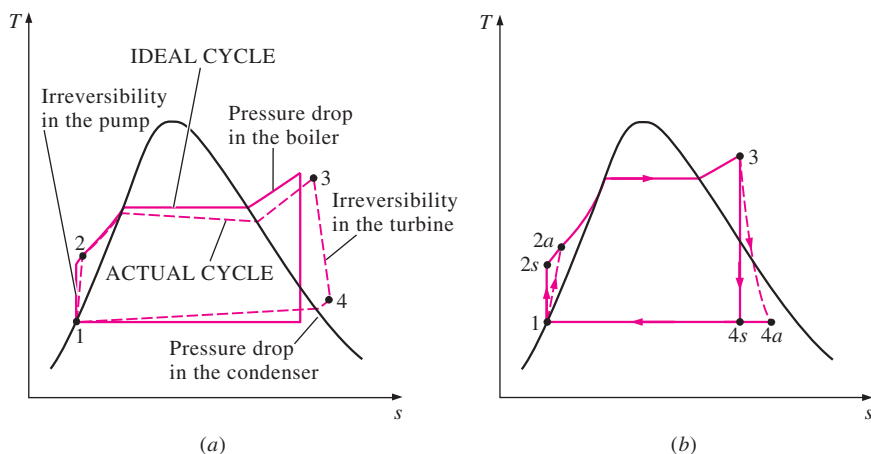
FIGURE 10–4

(*a*) Deviation of actual vapor power cycle from the ideal Rankine cycle. (*b*) The effect of pump and turbine irreversibilities on the ideal Rankine cycle.

the steam in the boiler to compensate for these undesired heat losses. As a result, cycle efficiency decreases.

Of particular importance are the irreversibilities occurring within the pump and the turbine. A pump requires a greater work input, and a turbine produces a smaller work output as a result of irreversibilities. Under ideal conditions, the flow through these devices is isentropic. The deviation of actual pumps and turbines from the isentropic ones can be accounted for by utilizing *isentropic efficiencies,* defined as

$$\eta_P = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1} \tag{10–10}$$

and

$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \tag{10–11}$$

where states $2a$ and $4a$ are the actual exit states of the pump and the turbine, respectively, and $2s$ and $4s$ are the corresponding states for the isentropic case (Fig. 10–4*b*).

Other factors also need to be considered in the analysis of actual vapor power cycles. In actual condensers, for example, the liquid is usually sub-cooled to prevent the onset of *cavitation,* the rapid vaporization and conden-sation of the fluid at the low-pressure side of the pump impeller, which may damage it. Additional losses occur at the bearings between the moving parts as a result of friction. Steam that leaks out during the cycle and air that leaks into the condenser represent two other sources of loss. Finally, the power consumed by the auxiliary equipment such as fans that supply air to the furnace should also be considered in evaluating the overall performance of power plants.

The effect of irreversibilities on the thermal efficiency of a steam power cycle is illustrated below with an example.

EXAMPLE 10–2   An Actual Steam Power Cycle

A steam power plant operates on the cycle shown in Fig. 10–5. If the isentropic efficiency of the turbine is 87 percent and the isentropic efficiency of the pump is 85 percent, determine (a) the thermal efficiency of the cycle and (b) the net power output of the plant for a mass flow rate of 15 kg/s.

Solution   A steam power cycle with specified turbine and pump efficiencies is considered. The thermal efficiency and the net power output are to be determined.
Assumptions   **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.
Analysis   The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–5. The temperatures and pressures of steam at various points are also indicated on the figure. We note that the power plant involves steady-flow components and operates on the Rankine cycle, but the imperfections at various components are accounted for.

(a) The thermal efficiency of a cycle is the ratio of the net work output to the heat input, and it is determined as follows:

Pump work input:

$$w_{pump,in} = \frac{w_{s,pump,in}}{\eta_p} = \frac{v_1(P_2 - P_1)}{\eta_p}$$

$$= \frac{(0.001009 \text{ m}^3/\text{kg})[(16{,}000 - 9) \text{ kPa}]}{0.85}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 19.0 \text{ kJ/kg}$$

FIGURE 10–5
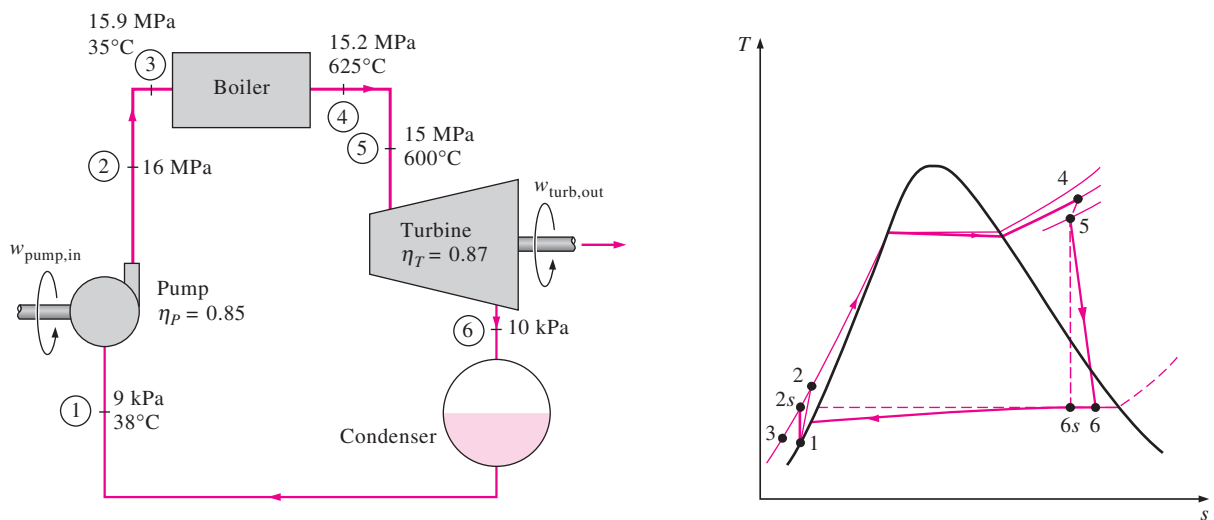Schematic and *T-s* diagram for Example 10–2.

Turbine work output:

$$w_{\text{turb,out}} = \eta_T w_{s,\text{turb,out}}$$
$$= \eta_T(h_5 - h_{6s}) = 0.87(3583.1 - 2115.3) \text{ kJ/kg}$$
$$= 1277.0 \text{ kJ/kg}$$

Boiler heat input: $\quad q_{\text{in}} = h_4 - h_3 = (3647.6 - 160.1) \text{ kJ/kg} = 3487.5 \text{ kJ/kg}$

Thus,

$$w_{\text{net}} = w_{\text{turb,out}} - w_{\text{pump,in}} = (1277.0 - 19.0) \text{ kJ/kg} = 1258.0 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1258.0 \text{ kJ/kg}}{3487.5 \text{ kJ/kg}} = \textbf{0.361 or 36.1\%}$$

(*b*) The power produced by this power plant is

$$\dot{W}_{\text{net}} = \dot{m}(w_{\text{net}}) = (15 \text{ kg/s})(1258.0 \text{ kJ/kg}) = \textbf{18.9 MW}$$

Discussion Without the irreversibilities, the thermal efficiency of this cycle would be 43.0 percent (see Example 10–3*c*).

10–4 ▪ HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

Steam power plants are responsible for the production of most electric power in the world, and even small increases in thermal efficiency can mean large savings from the fuel requirements. Therefore, every effort is made to improve the efficiency of the cycle on which steam power plants operate.

The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same: *Increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser.* That is, the average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection. Next we discuss three ways of accomplishing this for the simple ideal Rankine cycle.
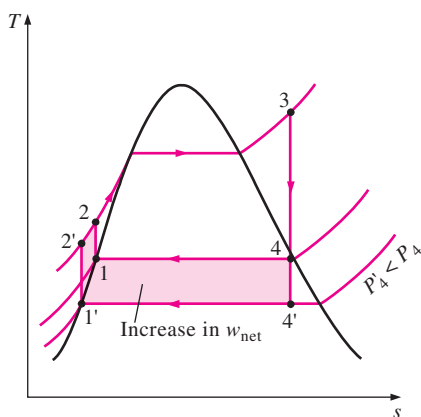
Lowering the Condenser Pressure (*Lowers* $T_{\text{low,avg}}$)

Steam exists as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. Therefore, lowering the operating pressure of the condenser automatically lowers the temperature of the steam, and thus the temperature at which heat is rejected.

The effect of lowering the condenser pressure on the Rankine cycle efficiency is illustrated on a *T-s* diagram in Fig. 10–6. For comparison purposes, the turbine inlet state is maintained the same. The colored area on this diagram represents the increase in net work output as a result of lowering the condenser pressure from P_4 to P_4'. The heat input requirements also increase (represented by the area under curve 2′-2), but this increase is very small. Thus the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle.

FIGURE 10–6

The effect of lowering the condenser pressure on the ideal Rankine cycle.

To take advantage of the increased efficiencies at low pressures, the condensers of steam power plants usually operate well below the atmospheric pressure. This does not present a major problem since the vapor power cycles operate in a closed loop. However, there is a lower limit on the condenser pressure that can be used. It cannot be lower than the saturation pressure corresponding to the temperature of the cooling medium. Consider, for example, a condenser that is to be cooled by a nearby river at 15°C. Allowing a temperature difference of 10°C for effective heat transfer, the steam temperature in the condenser must be above 25°C; thus the condenser pressure must be above 3.2 kPa, which is the saturation pressure at 25°C.

Lowering the condenser pressure is not without any side effects, however. For one thing, it creates the possibility of air leakage into the condenser. More importantly, it increases the moisture content of the steam at the final stages of the turbine, as can be seen from Fig. 10–6. The presence of large quantities of moisture is highly undesirable in turbines because it decreases the turbine efficiency and erodes the turbine blades. Fortunately, this problem can be corrected, as discussed next.

Superheating the Steam to High Temperatures (*Increases T*_{high,avg})

The average temperature at which heat is transferred to steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of vapor power cycles is illustrated on a *T-s* diagram in Fig. 10–7. The colored area on this diagram represents the increase in the net work. The total area under the process curve 3-3′ represents the increase in the heat input. Thus both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency, however, since the average temperature at which heat is added increases.

Superheating the steam to higher temperatures has another very desirable effect: It decreases the moisture content of the steam at the turbine exit, as can be seen from the *T-s* diagram (the quality at state 4′ is higher than that at state 4).

The temperature to which steam can be superheated is limited, however, by metallurgical considerations. Presently the highest steam temperature allowed at the turbine inlet is about 620°C (1150°F). Any increase in this value depends on improving the present materials or finding new ones that can withstand higher temperatures. Ceramics are very promising in this regard.

Increasing the Boiler Pressure (*Increases T*_{high,avg})

Another way of increasing the average temperature during the heat-addition process is to increase the operating pressure of the boiler, which automatically raises the temperature at which boiling takes place. This, in turn, raises the average temperature at which heat is transferred to the steam and thus raises the thermal efficiency of the cycle.

The effect of increasing the boiler pressure on the performance of vapor power cycles is illustrated on a *T-s* diagram in Fig. 10–8. Notice that for a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This undesirable side effect can be corrected, however, by reheating the steam, as discussed in the next section.
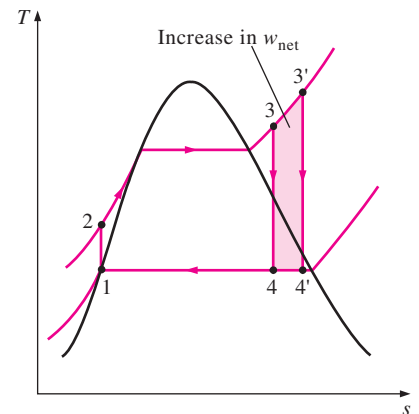

FIGURE 10–7

The effect of superheating the steam to higher temperatures on the ideal Rankine cycle.
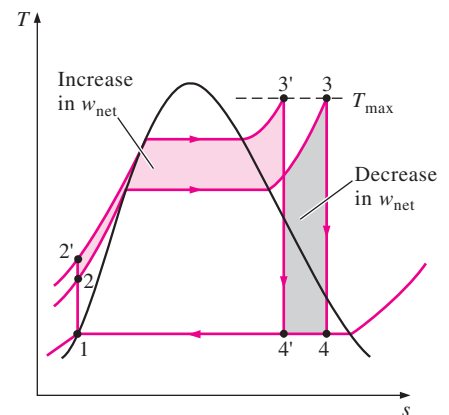

FIGURE 10–8

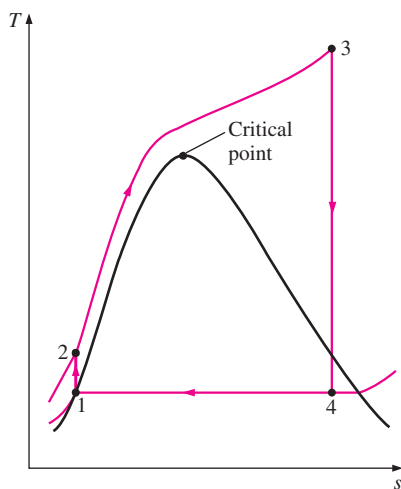The effect of increasing the boiler pressure on the ideal Rankine cycle.

FIGURE 10–9

A supercritical Rankine cycle.

Operating pressures of boilers have gradually increased over the years from about 2.7 MPa (400 psia) in 1922 to over 30 MPa (4500 psia) today, generating enough steam to produce a net power output of 1000 MW or more in a large power plant. Today many modern steam power plants operate at supercritical pressures ($P > 22.06$ MPa) and have thermal efficiencies of about 40 percent for fossil-fuel plants and 34 percent for nuclear plants. There are over 150 supercritical-pressure steam power plants in operation in the United States. The lower efficiencies of nuclear power plants are due to the lower maximum temperatures used in those plants for safety reasons. The *T-s* diagram of a supercritical Rankine cycle is shown in Fig. 10–9.

The effects of lowering the condenser pressure, superheating to a higher temperature, and increasing the boiler pressure on the thermal efficiency of the Rankine cycle are illustrated below with an example.

EXAMPLE 10–3   Effect of Boiler Pressure and Temperature on Efficiency

Consider a steam power plant operating on the ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350°C and is condensed in the condenser at a pressure of 10 kPa. Determine (*a*) the thermal efficiency of this power plant, (*b*) the thermal efficiency if steam is superheated to 600°C instead of 350°C, and (*c*) the thermal efficiency if the boiler pressure is raised to 15 MPa while the turbine inlet temperature is maintained at 600°C.

Solution   A steam power plant operating on the ideal Rankine cycle is considered. The effects of superheating the steam to a higher temperature and raising the boiler pressure on thermal efficiency are to be investigated.
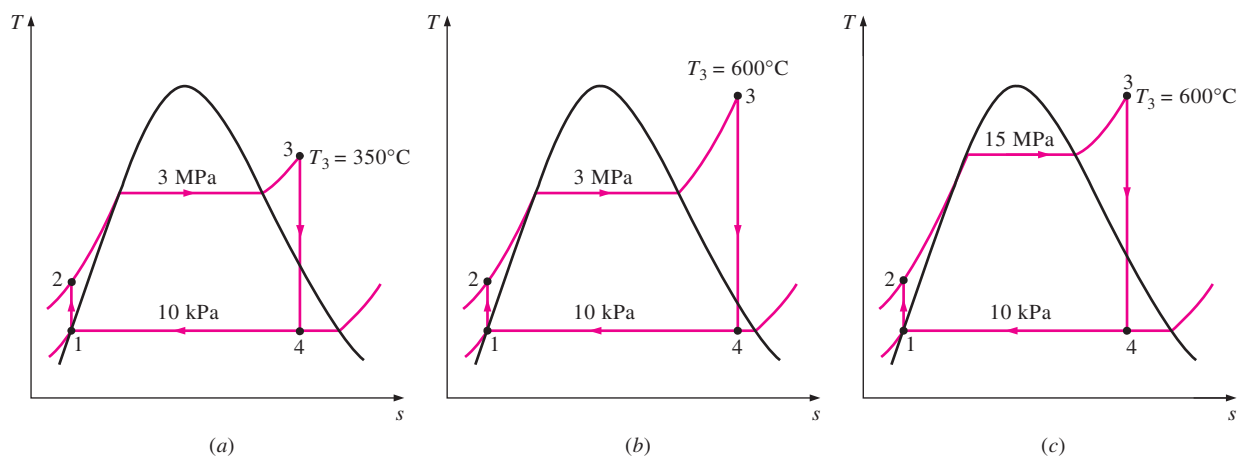**Analysis**   The *T-s* diagrams of the cycle for all three cases are given in Fig. 10–10.

(a)   (b)   (c)

FIGURE 10–10

T-s diagrams of the three cycles discussed in Example 10–3.

(*a*) This is the steam power plant discussed in Example 10–1, except that the condenser pressure is lowered to 10 kPa. The thermal efficiency is determined in a similar manner:

State 1: $\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\}$ $\begin{array}{l} h_1 = h_{f\,@\,10\,\text{kPa}} = 191.81 \text{ kJ/kg} \\ v_1 = v_{f\,@\,10\,\text{kPa}} = 0.00101 \text{ m}^3/\text{kg} \end{array}$

State 2: $\begin{array}{l} P_2 = 3 \text{ MPa} \\ s_2 = s_1 \end{array}$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg})[(3000 - 10)\text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

$$= 3.02 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (191.81 + 3.02) \text{ kJ/kg} = 194.83 \text{ kJ/kg}$$

State 3: $\left.\begin{array}{l} P_3 = 3 \text{ MPa} \\ T_3 = 350°C \end{array}\right\}$ $\begin{array}{l} h_3 = 3116.1 \text{ kJ/kg} \\ s_3 = 6.7450 \text{ kJ/kg}\cdot\text{K} \end{array}$

State 4: $\begin{array}{l} P_4 = 10 \text{ kPa} \quad (\text{sat. mixture}) \\ s_4 = s_3 \end{array}$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 0.6492}{7.4996} = 0.8128$$

Thus,

$$h_4 = h_f + x_4 h_{fg} = 191.81 + 0.8128(2392.1) = 2136.1 \text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = (3116.1 - 194.83) \text{ kJ/kg} = 2921.3 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = (2136.1 - 191.81) \text{ kJ/kg} = 1944.3 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1944.3 \text{ kJ/kg}}{2921.3 \text{ kJ/kg}} = \textbf{0.334 or 33.4\%}$$

Therefore, the thermal efficiency increases from 26.0 to 33.4 percent as a result of lowering the condenser pressure from 75 to 10 kPa. At the same time, however, the quality of the steam decreases from 88.6 to 81.3 percent (in other words, the moisture content increases from 11.4 to 18.7 percent).

(*b*) States 1 and 2 remain the same in this case, and the enthalpies at state 3 (3 MPa and 600°C) and state 4 (10 kPa and $s_4 = s_3$) are determined to be

$$h_3 = 3682.8 \text{ kJ/kg}$$

$$h_4 = 2380.3 \text{ kJ/kg} \quad (x_4 = 0.915)$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3682.8 - 194.83 = 3488.0 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 2380.3 - 191.81 = 2188.5 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2188.5 \text{ kJ/kg}}{3488.0 \text{ kJ/kg}} = \textbf{0.373 or 37.3\%}$$

Therefore, the thermal efficiency increases from 33.4 to 37.3 percent as a result of superheating the steam from 350 to 600°C. At the same time, the quality of the steam increases from 81.3 to 91.5 percent (in other words, the moisture content decreases from 18.7 to 8.5 percent).

(*c*) State 1 remains the same in this case, but the other states change. The enthalpies at state 2 (15 MPa and $s_2 = s_1$), state 3 (15 MPa and 600°C), and state 4 (10 kPa and $s_4 = s_3$) are determined in a similar manner to be

$$h_2 = 206.95 \text{ kJ/kg}$$

$$h_3 = 3583.1 \text{ kJ/kg}$$

$$h_4 = 2115.3 \text{ kJ/kg} \qquad (x_4 = 0.804)$$

Thus,

$$q_{in} = h_3 - h_2 = 3583.1 - 206.95 = 3376.2 \text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = 2115.3 - 191.81 = 1923.5 \text{ kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1923.5 \text{ kJ/kg}}{3376.2 \text{ kJ/kg}} = \textbf{0.430 or 43.0\%}$$

Discussion The thermal efficiency increases from 37.3 to 43.0 percent as a result of raising the boiler pressure from 3 to 15 MPa while maintaining the turbine inlet temperature at 600°C. At the same time, however, the quality of the steam decreases from 91.5 to 80.4 percent (in other words, the moisture content increases from 8.5 to 19.6 percent).

10–5 · THE IDEAL REHEAT RANKINE CYCLE

We noted in the last section that increasing the boiler pressure increases the thermal efficiency of the Rankine cycle, but it also increases the moisture content of the steam to unacceptable levels. Then it is natural to ask the following question:

How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?

Two possibilities come to mind:

1. Superheat the steam to very high temperatures before it enters the turbine. This would be the desirable solution since the average temperature at which heat is added would also increase, thus increasing the cycle efficiency. This is not a viable solution, however, since it requires raising the steam temperature to metallurgically unsafe levels.

2. Expand the steam in the turbine in two stages, and reheat it in between. In other words, modify the simple ideal Rankine cycle with a **reheat** process. Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants.

The *T-s* diagram of the ideal reheat Rankine cycle and the schematic of the power plant operating on this cycle are shown in Fig. 10–11. The ideal reheat Rankine cycle differs from the simple ideal Rankine cycle in that the

expansion process takes place in two stages. In the first stage (the high-pressure turbine), steam is expanded isentropically to an intermediate pressure and sent back to the boiler where it is reheated at constant pressure, usually to the inlet temperature of the first turbine stage. Steam then expands isentropically in the second stage (low-pressure turbine) to the condenser pressure. Thus the total heat input and the total turbine work output for a reheat cycle become

$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4) \qquad \textbf{(10–12)}$$

and

$$w_{turb,out} = w_{turb,I} + w_{turb,II} = (h_3 - h_4) + (h_5 - h_6) \qquad \textbf{(10–13)}$$

The incorporation of the single reheat in a modern power plant improves the cycle efficiency by 4 to 5 percent by increasing the average temperature at which heat is transferred to the steam.

The average temperature during the reheat process can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach an isothermal process at the maximum temperature, as shown in Fig. 10–12. The use of more than two reheat stages, however, is not practical. The theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat. If the turbine inlet pressure is not high enough, double reheat would result in superheated exhaust. This is undesirable as it would cause the average temperature for heat rejection to increase and thus the cycle efficiency to decrease. Therefore, double reheat is used only on supercritical-pressure ($P > 22.06$ MPa) power plants. A third reheat stage would increase the cycle efficiency by about half of the improvement attained by the second reheat. This gain is too small to justify the added cost and complexity.
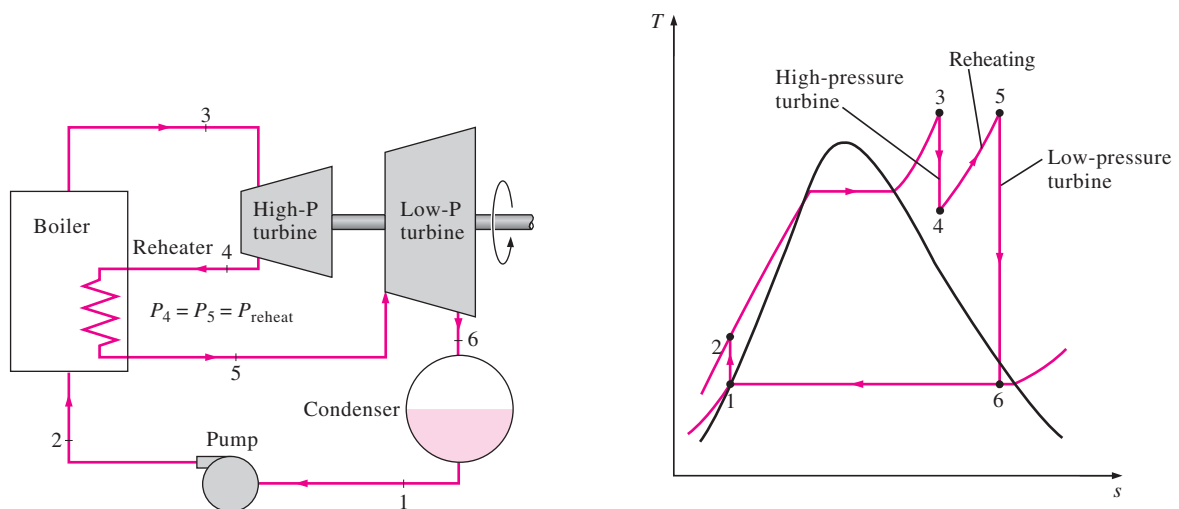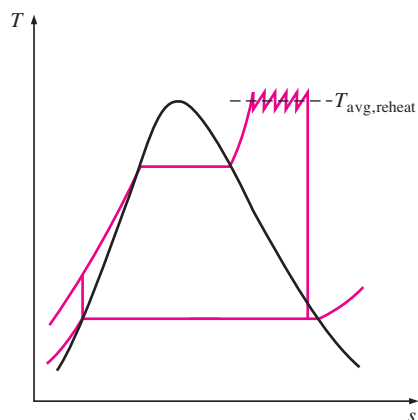

FIGURE 10–11

The ideal reheat Rankine cycle.

FIGURE 10–12

The average temperature at which heat is transferred during reheating increases as the number of reheat stages is increased.

The reheat cycle was introduced in the mid-1920s, but it was abandoned in the 1930s because of the operational difficulties. The steady increase in boiler pressures over the years made it necessary to reintroduce single reheat in the late 1940s and double reheat in the early 1950s.

The reheat temperatures are very close or equal to the turbine inlet temperature. The optimum reheat pressure is about one-fourth of the maximum cycle pressure. For example, the optimum reheat pressure for a cycle with a boiler pressure of 12 MPa is about 3 MPa.

Remember that the sole purpose of the reheat cycle is to reduce the moisture content of the steam at the final stages of the expansion process. If we had materials that could withstand sufficiently high temperatures, there would be no need for the reheat cycle.

EXAMPLE 10–4   **The Ideal Reheat Rankine Cycle**

Consider a steam power plant operating on the ideal reheat Rankine cycle. Steam enters the high-pressure turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. If the moisture content of the steam at the exit of the low-pressure turbine is not to exceed 10.4 percent, determine (*a*) the pressure at which the steam should be reheated and (*b*) the thermal efficiency of the cycle. Assume the steam is reheated to the inlet temperature of the high-pressure turbine.

Solution   A steam power plant operating on the ideal reheat Rankine cycle is considered. For a specified moisture content at the turbine exit, the reheat pressure and the thermal efficiency are to be determined.

Assumptions   **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis   The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–13. We note that the power plant operates on the ideal reheat Rankine cycle. Therefore, the pump and the turbines are isentropic, there are no pressure drops in the boiler and condenser, and steam leaves the condenser and enters the pump as saturated liquid at the condenser pressure.

(*a*) The reheat pressure is determined from the requirement that the entropies at states 5 and 6 be the same:

State 6: $P_6 = 10$ kPa

$$x_6 = 0.896 \quad \text{(sat. mixture)}$$

$$s_6 = s_f + x_6 s_{fg} = 0.6492 + 0.896(7.4996) = 7.3688 \text{ kJ/kg} \cdot \text{K}$$

Also,

$$h_6 = h_f + x_6 h_{fg} = 191.81 + 0.896(2392.1) = 2335.1 \text{ kJ/kg}$$

Thus,

State 5: $\left. \begin{array}{l} T_5 = 600°C \\ s_5 = s_6 \end{array} \right\}$ $\begin{array}{l} P_5 = \textbf{4.0 MPa} \\ h_5 = 3674.9 \text{ kJ/kg} \end{array}$

Therefore, steam should be reheated at a pressure of 4 MPa or lower to prevent a moisture content above 10.4 percent.

(*b*) To determine the thermal efficiency, we need to know the enthalpies at all other states:

State 1: $P_1 = 10 \text{ kPa}$ $h_1 = h_{f @ 10 \text{ kPa}} = 191.81 \text{ kJ/kg}$
             Sat. liquid $v_1 = v_{f @ 10 \text{ kPa}} = 0.00101 \text{ m}^3/\text{kg}$

State 2: $P_2 = 15 \text{ MPa}$

$$s_2 = s_1$$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg})$$

$$\times \left[(15{,}000 - 10)\text{kPa}\right]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 15.14 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (191.81 + 15.14) \text{ kJ/kg} = 206.95 \text{ kJ/kg}$$

State 3: $P_3 = 15 \text{ MPa}$ $h_3 = 3583.1 \text{ kJ/kg}$
             $T_3 = 600°C$ $s_3 = 6.6796 \text{ kJ/kg} \cdot \text{K}$

State 4: $P_4 = 4 \text{ MPa}$ $h_4 = 3155.0 \text{ kJ/kg}$
             $s_4 = s_3$ $(T_4 = 375.5°C)$

Thus

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4)$$

$$= (3583.1 - 206.95) \text{ kJ/kg} + (3674.9 - 3155.0) \text{ kJ/kg}$$

$$= 3896.1 \text{ kJ/kg}$$

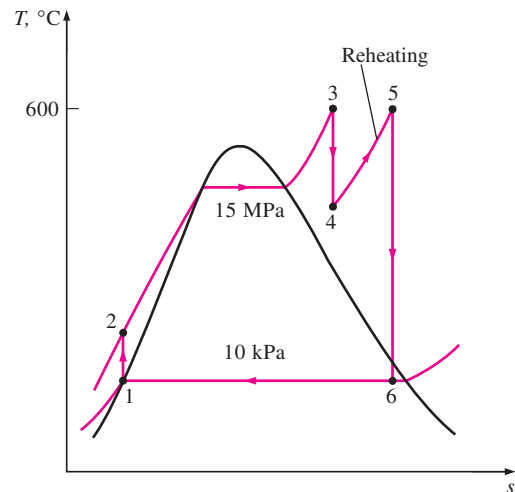$$q_{\text{out}} = h_6 - h_1 = (2335.1 - 191.81) \text{ kJ/kg}$$

$$= 2143.3 \text{ kJ/kg}$$

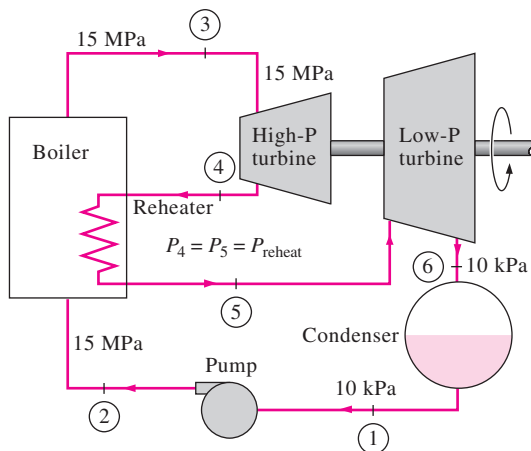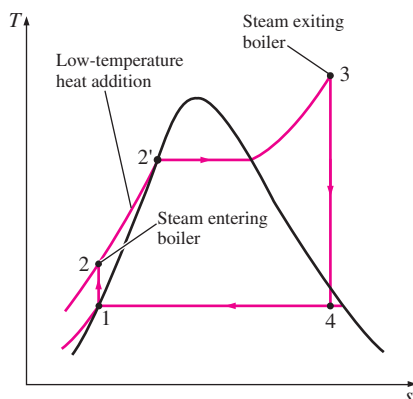**FIGURE 10–13**
Schematic and *T-s* diagram for Example 10–4.

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{2143.3 \text{ kJ/kg}}{3896.1 \text{ kJ/kg}} = \textbf{0.450 or 45.0\%}$$

Discussion This problem was solved in Example 10–3c for the same pressure and temperature limits but without the reheat process. A comparison of the two results reveals that reheating reduces the moisture content from 19.6 to 10.4 percent while increasing the thermal efficiency from 43.0 to 45.0 percent.

FIGURE 10–14

The first part of the heat-addition process in the boiler takes place at relatively low temperatures.

10–6 ▪ THE IDEAL REGENERATIVE RANKINE CYCLE

A careful examination of the *T-s* diagram of the Rankine cycle redrawn in Fig. 10–14 reveals that heat is transferred to the working fluid during process 2-2′ at a relatively low temperature. This lowers the average heat-addition temperature and thus the cycle efficiency.

To remedy this shortcoming, we look for ways to raise the temperature of the liquid leaving the pump (called the *feedwater*) before it enters the boiler. One such possibility is to transfer heat to the feedwater from the expanding steam in a counterflow heat exchanger built into the turbine, that is, to use **regeneration.** This solution is also impractical because it is difficult to design such a heat exchanger and because it would increase the moisture content of the steam at the final stages of the turbine.

A practical regeneration process in steam power plants is accomplished by extracting, or "bleeding," steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feedwater instead. The device where the feedwater is heated by regeneration is called a **regenerator,** or a **feedwater heater (FWH).**

Regeneration not only improves cycle efficiency, but also provides a convenient means of deaerating the feedwater (removing the air that leaks in at the condenser) to prevent corrosion in the boiler. It also helps control the large volume flow rate of the steam at the final stages of the turbine (due to the large specific volumes at low pressures). Therefore, regeneration has been used in all modern steam power plants since its introduction in the early 1920s.

A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (open feedwater heaters) or without mixing them (closed feedwater heaters). Regeneration with both types of feedwater heaters is discussed below.

Open Feedwater Heaters

An **open** (or **direct-contact**) **feedwater heater** is basically a *mixing chamber,* where the steam extracted from the turbine mixes with the feedwater exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure. The schematic of a steam power plant with one open feedwater heater (also called *single-stage regenerative cycle*) and the *T-s* diagram of the cycle are shown in Fig. 10–15.

In an ideal regenerative Rankine cycle, steam enters the turbine at the boiler pressure (state 5) and expands isentropically to an intermediate pres-

sure (state 6). Some steam is extracted at this state and routed to the feed-water heater, while the remaining steam continues to expand isentropically to the condenser pressure (state 7). This steam leaves the condenser as a saturated liquid at the condenser pressure (state 1). The condensed water, which is also called the *feedwater,* then enters an isentropic pump, where it is compressed to the feedwater heater pressure (state 2) and is routed to the feedwater heater, where it mixes with the steam extracted from the turbine. The fraction of the steam extracted is such that the mixture leaves the heater as a saturated liquid at the heater pressure (state 3). A second pump raises the pressure of the water to the boiler pressure (state 4). The cycle is completed by heating the water in the boiler to the turbine inlet state (state 5).

In the analysis of steam power plants, it is more convenient to work with quantities expressed per unit mass of the steam flowing through the boiler. For each 1 kg of steam leaving the boiler, y kg expands partially in the turbine and is extracted at state 6. The remaining $(1 - y)$ kg expands completely to the condenser pressure. Therefore, the mass flow rates are different in different components. If the mass flow rate through the boiler is $\dot{m}$, for example, it is $(1 - y)\dot{m}$ through the condenser. This aspect of the regenerative Rankine cycle should be considered in the analysis of the cycle as well as in the interpretation of the areas on the *T-s* diagram. In light of Fig. 10–15, the heat and work interactions of a regenerative Rankine cycle with one feed-water heater can be expressed per unit mass of steam flowing through the boiler as follows:

$$q_{\text{in}} = h_5 - h_4 \qquad (10\text{–}14)$$

$$q_{\text{out}} = (1 - y)(h_7 - h_1) \qquad (10\text{–}15)$$

$$w_{\text{turb,out}} = (h_5 - h_6) + (1 - y)(h_6 - h_7) \qquad (10\text{–}16)$$

$$w_{\text{pump,in}} = (1 - y)w_{\text{pump I,in}} + w_{\text{pump II,in}} \qquad (10\text{–}17)$$
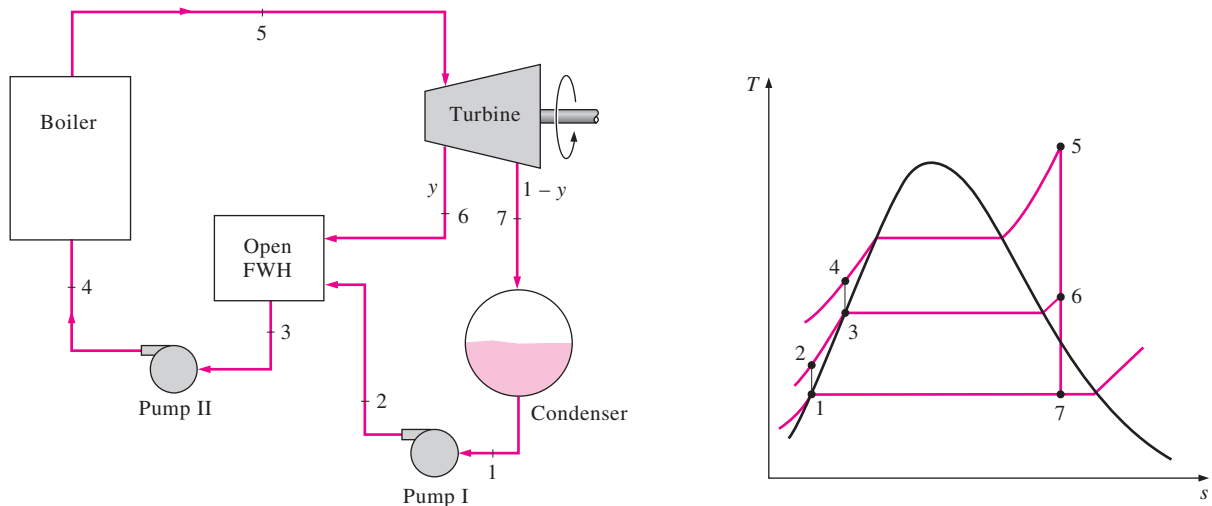

FIGURE 10–15
The ideal regenerative Rankine cycle with an open feedwater heater.

where

$$y = \dot{m}_6/\dot{m}_5 \qquad \text{(fraction of steam extracted)}$$

$$w_{\text{pump I,in}} = v_1(P_2 - P_1)$$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3)$$

The thermal efficiency of the Rankine cycle increases as a result of regeneration. This is because regeneration raises the average temperature at which heat is transferred to the steam in the boiler by raising the temperature of the water before it enters the boiler. The cycle efficiency increases further as the number of feedwater heaters is increased. Many large plants in operation today use as many as eight feedwater heaters. The optimum number of feedwater heaters is determined from economical considerations. The use of an additional feedwater heater cannot be justified unless it saves more from the fuel costs than its own cost.

Closed Feedwater Heaters

Another type of feedwater heater frequently used in steam power plants is the **closed feedwater heater,** in which heat is transferred from the extracted steam to the feedwater without any mixing taking place. The two streams now can be at different pressures, since they do not mix. The schematic of a steam power plant with one closed feedwater heater and the *T-s* diagram of the cycle are shown in Fig. 10–16. In an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure. In actual power plants, the feedwater leaves the heater below the exit tempera-
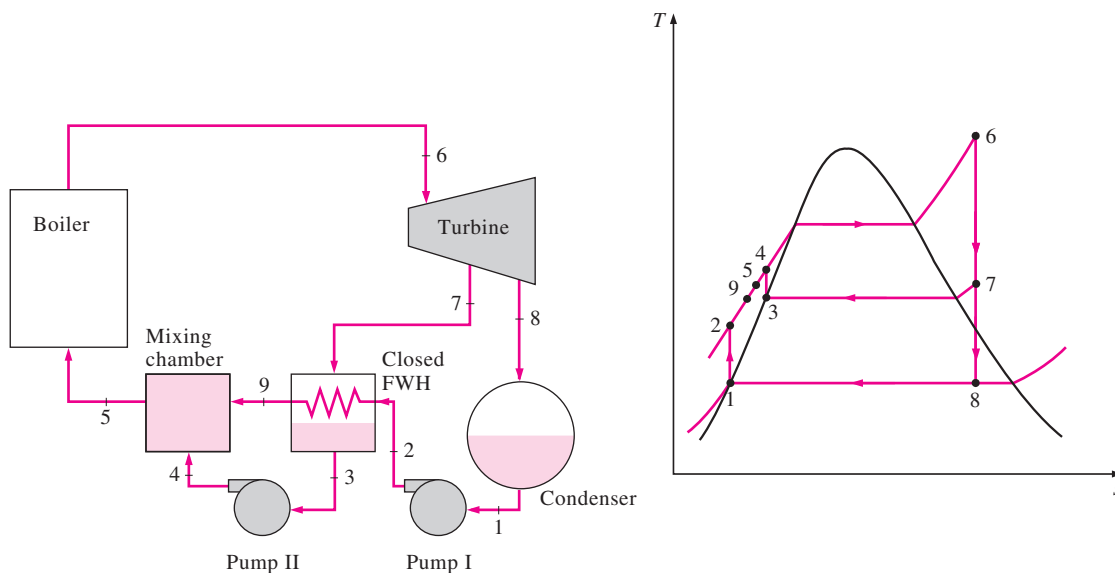

FIGURE 10–16
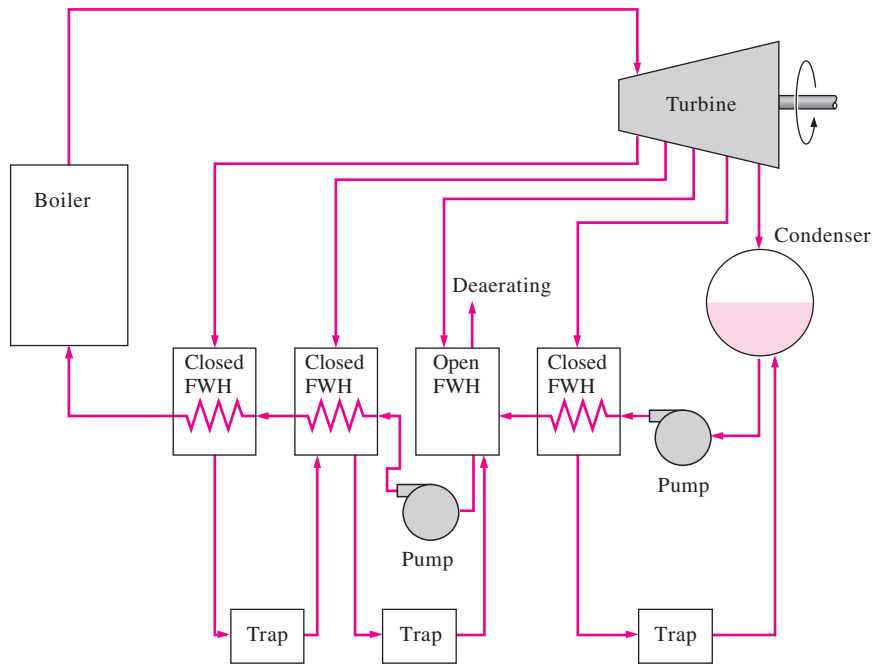The ideal regenerative Rankine cycle with a closed feedwater heater.

FIGURE 10–17

A steam power plant with one open and three closed feedwater heaters.

ture of the extracted steam because a temperature difference of at least a few degrees is required for any effective heat transfer to take place.

The condensed steam is then either pumped to the feedwater line or routed to another heater or to the condenser through a device called a **trap.** A trap allows the liquid to be throttled to a lower pressure region but *traps* the vapor. The enthalpy of steam remains constant during this throttling process.

The open and closed feedwater heaters can be compared as follows. Open feedwater heaters are simple and inexpensive and have good heat transfer characteristics. They also bring the feedwater to the saturation state. For each heater, however, a pump is required to handle the feedwater. The closed feedwater heaters are more complex because of the internal tubing network, and thus they are more expensive. Heat transfer in closed feedwater heaters is also less effective since the two streams are not allowed to be in direct contact. However, closed feedwater heaters do not require a separate pump for each heater since the extracted steam and the feedwater can be at different pressures. Most steam power plants use a combination of open and closed feedwater heaters, as shown in Fig. 10–17.

EXAMPLE 10–5   **The Ideal Regenerative Rankine Cycle**

Consider a steam power plant operating on the ideal regenerative Rankine cycle with one open feedwater heater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa.

Some steam leaves the turbine at a pressure of 1.2 MPa and enters the open feedwater heater. Determine the fraction of steam extracted from the turbine and the thermal efficiency of the cycle.

Solution A steam power plant operates on the ideal regenerative Rankine cycle with one open feedwater heater. The fraction of steam extracted from the turbine and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–18. We note that the power plant operates on the ideal regenerative Rankine cycle. Therefore, the pumps and the turbines are isentropic; there are no pressure drops in the boiler, condenser, and feedwater heater; and steam leaves the condenser and the feedwater heater as saturated liquid. First, we determine the enthalpies at various states:

State 1: $\left. \begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array} \right\}$ $\begin{array}{l} h_1 = h_{f\ @\ 10\ kPa} = 191.81 \text{ kJ/kg} \\ v_1 = v_{f\ @\ 10\ kPa} = 0.00101 \text{ m}^3/\text{kg} \end{array}$

State 2: $P_2 = 1.2 \text{ MPa}$

$s_2 = s_1$

$$w_{\text{pump I,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg})[(1200 - 10) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 1.20 \text{ kJ/kg}$$

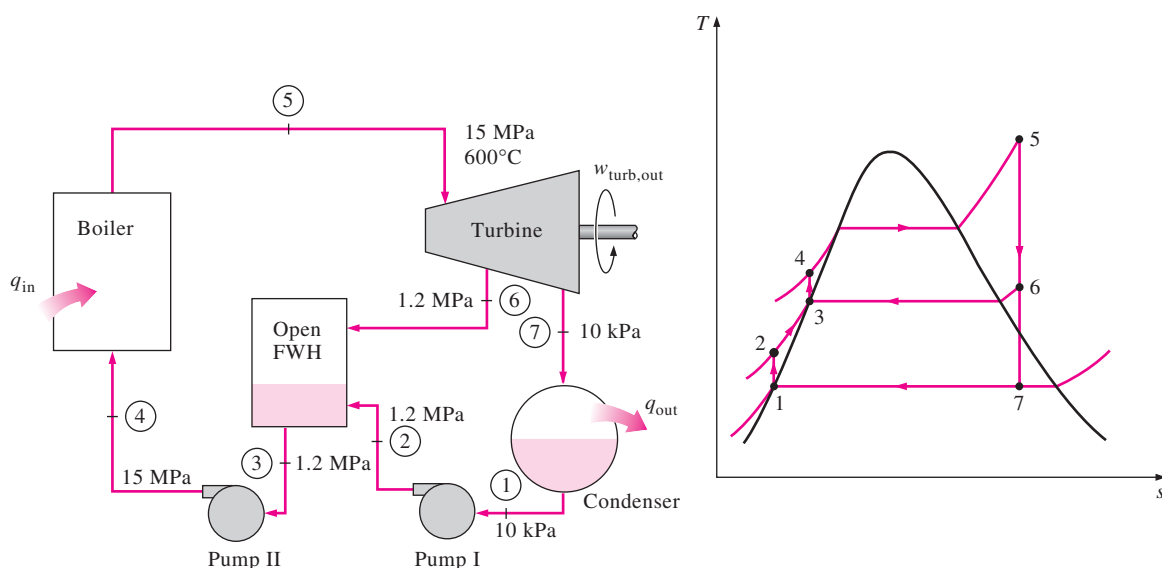$$h_2 = h_1 + w_{\text{pump I,in}} = (191.81 + 1.20) \text{ kJ/kg} = 193.01 \text{ kJ/kg}$$

FIGURE 10–18
Schematic and *T-s* diagram for Example 10–5.

State 3: $P_3 = 1.2 \text{ MPa}$ \} $\quad v_3 = v_{f @ 1.2 \text{ MPa}} = 0.001138 \text{ m}^3/\text{kg}$
$\quad\quad\quad$ Sat. liquid $\quad\quad h_3 = h_{f @ 1.2 \text{ MPa}} = 798.33 \text{ kJ/kg}$

State 4: $P_4 = 15 \text{ MPa}$

$$s_4 = s_3$$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3)$$

$$= (0.001138 \text{ m}^3/\text{kg})[(15,000 - 1200) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 15.70 \text{ kJ/kg}$$

$$h_4 = h_3 + w_{\text{pump II,in}} = (798.33 + 15.70) \text{ kJ/kg} = 814.03 \text{ kJ/kg}$$

State 5: $P_5 = 15 \text{ MPa}$ \} $\quad h_5 = 3583.1 \text{ kJ/kg}$
$\quad\quad\quad T_5 = 600°C \quad\quad s_5 = 6.6796 \text{ kJ/kg} \cdot \text{K}$

State 6: $P_6 = 1.2 \text{ MPa}$ \} $\quad h_6 = 2860.2 \text{ kJ/kg}$
$\quad\quad\quad s_6 = s_5 \quad\quad\quad (T_6 = 218.4°C)$

State 7: $P_7 = 10 \text{ kPa}$

$$s_7 = s_5 \quad\quad x_7 = \frac{s_7 - s_f}{s_{fg}} = \frac{6.6796 - 0.6492}{7.4996} = 0.8041$$

$$h_7 = h_f + x_7 h_{fg} = 191.81 + 0.8041(2392.1) = 2115.3 \text{ kJ/kg}$$

The energy analysis of open feedwater heaters is identical to the energy analysis of mixing chambers. The feedwater heaters are generally well insulated ($\dot{Q} = 0$), and they do not involve any work interactions ($\dot{W} = 0$). By neglecting the kinetic and potential energies of the streams, the energy balance reduces for a feedwater heater to

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \quad \rightarrow \quad \sum_{\text{in}} \dot{m}h = \sum_{\text{out}} \dot{m}h$$

or

$$yh_6 + (1 - y)h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($=\dot{m}_6/\dot{m}_5$). Solving for y and substituting the enthalpy values, we find

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{798.33 - 193.01}{2860.2 - 193.01} = \textbf{\textcolor{magenta}{0.2270}}$$

Thus,

$$q_{\text{in}} = h_5 - h_4 = (3583.1 - 814.03) \text{ kJ/kg} = 2769.1 \text{ kJ/kg}$$

$$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.2270)(2115.3 - 191.81) \text{ kJ/kg}$$

$$= 1486.9 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1486.9 \text{ kJ/kg}}{2769.1 \text{ kJ/kg}} = \textbf{\textcolor{magenta}{0.463 or 46.3\%}}$$

Discussion This problem was worked out in Example 10–3c for the same pressure and temperature limits but without the regeneration process. A comparison of the two results reveals that the thermal efficiency of the cycle has increased from 43.0 to 46.3 percent as a result of regeneration. The net work output decreased by 171 kJ/kg, but the heat input decreased by 607 kJ/kg, which results in a net increase in the thermal efficiency.

EXAMPLE 10–6 The Ideal Reheat–Regenerative Rankine Cycle

Consider a steam power plant that operates on an ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam is extracted from the turbine at 4 MPa for the closed feedwater heater, and the remaining steam is reheated at the same pressure to 600°C. The extracted steam is completely condensed in the heater and is pumped to 15 MPa before it mixes with the feedwater at the same pressure. Steam for the open feedwater heater is extracted from the low-pressure turbine at a pressure of 0.5 MPa. Determine the fractions of steam extracted from the turbine as well as the thermal efficiency of the cycle.

Solution A steam power plant operates on the ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. The fractions of steam extracted from the turbine and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** In both open and closed feedwater heaters, feedwater is heated to the saturation temperature at the feedwater heater pressure. (Note that this is a conservative assumption since extracted steam enters the closed feedwater heater at 376°C and the saturation temperature at the closed feedwater pressure of 4 MPa is 250°C).

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–19. The power plant operates on the ideal reheat–regenerative Rankine cycle and thus the pumps and the turbines are isentropic; there are no pressure drops in the boiler, reheater, condenser, and feedwater heaters; and steam leaves the condenser and the feedwater heaters as saturated liquid.

The enthalpies at the various states and the pump work per unit mass of fluid flowing through them are

$$h_1 = 191.81 \text{ kJ/kg} \qquad\qquad h_9 = 3155.0 \text{ kJ/kg}$$

$$h_2 = 192.30 \text{ kJ/kg} \qquad\qquad h_{10} = 3155.0 \text{ kJ/kg}$$

$$h_3 = 640.09 \text{ kJ/kg} \qquad\qquad h_{11} = 3674.9 \text{ kJ/kg}$$

$$h_4 = 643.92 \text{ kJ/kg} \qquad\qquad h_{12} = 3014.8 \text{ kJ/kg}$$

$$h_5 = 1087.4 \text{ kJ/kg} \qquad\qquad h_{13} = 2335.7 \text{ kJ/kg}$$

$$h_6 = 1087.4 \text{ kJ/kg} \qquad\qquad w_{\text{pump I,in}} = 0.49 \text{ kJ/kg}$$

$$h_7 = 1101.2 \text{ kJ/kg} \qquad\qquad w_{\text{pump II,in}} = 3.83 \text{ kJ/kg}$$

$$h_8 = 1089.8 \text{ kJ/kg} \qquad\qquad w_{\text{pump III,in}} = 13.77 \text{ kJ/kg}$$

The fractions of steam extracted are determined from the mass and energy balances of the feedwater heaters:

Closed feedwater heater:

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

$$yh_{10} + (1 - y)h_4 = (1 - y)h_5 + yh_6$$

$$y = \frac{h_5 - h_4}{(h_{10} - h_6) + (h_5 - h_4)} = \frac{1087.4 - 643.92}{(3155.0 - 1087.4) + (1087.4 - 643.92)} = \mathbf{0.1766}$$

Open feedwater heater:

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

$$zh_{12} + (1 - y - z)h_2 = (1 - y)h_3$$

$$z = \frac{(1 - y)(h_3 - h_2)}{h_{12} - h_2} = \frac{(1 - 0.1766)(640.09 - 192.30)}{3014.8 - 192.30} = \mathbf{0.1306}$$

The enthalpy at state 8 is determined by applying the mass and energy equations to the mixing chamber, which is assumed to be insulated:

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

$$(1)h_8 = (1 - y)h_5 + yh_7$$

$$h_8 = (1 - 0.1766)(1087.4) \text{ kJ/kg} + 0.1766(1101.2) \text{ kJ/kg}$$
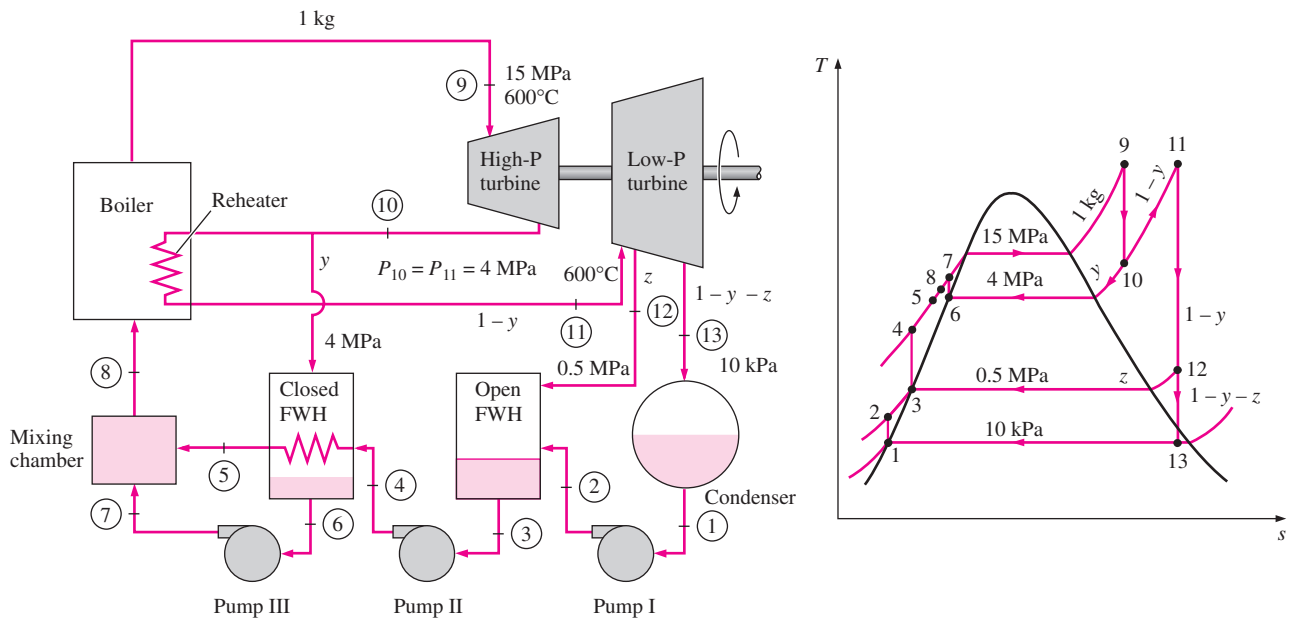
$$= 1089.8 \text{ kJ/kg}$$

FIGURE 10–19

Schematic and *T-s* diagram for Example 10–6.

Thus,

$$q_{in} = (h_9 - h_8) + (1 - y)(h_{11} - h_{10})$$

$$= (3583.1 - 1089.8) \text{ kJ/kg} + (1 - 0.1766)(3674.9 - 3155.0) \text{ kJ/kg}$$

$$= 2921.4 \text{ kJ/kg}$$

$$q_{out} = (1 - y - z)(h_{13} - h_1)$$

$$= (1 - 0.1766 - 0.1306)(2335.7 - 191.81) \text{ kJ/kg}$$

$$= 1485.3 \text{ kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1485.3 \text{ kJ/kg}}{2921.4 \text{ kJ/kg}} = \textbf{0.492 or 49.2\%}$$

Discussion   This problem was worked out in Example 10–4 for the same pressure and temperature limits with reheat but without the regeneration process. A comparison of the two results reveals that the thermal efficiency of the cycle has increased from 45.0 to 49.2 percent as a result of regeneration.

The thermal efficiency of this cycle could also be determined from

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{w_{turb,out} - w_{pump,in}}{q_{in}}$$

where

$$w_{turb,out} = (h_9 - h_{10}) + (1 - y)(h_{11} - h_{12}) + (1 - y - z)(h_{12} - h_{13})$$

$$w_{pump,in} = (1 - y - z)w_{pump\ I,in} + (1 - y)w_{pump\ II,in} + (y)w_{pump\ III,in}$$

Also, if we assume that the feedwater leaves the closed FWH as a saturated liquid at 15 MPa (and thus at $T_5 = 342°C$ and $h_5 = 1610.3$ kJ/kg), it can be shown that the thermal efficiency would be 50.6.

10–7 ▪ SECOND-LAW ANALYSIS OF VAPOR POWER CYCLES

The ideal Carnot cycle is a *totally reversible cycle,* and thus it does not involve any irreversibilities. The ideal Rankine cycles (simple, reheat, or regenerative), however, are only *internally reversible,* and they may involve irreversibilities external to the system, such as heat transfer through a finite temperature difference. A second-law analysis of these cycles reveals where the largest irreversibilities occur and what their magnitudes are.

Relations for exergy and exergy destruction for steady-flow systems are developed in Chap. 8. The exergy destruction for a steady-flow system can be expressed, in the rate form, as

$$\dot{X}_{dest} = T_0\dot{S}_{gen} = T_0(\dot{S}_{out} - \dot{S}_{in}) = T_0\left(\sum_{out} \dot{m}s + \frac{\dot{Q}_{out}}{T_{b,out}} - \sum_{in} \dot{m}s - \frac{\dot{Q}_{in}}{T_{b,in}}\right) \quad \text{(kW)}$$

(10–18)

or on a unit mass basis for a one-inlet, one-exit, steady-flow device as

$$x_{dest} = T_0 s_{gen} = T_0\left(s_e - s_i + \frac{q_{out}}{T_{b,out}} - \frac{q_{in}}{T_{b,in}}\right) \quad \text{(kJ/kg)} \quad \textbf{(10–19)}$$

where $T_{b,\text{in}}$ and $T_{b,\text{out}}$ are the temperatures of the system boundary where heat is transferred into and out of the system, respectively.

The exergy destruction associated with a *cycle* depends on the magnitude of the heat transfer with the high- and low-temperature reservoirs involved, and their temperatures. It can be expressed on a unit mass basis as

$$x_{\text{dest}} = T_0 \left(\sum \frac{q_{\text{out}}}{T_{b,\text{out}}} - \sum \frac{q_{\text{in}}}{T_{b,\text{in}}} \right) \qquad \text{(kJ/kg)} \qquad \textbf{(10–20)}$$

For a cycle that involves heat transfer only with a source at T_H and a sink at T_L, the exergy destruction becomes

$$x_{\text{dest}} = T_0 \left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H} \right) \qquad \text{(kJ/kg)} \qquad \textbf{(10–21)}$$

The exergy of a fluid stream ψ at any state can be determined from

$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \qquad \text{(kJ/kg)} \qquad \textbf{(10–22)}$$

where the subscript "0" denotes the state of the surroundings.

EXAMPLE 10–7   Second-Law Analysis of an Ideal Rankine Cycle

Determine the exergy destruction associated with the Rankine cycle (all four processes as well as the cycle) discussed in Example 10–1, assuming that heat is transferred to the steam in a furnace at 1600 K and heat is rejected to a cooling medium at 290 K and 100 kPa. Also, determine the exergy of the steam leaving the turbine.

Solution   The Rankine cycle analyzed in Example 10–1 is reconsidered. For specified source and sink temperatures, the exergy destruction associated with the cycle and exergy of the steam at turbine exit are to be determined.
Analysis   In Example 10–1, the heat input was determined to be 2728.6 kJ/kg, and the heat rejected to be 2018.6 kJ/kg.

Processes 1-2 and 3-4 are isentropic ($s_1 = s_2$, $s_3 = s_4$) and therefore do not involve any internal or external irreversibilities, that is,

$$x_{\text{dest,12}} = 0 \quad \text{and} \quad x_{\text{dest,34}} = 0$$

Processes 2-3 and 4-1 are constant-pressure heat-addition and heat-rejection processes, respectively, and they are internally reversible. But the heat transfer between the working fluid and the source or the sink takes place through a finite temperature difference, rendering both processes irreversible. The irreversibility associated with each process is determined from Eq. 10–19. The entropy of the steam at each state is determined from the steam tables:

$$s_2 = s_1 = s_{f \ @ \ 75 \text{ kPa}} = 1.2132 \text{ kJ/kg} \cdot \text{K}$$

$$s_4 = s_3 = 6.7450 \text{ kJ/kg} \cdot \text{K} \qquad \text{(at 3 MPa, 350°C)}$$

Thus,

$$x_{\text{dest,23}} = T_0 \left(s_3 - s_2 - \frac{q_{\text{in,23}}}{T_{\text{source}}} \right)$$

$$= (290 \text{ K}) \left[(6.7450 - 1.2132) \text{ kJ/kg} \cdot \text{K} - \frac{2728.6 \text{ kJ/kg}}{1600 \text{ K}} \right]$$

$$= \textbf{1110 kJ/kg}$$

$$x_{\text{dest,41}} = T_0 \left(s_1 - s_4 + \frac{q_{\text{out,41}}}{T_{\text{sink}}} \right)$$

$$= (290 \text{ K}) \left[(1.2132 - 6.7450) \text{ kJ/kg} \cdot \text{K} + \frac{2018.6 \text{ kJ/kg}}{290 \text{ K}} \right]$$

$$= \mathbf{414 \text{ kJ/kg}}$$

Therefore, the irreversibility of the cycle is

$$x_{\text{dest,cycle}} = x_{\text{dest,12}} + x_{\text{dest,23}} + x_{\text{dest,34}} + x_{\text{dest,41}}$$

$$= 0 + 1110 \text{ kJ/kg} + 0 + 414 \text{ kJ/kg}$$

$$= \mathbf{1524 \text{ kJ/kg}}$$

The total exergy destroyed during the cycle could also be determined from Eq. 10–21. Notice that the largest exergy destruction in the cycle occurs during the heat-addition process. Therefore, any attempt to reduce the exergy destruction should start with this process. Raising the turbine inlet temperature of the steam, for example, would reduce the temperature difference and thus the exergy destruction.

The exergy (work potential) of the steam leaving the turbine is determined from Eq. 10–22. Disregarding the kinetic and potential energies, it reduces to

$$\psi_4 = (h_4 - h_0) - T_0(s_4 - s_0) + \frac{V_4^2}{2}\nearrow^0 + gz_4 \nearrow^0$$

$$= (h_4 - h_0) - T_0(s_4 - s_0)$$

where

$$h_0 = h_{@ 290 \text{ K,100 kPa}} \cong h_{f @ 290 \text{ K}} = 71.355 \text{ kJ/kg}$$

$$s_0 = s_{@ 290 \text{ K,100 kPa}} \cong s_{f @ 290 \text{ K}} = 0.2533 \text{ kJ/kg} \cdot \text{K}$$

Thus,

$$\psi_4 = (2403.0 - 71.355) \text{ kJ/kg} - (290 \text{ K}) [(6.7450 - 0.2533) \text{ kJ/kg} \cdot \text{K}]$$

$$= \mathbf{449 \text{ kJ/kg}}$$

Discussion Note that 449 kJ/kg of work could be obtained from the steam leaving the turbine if it is brought to the state of the surroundings in a reversible manner.

10–8 · COGENERATION

In all the cycles discussed so far, the sole purpose was to convert a portion of the heat transferred to the working fluid to work, which is the most valuable form of energy. The remaining portion of the heat is rejected to rivers, lakes, oceans, or the atmosphere as waste heat, because its quality (or grade) is too low to be of any practical use. Wasting a large amount of heat is a price we have to pay to produce work, because electrical or mechanical work is the only form of energy on which many engineering devices (such as a fan) can operate.

Many systems or devices, however, require energy input in the form of heat, called *process heat.* Some industries that rely heavily on process heat are chemical, pulp and paper, oil production and refining, steel making,

food processing, and textile industries. Process heat in these industries is usually supplied by steam at 5 to 7 atm and 150 to 200°C (300 to 400°F). Energy is usually transferred to the steam by burning coal, oil, natural gas, or another fuel in a furnace.

Now let us examine the operation of a process-heating plant closely. Disregarding any heat losses in the piping, all the heat transferred to the steam in the boiler is used in the process-heating units, as shown in Fig. 10–20. Therefore, process heating seems like a perfect operation with practically no waste of energy. From the second-law point of view, however, things do not look so perfect. The temperature in furnaces is typically very high (around 1400°C), and thus the energy in the furnace is of very high quality. This high-quality energy is transferred to water to produce steam at about 200°C or below (a highly irreversible process). Associated with this irreversibility is, of course, a loss in exergy or work potential. It is simply not wise to use high-quality energy to accomplish a task that could be accomplished with low-quality energy.

Industries that use large amounts of process heat also consume a large amount of electric power. Therefore, it makes economical as well as engineering sense to use the already-existing work potential to produce power instead of letting it go to waste. The result is a plant that produces electricity while meeting the process-heat requirements of certain industrial processes. Such a plant is called a *cogeneration plant*. In general, **cogeneration** is *the production of more than one useful form of energy (such as process heat and electric power) from the same energy source.*

Either a steam-turbine (Rankine) cycle or a gas-turbine (Brayton) cycle or even a combined cycle (discussed later) can be used as the power cycle in a cogeneration plant. The schematic of an ideal steam-turbine cogeneration plant is shown in Fig. 10–21. Let us say this plant is to supply process heat $\dot{Q}_p$ at 500 kPa at a rate of 100 kW. To meet this demand, steam is expanded in the turbine to a pressure of 500 kPa, producing power at a rate of, say, 20 kW. The flow rate of the steam can be adjusted such that steam leaves the process-heating section as a saturated liquid at 500 kPa. Steam is then pumped to the boiler pressure and is heated in the boiler to state 3. The pump work is usually very small and can be neglected. Disregarding any heat losses, the rate of heat input in the boiler is determined from an energy balance to be 120 kW.

Probably the most striking feature of the ideal steam-turbine cogeneration plant shown in Fig. 10–21 is the absence of a condenser. Thus no heat is rejected from this plant as waste heat. In other words, all the energy transferred to the steam in the boiler is utilized as either process heat or electric power. Thus it is appropriate to define a **utilization factor** ϵ_u for a cogeneration plant as

$$\epsilon_u = \frac{\text{Net work output } + \text{ Process heat delivered}}{\text{Total heat input}} = \frac{\dot{W}_{\text{net}} + \dot{Q}_p}{\dot{Q}_{\text{in}}} \qquad (10\text{–}23)$$

or

$$\epsilon_u = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} \qquad (10\text{–}24)$$

where $\dot{Q}_{\text{out}}$ represents the heat rejected in the condenser. Strictly speaking, $\dot{Q}_{\text{out}}$ also includes all the undesirable heat losses from the piping and other components, but they are usually small and thus neglected. It also includes combustion inefficiencies such as incomplete combustion and stack losses
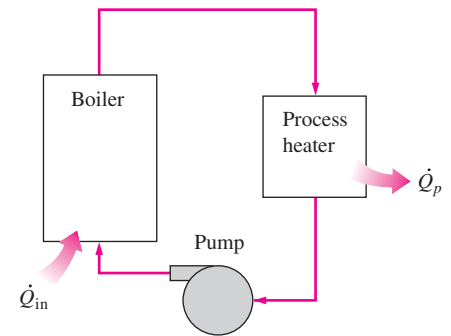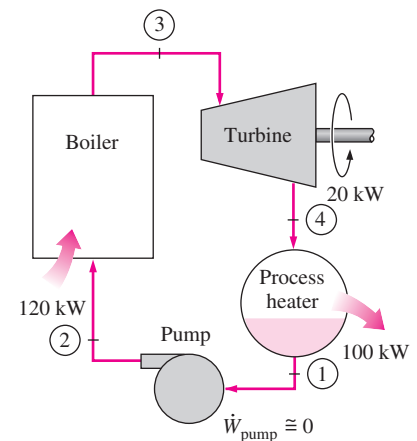

FIGURE 10–20

A simple process-heating plant.

FIGURE 10–21
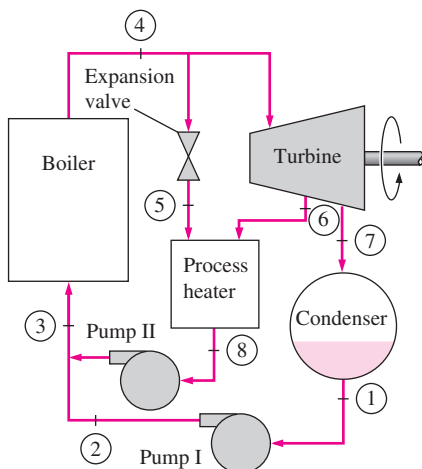
An ideal cogeneration plant.

FIGURE 10–22

A cogeneration plant with adjustable loads.

when the utilization factor is defined on the basis of the heating value of the fuel. The utilization factor of the ideal steam-turbine cogeneration plant is obviously 100 percent. Actual cogeneration plants have utilization factors as high as 80 percent. Some recent cogeneration plants have even higher utilization factors.

Notice that without the turbine, we would need to supply heat to the steam in the boiler at a rate of only 100 kW instead of at 120 kW. The additional 20 kW of heat supplied is converted to work. Therefore, a cogeneration power plant is equivalent to a process-heating plant combined with a power plant that has a thermal efficiency of 100 percent.

The ideal steam-turbine cogeneration plant described above is not practical because it cannot adjust to the variations in power and process-heat loads. The schematic of a more practical (but more complex) cogeneration plant is shown in Fig. 10–22. Under normal operation, some steam is extracted from the turbine at some predetermined intermediate pressure P_6. The rest of the steam expands to the condenser pressure P_7 and is then cooled at constant pressure. The heat rejected from the condenser represents the waste heat for the cycle.

At times of high demand for process heat, all the steam is routed to the process-heating units and none to the condenser ($\dot{m}_7 = 0$). The waste heat is zero in this mode. If this is not sufficient, some steam leaving the boiler is throttled by an expansion or pressure-reducing valve (PRV) to the extraction pressure P_6 and is directed to the process-heating unit. Maximum process heating is realized when all the steam leaving the boiler passes through the PRV ($\dot{m}_5 = \dot{m}_4$). No power is produced in this mode. When there is no demand for process heat, all the steam passes through the turbine and the condenser ($\dot{m}_5 = \dot{m}_6 = 0$), and the cogeneration plant operates as an ordinary steam power plant. The rates of heat input, heat rejected, and process heat supply as well as the power produced for this cogeneration plant can be expressed as follows:

$$\dot{Q}_{in} = \dot{m}_3(h_4 - h_3) \tag{10–25}$$

$$\dot{Q}_{out} = \dot{m}_7(h_7 - h_1) \tag{10–26}$$

$$\dot{Q}_p = \dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_8 h_8 \tag{10–27}$$

$$\dot{W}_{turb} = (\dot{m}_4 - \dot{m}_5)(h_4 - h_6) + \dot{m}_7(h_6 - h_7) \tag{10–28}$$

Under optimum conditions, a cogeneration plant simulates the ideal cogeneration plant discussed earlier. That is, all the steam expands in the turbine to the extraction pressure and continues to the process-heating unit. No steam passes through the PRV or the condenser; thus, no waste heat is rejected ($\dot{m}_4 = \dot{m}_6$ and $\dot{m}_5 = \dot{m}_7 = 0$). This condition may be difficult to achieve in practice because of the constant variations in the process-heat and power loads. But the plant should be designed so that the optimum operating conditions are approximated most of the time.

The use of cogeneration dates to the beginning of this century when power plants were integrated to a community to provide district heating, that is, space, hot water, and process heating for residential and commercial buildings. The district heating systems lost their popularity in the 1940s owing to low fuel prices. However, the rapid rise in fuel prices in the 1970s brought about renewed interest in district heating.

Cogeneration plants have proved to be economically very attractive. Consequently, more and more such plants have been installed in recent years, and more are being installed.

EXAMPLE 10–8   An Ideal Cogeneration Plant

Consider the cogeneration plant shown in Fig. 10–23. Steam enters the turbine at 7 MPa and 500°C. Some steam is extracted from the turbine at 500 kPa for process heating. The remaining steam continues to expand to 5 kPa. Steam is then condensed at constant pressure and pumped to the boiler pressure of 7 MPa. At times of high demand for process heat, some steam leaving the boiler is throttled to 500 kPa and is routed to the process heater. The extraction fractions are adjusted so that steam leaves the process heater as a saturated liquid at 500 kPa. It is subsequently pumped to 7 MPa. The mass flow rate of steam through the boiler is 15 kg/s. Disregarding any pressure drops and heat losses in the piping and assuming the turbine and the pump to be isentropic, determine (a) the maximum rate at which process heat can be supplied, (b) the power produced and the utilization factor when no process heat is supplied, and (c) the rate of process heat supply when 10 percent of the steam is extracted before it enters the turbine and 70 percent of the steam is extracted from the turbine at 500 kPa for process heating.

Solution   A cogeneration plant is considered. The maximum rate of process heat supply, the power produced and the utilization factor when no process heat is supplied, and the rate of process heat supply when steam is extracted from the steam line and turbine at specified ratios are to be determined.

Assumptions   **1** Steady operating conditions exist. **2** Pressure drops and heat losses in piping are negligible. **3** Kinetic and potential energy changes are negligible.

Analysis   The schematic of the cogeneration plant and the *T-s* diagram of the cycle are shown in Fig. 10–23. The power plant operates on an ideal
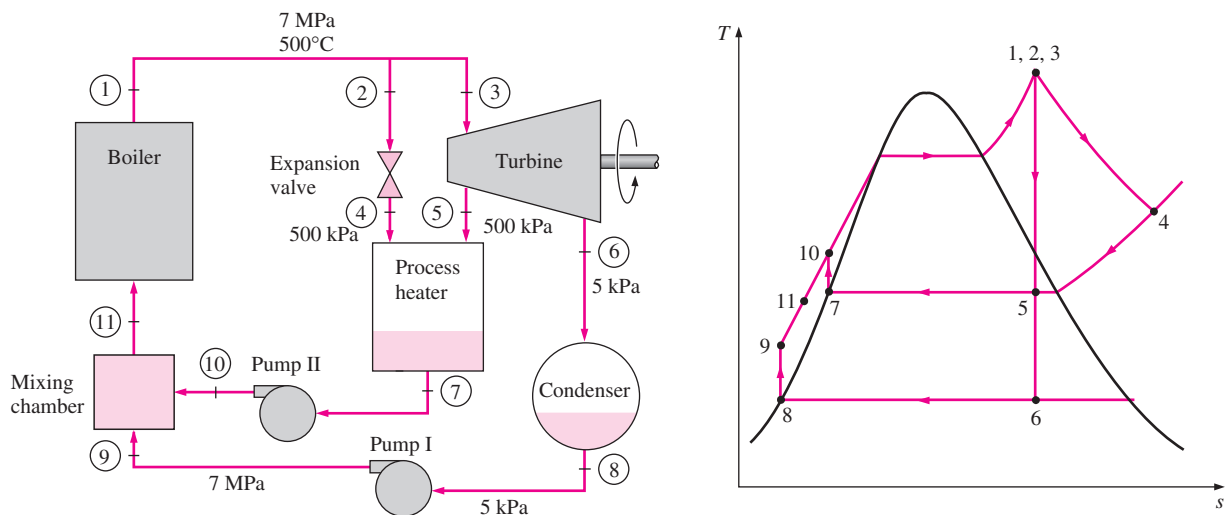

FIGURE 10–23

Schematic and *T-s* diagram for Example 10–8.

cycle and thus the pumps and the turbines are isentropic; there are no pressure drops in the boiler, process heater, and condenser; and steam leaves the condenser and the process heater as saturated liquid.

The work inputs to the pumps and the enthalpies at various states are as follows:

$$w_{\text{pump I,in}} = v_8(P_9 - P_8) = (0.001005 \text{ m}^3/\text{kg})[(7000 - 5)\text{kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 7.03 \text{ kJ/kg}$$

$$w_{\text{pump II,in}} = v_7(P_{10} - P_7) = (0.001093 \text{ m}^3/\text{kg})[(7000 - 500) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 7.10 \text{ kJ/kg}$$

$$h_1 = h_2 = h_3 = h_4 = 3411.4 \text{ kJ/kg}$$

$$h_5 = 2739.3 \text{ kJ/kg}$$

$$h_6 = 2073.0 \text{ kJ/kg}$$

$$h_7 = h_{f @ 500 \text{ kPa}} = 640.09 \text{ kJ/kg}$$

$$h_8 = h_{f @ 5 \text{ kPa}} = 137.75 \text{ kJ/kg}$$

$$h_9 = h_8 + w_{\text{pump I,in}} = (137.75 + 7.03) \text{ kJ/kg} = 144.78 \text{ kJ/kg}$$

$$h_{10} = h_7 + w_{\text{pump II,in}} = (640.09 + 7.10) \text{ kJ/kg} = 647.19 \text{ kJ/kg}$$

(*a*) The maximum rate of process heat is achieved when all the steam leaving the boiler is throttled and sent to the process heater and none is sent to the turbine (that is, $\dot{m}_4 = \dot{m}_7 = \dot{m}_1 = 15$ kg/s and $\dot{m}_3 = \dot{m}_5 = \dot{m}_6 = 0$). Thus,

$$\dot{Q}_{p,\text{max}} = \dot{m}_1(h_4 - h_7) = (15 \text{ kg/s})[(3411.4 - 640.09) \text{ kJ/kg}] = \textbf{41,570 kW}$$

The utilization factor is 100 percent in this case since no heat is rejected in the condenser, heat losses from the piping and other components are assumed to be negligible, and combustion losses are not considered.

(*b*) When no process heat is supplied, all the steam leaving the boiler passes through the turbine and expands to the condenser pressure of 5 kPa (that is, $\dot{m}_3 = \dot{m}_6 = \dot{m}_1 = 15$ kg/s and $\dot{m}_2 = \dot{m}_5 = 0$). Maximum power is produced in this mode, which is determined to be

$$\dot{W}_{\text{turb,out}} = \dot{m}(h_3 - h_6) = (15 \text{ kg/s})[(3411.4 - 2073.0) \text{ kJ/kg}] = 20,076 \text{ kW}$$

$$\dot{W}_{\text{pump,in}} = (15 \text{ kg/s})(7.03 \text{ kJ/kg}) = 105 \text{ kW}$$

$$\dot{W}_{\text{net,out}} = \dot{W}_{\text{turb,out}} - \dot{W}_{\text{pump,in}} = (20,076 - 105) \text{ kW} = 19,971 \text{ kW} \cong \textbf{20.0 MW}$$

$$\dot{Q}_{\text{in}} = \dot{m}_1(h_1 - h_{11}) = (15 \text{ kg/s})[(3411.4 - 144.78) \text{ kJ/kg}] = 48,999 \text{ kW}$$

Thus,

$$\epsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_p}{\dot{Q}_{\text{in}}} = \frac{(19,971 + 0) \text{ kW}}{48,999 \text{ kW}} = \textbf{0.408 or 40.8\%}$$

That is, 40.8 percent of the energy is utilized for a useful purpose. Notice that the utilization factor is equivalent to the thermal efficiency in this case.

(*c*) Neglecting any kinetic and potential energy changes, an energy balance on the process heater yields

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

$$\dot{m}_4 h_4 + \dot{m}_5 h_5 = \dot{Q}_{p,out} + \dot{m}_7 h_7$$

or

$$\dot{Q}_{p,out} = \dot{m}_4 h_4 + \dot{m}_5 h_5 - \dot{m}_7 h_7$$

where

$$\dot{m}_4 = (0.1)(15 \text{ kg/s}) = 1.5 \text{ kg/s}$$

$$\dot{m}_5 = (0.7)(15 \text{ kg/s}) = 10.5 \text{ kg/s}$$

$$\dot{m}_7 = \dot{m}_4 + \dot{m}_5 = 1.5 + 10.5 = 12 \text{ kg/s}$$

Thus

$$\dot{Q}_{p,out} = (1.5 \text{ kg/s})(3411.4 \text{ kJ/kg}) + (10.5 \text{ kg/s})(2739.3 \text{ kJ/kg})$$

$$- (12 \text{ kg/s})(640.09 \text{ kJ/kg})$$

$$= \textbf{26.2 MW}$$

Discussion Note that 26.2 MW of the heat transferred will be utilized in the process heater. We could also show that 11.0 MW of power is produced in this case, and the rate of heat input in the boiler is 43.0 MW. Thus the utilization factor is 86.5 percent.

10–9 ▪ COMBINED GAS–VAPOR POWER CYCLES

The continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants. The *binary vapor cycle* discussed later is one such modification. A more popular modification involves a gas power cycle topping a vapor power cycle, which is called the **combined gas–vapor cycle,** or just the **combined cycle.** The combined cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam-turbine (Rankine) cycle, which has a higher thermal efficiency than either of the cycles executed individually.

Gas-turbine cycles typically operate at considerably higher temperatures than steam cycles. The maximum fluid temperature at the turbine inlet is about 620°C (1150°F) for modern steam power plants, but over 1425°C (2600°F) for gas-turbine power plants. It is over 1500°C at the burner exit of turbojet engines. The use of higher temperatures in gas turbines is made possible by recent developments in cooling the turbine blades and coating the blades with high-temperature-resistant materials such as ceramics. Because of the higher average temperature at which heat is supplied, gas-turbine cycles have a greater potential for higher thermal efficiencies. However, the gas-turbine cycles have one inherent disadvantage: The gas leaves the gas turbine at very high temperatures (usually above 500°C), which erases any potential gains in the thermal efficiency. The situation can be improved somewhat by using regeneration, but the improvement is limited.

It makes engineering sense to take advantage of the very desirable characteristics of the gas-turbine cycle at high temperatures *and* to use the high-temperature exhaust gases as the energy source for the bottoming cycle such as a steam power cycle. The result is a combined gas–steam cycle, as shown

in Fig. 10–24. In this cycle, energy is recovered from the exhaust gases by transferring it to the steam in a heat exchanger that serves as the boiler. In general, more than one gas turbine is needed to supply sufficient heat to the steam. Also, the steam cycle may involve regeneration as well as reheating. Energy for the reheating process can be supplied by burning some additional fuel in the oxygen-rich exhaust gases.

Recent developments in gas-turbine technology have made the combined gas–steam cycle economically very attractive. The combined cycle increases the efficiency without increasing the initial cost greatly. Consequently, many new power plants operate on combined cycles, and many more existing steam- or gas-turbine plants are being converted to combined-cycle power plants. Thermal efficiencies well over 40 percent are reported as a result of conversion.

A 1090-MW Tohoku combined plant that was put in commercial operation in 1985 in Niigata, Japan, is reported to operate at a thermal efficiency of 44 percent. This plant has two 191-MW steam turbines and six 118-MW gas turbines. Hot combustion gases enter the gas turbines at 1154°C, and steam enters the steam turbines at 500°C. Steam is cooled in the condenser by cooling water at an average temperature of 15°C. The compressors have a pressure ratio of 14, and the mass flow rate of air through the compressors is 443 kg/s.
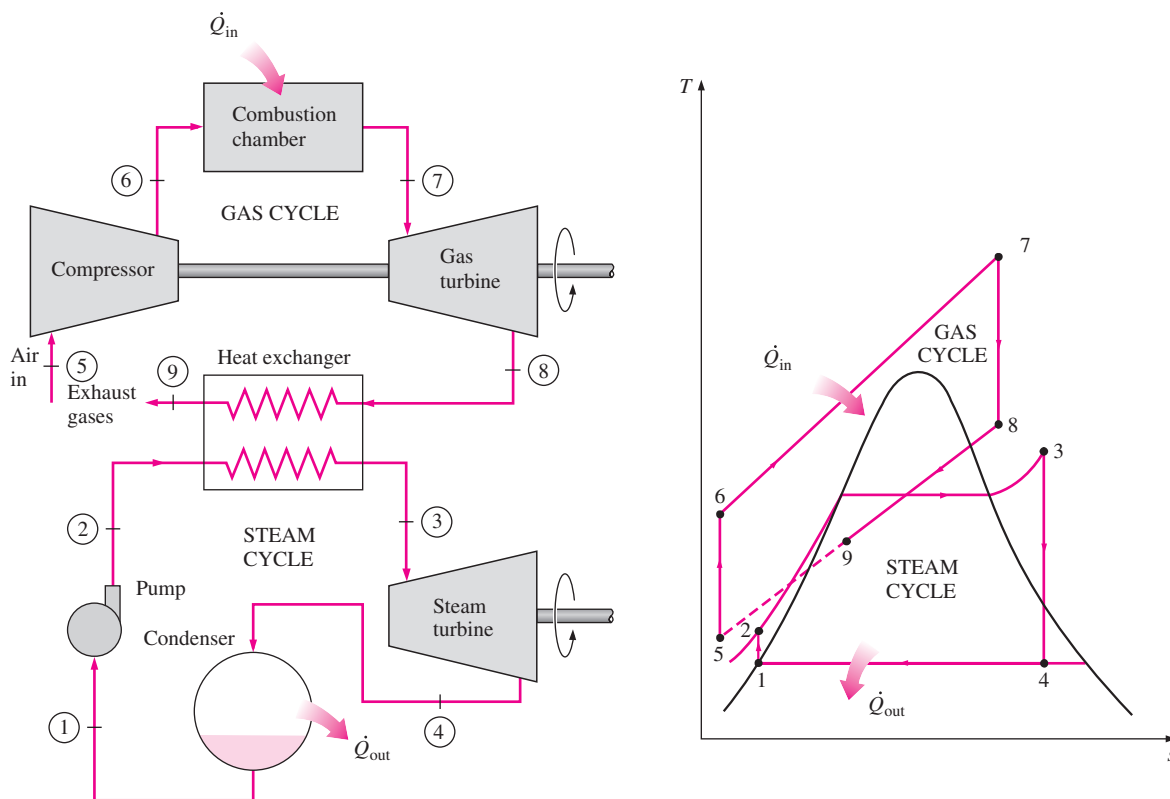

FIGURE 10–24
Combined gas–steam power plant.

A 1350-MW combined-cycle power plant built in Ambarli, Turkey, in 1988 by Siemens of Germany is the first commercially operating thermal plant in the world to attain an efficiency level as high as 52.5 percent at design operating conditions. This plant has six 150-MW gas turbines and three 173-MW steam turbines. Some recent combined-cycle power plants have achieved efficiencies above 60 percent.

EXAMPLE 10–9   A Combined Gas–Steam Power Cycle

Consider the combined gas–steam power cycle shown in Fig. 10–25. The topping cycle is a gas-turbine cycle that has a pressure ratio of 8. Air enters the compressor at 300 K and the turbine at 1300 K. The isentropic efficiency of the compressor is 80 percent, and that of the gas turbine is 85 percent. The bottoming cycle is a simple ideal Rankine cycle operating between the pressure limits of 7 MPa and 5 kPa. Steam is heated in a heat exchanger by the exhaust gases to a temperature of 500°C. The exhaust gases leave the heat exchanger at 450 K. Determine (*a*) the ratio of the mass flow rates of the steam and the combustion gases and (*b*) the thermal efficiency of the combined cycle.

Solution   A combined gas–steam cycle is considered. The ratio of the mass flow rates of the steam and the combustion gases and the thermal efficiency are to be determined.

Analysis   The *T-s* diagrams of both cycles are given in Fig. 10–25. The gas-turbine cycle alone was analyzed in Example 9–6, and the steam cycle in Example 10–8*b*, with the following results:

Gas cycle: $h_4' = 880.36 \text{ kJ/kg}$ $(T_4' = 853 \text{ K})$

$\qquad\qquad q_{in} = 790.58 \text{ kJ/kg} \quad w_{net} = 210.41 \text{ kJ/kg} \quad \eta_{th} = 26.6\%$
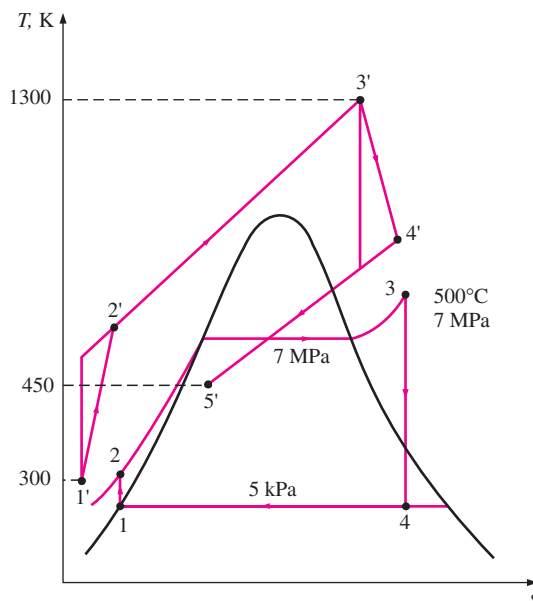
$\qquad\qquad h_5' = h_{@\,450\,K} = 451.80 \text{ kJ/kg}$

FIGURE 10–25

T-s diagram of the gas–steam combined cycle described in Example 10–9.

Steam cycle: $\quad h_2 = 144.78 \text{ kJ/kg} \qquad (T_2 = 33°C)$

$\qquad\qquad\qquad h_3 = 3411.4 \text{ kJ/kg} \qquad (T_3 = 500°C)$

$\qquad\qquad\qquad w_{\text{net}} = 1331.4 \text{ kJ/kg} \qquad \eta_{\text{th}} = 40.8\%$

(*a*) The ratio of mass flow rates is determined from an energy balance on the heat exchanger:

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

$$\dot{m}_g h_5' + \dot{m}_s h_3 = \dot{m}_g h_4' + \dot{m}_s h_2$$

$$\dot{m}_s(h_3 - h_2) = \dot{m}_g(h_4' - h_5')$$

$$\dot{m}_s(3411.4 - 144.78) = \dot{m}_g(880.36 - 451.80)$$

Thus,

$$\frac{\dot{m}_s}{\dot{m}_g} = y = \mathbf{0.131}$$

That is, 1 kg of exhaust gases can heat only 0.131 kg of steam from 33 to 500°C as they are cooled from 853 to 450 K. Then the total net work output per kilogram of combustion gases becomes

$w_{\text{net}} = w_{\text{net,gas}} + y w_{\text{net,steam}}$

$\qquad = (210.41 \text{ kJ/kg gas}) + (0.131 \text{ kg steam/kg gas})(1331.4 \text{ kJ/kg steam})$

$\qquad = 384.8 \text{ kJ/kg gas}$

Therefore, for each kg of combustion gases produced, the combined plant will deliver 384.8 kJ of work. The net power output of the plant is determined by multiplying this value by the mass flow rate of the working fluid in the gas-turbine cycle.

(*b*) The thermal efficiency of the combined cycle is determined from

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{384.8 \text{ kJ/kg gas}}{790.6 \text{ kJ/kg gas}} = \mathbf{0.487 \text{ or } 48.7\%}$$

Discussion   Note that this combined cycle converts to useful work 48.7 percent of the energy supplied to the gas in the combustion chamber. This value is considerably higher than the thermal efficiency of the gas-turbine cycle (26.6 percent) or the steam-turbine cycle (40.8 percent) operating alone.

TOPIC OF SPECIAL INTEREST*   Binary Vapor Cycles

With the exception of a few specialized applications, the working fluid predominantly used in vapor power cycles is water. Water is the *best* working fluid presently available, but it is far from being the *ideal* one. The binary cycle is an attempt to overcome some of the shortcomings of water and to approach the *ideal* working fluid by using two fluids. Before we discuss the binary cycle, let us list the characteristics of a working fluid most suitable for vapor power cycles:

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*This section can be skipped without a loss in continuity.

1. A high critical temperature and a safe maximum pressure. A critical temperature above the metallurgically allowed maximum temperature (about 620°C) makes it possible to transfer a considerable portion of the heat isothermally at the maximum temperature as the fluid changes phase. This makes the cycle approach the Carnot cycle. Very high pressures at the maximum temperature are undesirable because they create material-strength problems.

2. Low triple-point temperature. A triple-point temperature below the temperature of the cooling medium prevents any solidification problems.

3. A condenser pressure that is not too low. Condensers usually operate below atmospheric pressure. Pressures well below the atmospheric pressure create air-leakage problems. Therefore, a substance whose saturation pressure at the ambient temperature is too low is not a good candidate.

4. A high enthalpy of vaporization (h_{fg}) so that heat transfer to the working fluid is nearly isothermal and large mass flow rates are not needed.

5. A saturation dome that resembles an inverted U. This eliminates the formation of excessive moisture in the turbine and the need for reheating.

6. Good heat transfer characteristics (high thermal conductivity).

7. Other properties such as being inert, inexpensive, readily available, and nontoxic.

Not surprisingly, no fluid possesses all these characteristics. Water comes the closest, although it does not fare well with respect to characteristics 1, 3, and 5. We can cope with its subatmospheric condenser pressure by careful sealing, and with the inverted V-shaped saturation dome by reheating, but there is not much we can do about item 1. Water has a low critical temperature (374°C, well below the metallurgical limit) and very high saturation pressures at high temperatures (16.5 MPa at 350°C).

Well, we cannot change the way water behaves during the high-temperature part of the cycle, but we certainly can replace it with a more suitable fluid. The result is a power cycle that is actually a combination of two cycles, one in the high-temperature region and the other in the low-temperature region. Such a cycle is called a **binary vapor cycle.** In binary vapor cycles, the condenser of the high-temperature cycle (also called the *topping cycle*) serves as the boiler of the low-temperature cycle (also called the *bottoming cycle*). That is, the heat output of the high-temperature cycle is used as the heat input to the low-temperature one.

Some working fluids found suitable for the high-temperature cycle are mercury, sodium, potassium, and sodium–potassium mixtures. The schematic and *T-s* diagram for a mercury–water binary vapor cycle are shown in Fig. 10–26. The critical temperature of mercury is 898°C (well above the current metallurgical limit), and its critical pressure is only about 18 MPa. This makes mercury a very suitable working fluid for the topping cycle. Mercury is not suitable as the sole working fluid for the entire cycle, however, since at a condenser temperature of 32°C its saturation pressure is 0.07 Pa. A power plant cannot operate at this vacuum because of air-leakage problems. At an acceptable condenser pressure of 7 kPa, the saturation temperature of mercury is
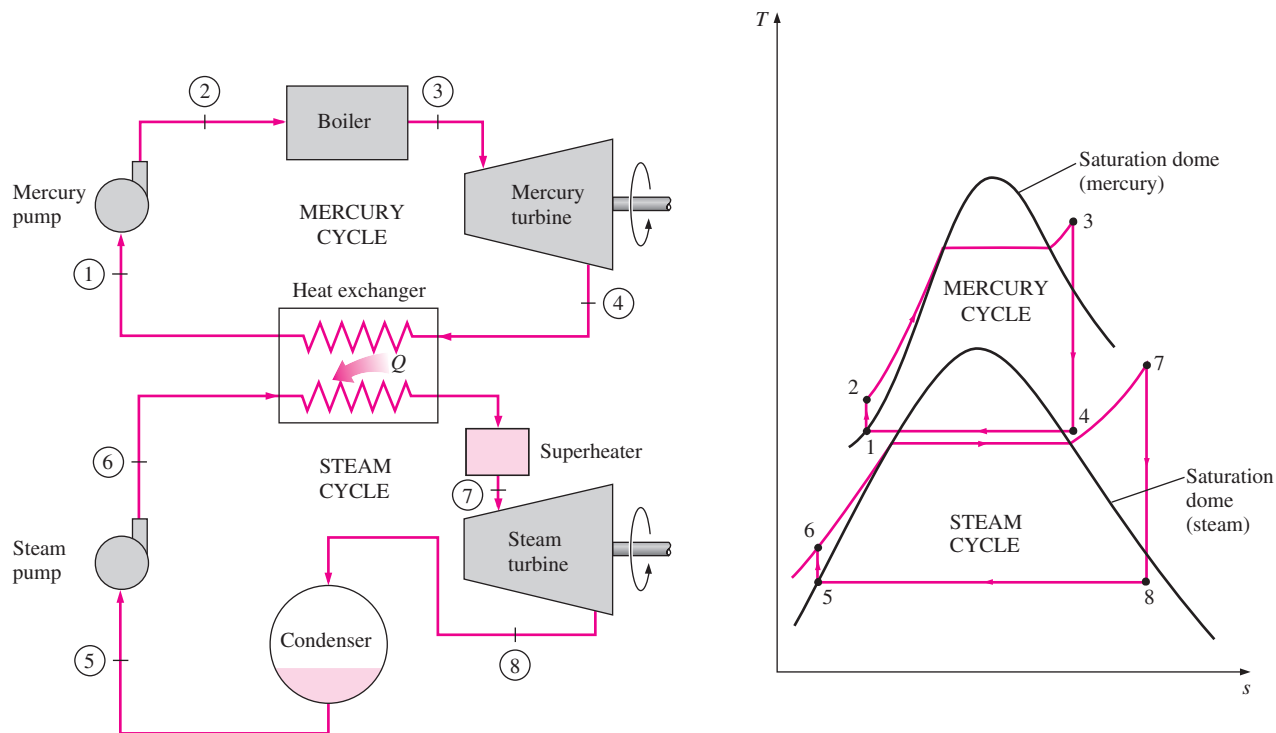
FIGURE 10–26

Mercury–water binary vapor cycle.

237°C, which is too high as the minimum temperature in the cycle. Therefore, the use of mercury as a working fluid is limited to the high-temperature cycles. Other disadvantages of mercury are its toxicity and high cost. The mass flow rate of mercury in binary vapor cycles is several times that of water because of its low enthalpy of vaporization.

It is evident from the *T-s* diagram in Fig. 10–26 that the binary vapor cycle approximates the Carnot cycle more closely than the steam cycle for the same temperature limits. Therefore, the thermal efficiency of a power plant can be increased by switching to binary cycles. The use of mercury–water binary cycles in the United States dates back to 1928. Several such plants have been built since then in the New England area, where fuel costs are typically higher. A small (40-MW) mercury–steam power plant that was in service in New Hampshire in 1950 had a higher thermal efficiency than most of the large modern power plants in use at that time.

Studies show that thermal efficiencies of 50 percent or higher are possible with binary vapor cycles. However, binary vapor cycles are not economically attractive because of their high initial cost and the competition offered by the combined gas–steam power plants.

SUMMARY

The *Carnot cycle* is not a suitable model for vapor power cycles because it cannot be approximated in practice. The model cycle for vapor power cycles is the *Rankine cycle,* which is composed of four internally reversible processes: constant-pressure heat addition in a boiler, isentropic expansion in a turbine, constant-pressure heat rejection in a condenser, and isentropic compression in a pump. Steam leaves the condenser as a saturated liquid at the condenser pressure.

The thermal efficiency of the Rankine cycle can be increased by increasing the average temperature at which heat is transferred to the working fluid and/or by decreasing the average temperature at which heat is rejected to the cooling medium. The average temperature during heat rejection can be decreased by lowering the turbine exit pressure. Consequently, the condenser pressure of most vapor power plants is well below the atmospheric pressure. The average temperature during heat addition can be increased by raising the boiler pressure or by superheating the fluid to high temperatures. There is a limit to the degree of superheating, however, since the fluid temperature is not allowed to exceed a metallurgically safe value.

Superheating has the added advantage of decreasing the moisture content of the steam at the turbine exit. Lowering the exhaust pressure or raising the boiler pressure, however, increases the moisture content. To take advantage of the improved efficiencies at higher boiler pressures and lower condenser pressures, steam is usually *reheated* after expanding partially in the high-pressure turbine. This is done by extracting the steam after partial expansion in the high-pressure turbine, sending it back to the boiler where it is reheated at constant pressure, and returning it to the low-pressure turbine for complete expansion to the condenser pressure. The average temperature during the reheat process, and thus the thermal efficiency of the cycle, can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach an isothermal process at maximum temperature. Reheating also decreases the moisture content at the turbine exit.

Another way of increasing the thermal efficiency of the Rankine cycle is *regeneration.* During a regeneration process, liquid water (feedwater) leaving the pump is heated by steam bled off the turbine at some intermediate pressure in devices called *feedwater heaters.* The two streams are mixed in open feedwater heaters, and the mixture leaves as a saturated liquid at the heater pressure. In closed feedwater heaters, heat is transferred from the steam to the feedwater without mixing.

The production of more than one useful form of energy (such as process heat and electric power) from the same energy source is called *cogeneration.* Cogeneration plants produce electric power while meeting the process heat requirements of certain industrial processes. This way, more of the energy transferred to the fluid in the boiler is utilized for a useful purpose. The fraction of energy that is used for either process heat or power generation is called the *utilization factor* of the cogeneration plant.

The overall thermal efficiency of a power plant can be increased by using a *combined cycle.* The most common combined cycle is the gas–steam combined cycle where a gas-turbine cycle operates at the high-temperature range and a steam-turbine cycle at the low-temperature range. Steam is heated by the high-temperature exhaust gases leaving the gas turbine. Combined cycles have a higher thermal efficiency than the steam- or gas-turbine cycles operating alone.

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Carnot Vapor Cycle

10–1C Why is excessive moisture in steam undesirable in steam turbines? What is the highest moisture content allowed?

10–2C Why is the Carnot cycle not a realistic model for steam power plants?

10–3E Water enters the boiler of a steady-flow Carnot engine as a saturated liquid at 180 psia and leaves with a quality of 0.90. Steam leaves the turbine at a pressure of 14.7 psia. Show the cycle on a *T-s* diagram relative to the saturation lines, and determine (*a*) the thermal efficiency, (*b*) the quality at the end of the isothermal heat-rejection process, and (*c*) the net work output. *Answers: (a) 19.3 percent, (b) 0.153, (c) 148 Btu/lbm*

10–4 A steady-flow Carnot cycle uses water as the working fluid. Water changes from saturated liquid to saturated vapor as heat is transferred to it from a source at 250°C. Heat rejection takes place at a pressure of 20 kPa. Show the cycle on a *T-s* diagram relative to the saturation lines, and determine (*a*) the thermal efficiency, (*b*) the amount of heat rejected, in kJ/kg, and (*c*) the net work output.

10–5 Repeat Prob. 10–4 for a heat rejection pressure of 10 kPa.

10–6 Consider a steady-flow Carnot cycle with water as the working fluid. The maximum and minimum temperatures in the cycle are 350 and 60°C. The quality of water is 0.891 at the beginning of the heat-rejection process and 0.1 at the end. Show the cycle on a *T-s* diagram relative to the saturation lines, and determine (*a*) the thermal efficiency, (*b*) the pressure at the turbine inlet, and (*c*) the net work output. *Answers: (a) 0.465, (b) 1.40 MPa, (c) 1623 kJ/kg*

The Simple Rankine Cycle

10–7C What four processes make up the simple ideal Rankine cycle?

10–8C Consider a simple ideal Rankine cycle with fixed turbine inlet conditions. What is the effect of lowering the condenser pressure on

Pump work input:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Turbine work output:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat supplied:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat rejected:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Cycle efficiency:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Moisture content at turbine exit:	(*a*) increases, (*b*) decreases, (*c*) remains the same

10–9C Consider a simple ideal Rankine cycle with fixed turbine inlet temperature and condenser pressure. What is the effect of increasing the boiler pressure on

Pump work input:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Turbine work output:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat supplied:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat rejected:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Cycle efficiency:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Moisture content at turbine exit:	(*a*) increases, (*b*) decreases, (*c*) remains the same

10–10C Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. What is the effect of superheating the steam to a higher temperature on

Pump work input:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Turbine work output:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat supplied:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat rejected:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Cycle efficiency:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Moisture content at turbine exit:	(*a*) increases, (*b*) decreases, (*c*) remains the same

10–11C How do actual vapor power cycles differ from idealized ones?

10–12C Compare the pressures at the inlet and the exit of the boiler for (*a*) actual and (*b*) ideal cycles.

10–13C The entropy of steam increases in actual steam turbines as a result of irreversibilities. In an effort to control entropy increase, it is proposed to cool the steam in the turbine by running cooling water around the turbine casing. It is argued that this will reduce the entropy and the enthalpy of the steam at the turbine exit and thus increase the work output. How would you evaluate this proposal?

10–14C Is it possible to maintain a pressure of 10 kPa in a condenser that is being cooled by river water entering at 20°C?

10–15 A steam power plant operates on a simple ideal Rankine cycle between the pressure limits of 3 MPa and 50 kPa. The temperature of the steam at the turbine inlet is 300°C, and the mass flow rate of steam through the cycle is 35 kg/s. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the thermal efficiency of the cycle and (*b*) the net power output of the power plant.

10–16 Consider a 210-MW steam power plant that operates on a simple ideal Rankine cycle. Steam enters the turbine at 10 MPa and 500°C and is cooled in the condenser at a pressure of 10 kPa. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the quality of the steam at the turbine exit, (*b*) the thermal efficiency of the cycle, and (*c*) the mass flow rate of the steam. *Answers:* (*a*) 0.793, (*b*) 40.2 percent, (*c*) 165 kg/s

10–17 Repeat Prob. 10–16 assuming an isentropic efficiency of 85 percent for both the turbine and the pump. *Answers:* (*a*) 0.874, (*b*) 34.1 percent, (*c*) 194 kg/s

10–18E A steam power plant operates on a simple ideal Rankine cycle between the pressure limits of 1250 and 2 psia. The mass flow rate of steam through the cycle is 75 lbm/s. The moisture content of the steam at the turbine exit is not to exceed 10 percent. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the minimum turbine inlet temperature, (*b*) the rate of heat input in the boiler, and (*c*) the thermal efficiency of the cycle.

10–19E Repeat Prob. 10–18E assuming an isentropic efficiency of 85 percent for both the turbine and the pump.

10–20 Consider a coal-fired steam power plant that produces 300 MW of electric power. The power plant operates on a simple ideal Rankine cycle with turbine inlet conditions of 5 MPa and 450°C and a condenser pressure of 25 kPa. The coal has a heating value (energy released when the fuel is burned) of 29,300 kJ/kg. Assuming that 75 percent of this energy is transferred to the steam in the boiler and that the electric generator has an efficiency of 96 percent, determine (*a*) the overall plant efficiency (the ratio of net electric power output to the energy input as fuel) and (*b*) the required rate of coal supply. *Answers:* (*a*) 24.5 percent, (*b*) 150 t/h

10–21 Consider a solar-pond power plant that operates on a simple ideal Rankine cycle with refrigerant-134a as the working fluid. The refrigerant enters the turbine as a saturated vapor at 1.4 MPa and leaves at 0.7 MPa. The mass flow rate of the refrigerant is 3 kg/s. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the thermal efficiency of the cycle and (*b*) the power output of this plant.

10–22 Consider a steam power plant that operates on a simple ideal Rankine cycle and has a net power output of 45 MW. Steam enters the turbine at 7 MPa and 500°C and is cooled in the condenser at a pressure of 10 kPa by running cooling water from a lake through the tubes of the condenser at a rate of 2000 kg/s. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the thermal efficiency of the cycle, (*b*) the mass flow rate of the steam, and (*c*) the temperature rise of the cooling water. *Answers:* (*a*) 38.9 percent, (*b*) 36 kg/s, (*c*) 8.4°C

10–23 Repeat Prob. 10–22 assuming an isentropic efficiency of 87 percent for both the turbine and the pump. *Answers:* (*a*) 33.8 percent, (*b*) 41.4 kg/s, (*c*) 10.5°C

10–24 The net work output and the thermal efficiency for the Carnot and the simple ideal Rankine cycles with steam as the working fluid are to be calculated and compared. Steam enters the turbine in both cases at 10 MPa as a saturated vapor, and the condenser pressure is 20 kPa. In the Rankine cycle, the condenser exit state is saturated liquid and in the Carnot cycle, the boiler inlet state is saturated liquid. Draw the *T-s* diagrams for both cycles.

10–25 A binary geothermal power plant uses geothermal water at 160°C as the heat source. The cycle operates on the simple Rankine cycle with isobutane as the working fluid. Heat is transferred to the cycle by a heat exchanger in which geothermal liquid water enters at 160°C at a rate of 555.9 kg/s and leaves at 90°C. Isobutane enters the turbine at 3.25 MPa and 147°C at a rate of 305.6 kg/s, and leaves at 79.5°C and
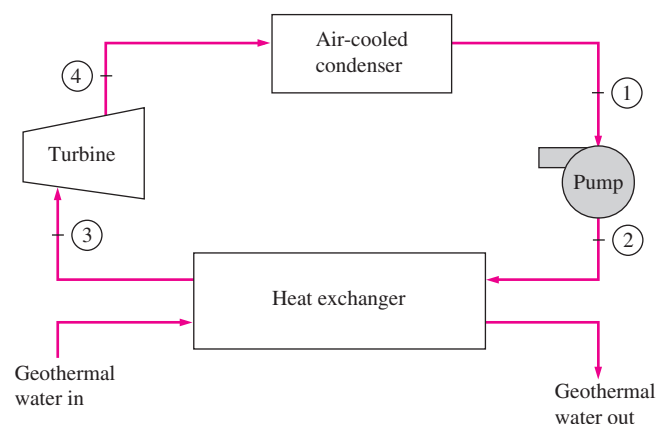

FIGURE P10–25

410 kPa. Isobutane is condensed in an air-cooled condenser and pumped to the heat exchanger pressure. Assuming the pump to have an isentropic efficiency of 90 percent, determine (*a*) the isentropic efficiency of the turbine, (*b*) the net power output of the plant, and (*c*) the thermal efficiency of the cycle.

10–26 The schematic of a single-flash geothermal power plant with state numbers is given in Fig. P10–26. Geothermal resource exists as saturated liquid at 230°C. The geothermal liquid is withdrawn from the production well at a rate of 230 kg/s, and is flashed to a pressure of 500 kPa by an essentially isenthalpic flashing process where the resulting vapor is separated from the liquid in a separator and directed to the turbine. The steam leaves the turbine at 10 kPa with a moisture content of 10 percent and enters the condenser where it is condensed and routed to a reinjection well along with the liquid coming off the separator. Determine (*a*) the mass flow rate of steam through the turbine, (*b*) the isentropic efficiency of the turbine, (*c*) the power output of the turbine, and (*d*) the thermal efficiency of the plant (the ratio of the turbine work output to the energy of the geothermal fluid relative to standard ambient conditions). *Answers:* (*a*) 38.2 kg/s, (*b*) 0.686, (*c*) 15.4 MW, (*d*) 7.6 percent
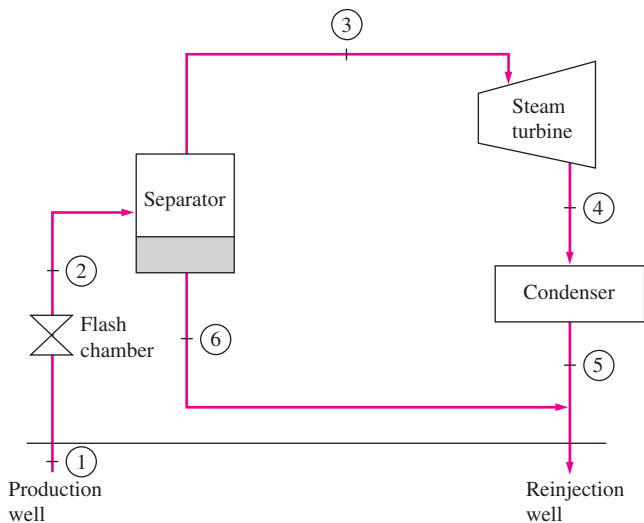

FIGURE P10–26

10–27 Reconsider Prob. 10–26. Now, it is proposed that the liquid water coming out of the separator be routed through another flash chamber maintained at 150 kPa, and the steam produced be directed to a lower stage of the same turbine. Both streams of steam leave the turbine at the same state of 10 kPa and 90 percent quality. Determine (*a*) the temperature of steam at the outlet of the second flash chamber, (*b*) the power produced by the lower stage of the turbine, and (*c*) the thermal efficiency of the plant.
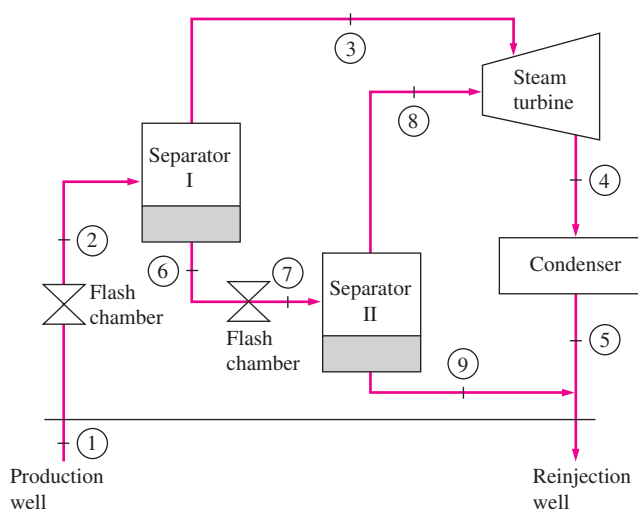

FIGURE P10–27

10–28 Reconsider Prob. 10–26. Now, it is proposed that the liquid water coming out of the separator be used as the heat source in a binary cycle with isobutane as the working fluid. Geothermal liquid water leaves the heat exchanger at 90°C while isobutane enters the turbine at 3.25 MPa and 145°C and leaves at 80°C and 400 kPa. Isobutane is condensed in an air-cooled condenser and then pumped to the heat exchanger pressure. Assuming an isentropic efficiency of 90 percent for the pump, determine (*a*) the mass flow rate of isobutane in the binary cycle, (*b*) the net power outputs of both the flashing and the binary sections of the plant, and (*c*) the thermal efficiencies of the binary cycle and the combined plant. *Answers:* (*a*) 105.5 kg/s, (*b*) 15.4 MW, 6.14 MW, (*c*) 12.2 percent, 10.6 percent
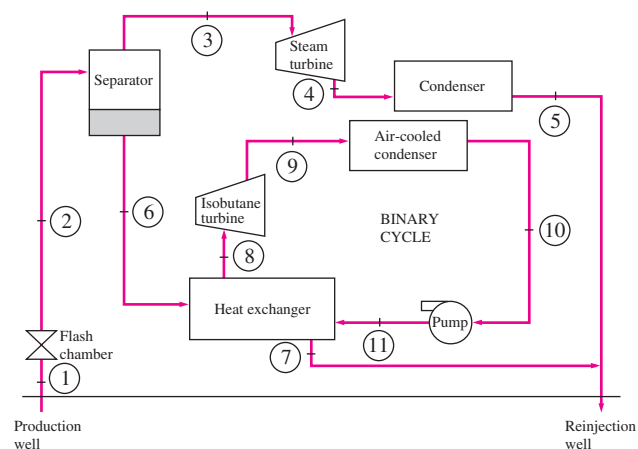

FIGURE P10–28

The Reheat Rankine Cycle

10–29C How do the following quantities change when a simple ideal Rankine cycle is modified with reheating? Assume the mass flow rate is maintained the same.

Pump work input:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Turbine work output:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat supplied:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Heat rejected:	(*a*) increases, (*b*) decreases, (*c*) remains the same
Moisture content at turbine exit:	(*a*) increases, (*b*) decreases, (*c*) remains the same

10–30C Show the ideal Rankine cycle with three stages of reheating on a *T-s* diagram. Assume the turbine inlet temperature is the same for all stages. How does the cycle efficiency vary with the number of reheat stages?

10–31C Consider a simple Rankine cycle and an ideal Rankine cycle with three reheat stages. Both cycles operate between the same pressure limits. The maximum temperature is 700°C in the simple cycle and 450°C in the reheat cycle. Which cycle do you think will have a higher thermal efficiency?

10–32 A steam power plant operates on the ideal reheat Rankine cycle. Steam enters the high-pressure turbine at 8 MPa and 500°C and leaves at 3 MPa. Steam is then reheated at constant pressure to 500°C before it expands to 20 kPa in the low-pressure turbine. Determine the turbine work output, in kJ/kg, and the thermal efficiency of the cycle. Also, show the cycle on a *T-s* diagram with respect to saturation lines.

10–33 Reconsider Prob. 10–32. Using EES (or other) software, solve this problem by the diagram window data entry feature of EES. Include the effects of the turbine and pump efficiencies and also show the effects of reheat on the steam quality at the low-pressure turbine exit. Plot the cycle on a *T-s* diagram with respect to the saturation lines. Discuss the results of your parametric studies.

10–34 Consider a steam power plant that operates on a reheat Rankine cycle and has a net power output of 80 MW. Steam enters the high-pressure turbine at 10 MPa and 500°C and the low-pressure turbine at 1 MPa and 500°C. Steam leaves the condenser as a saturated liquid at a pressure of 10 kPa. The isentropic efficiency of the turbine is 80 percent, and that of the pump is 95 percent. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the quality (or temperature, if superheated) of the steam at the turbine exit, (*b*) the thermal efficiency of the cycle, and (*c*) the mass flow rate of the steam. *Answers:* (*a*) 88.1°C, (*b*) 34.1 percent, (*c*) 62.7 kg/s

10–35 Repeat Prob. 10–34 assuming both the pump and the turbine are isentropic. *Answers:* (*a*) 0.949, (*b*) 41.3 percent, (*c*) 50.0 kg/s

10–36E Steam enters the high-pressure turbine of a steam power plant that operates on the ideal reheat Rankine cycle at 800 psia and 900°F and leaves as saturated vapor. Steam is then reheated to 800°F before it expands to a pressure of 1 psia. Heat is transferred to the steam in the boiler at a rate of 6×10^4 Btu/s. Steam is cooled in the condenser by the cooling water from a nearby river, which enters the condenser at 45°F. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the pressure at which reheating takes place, (*b*) the net power output and thermal efficiency, and (*c*) the minimum mass flow rate of the cooling water required.

10–37 A steam power plant operates on an ideal reheat Rankine cycle between the pressure limits of 15 MPa and 10 kPa. The mass flow rate of steam through the cycle is 12 kg/s. Steam enters both stages of the turbine at 500°C. If the moisture content of the steam at the exit of the low-pressure turbine is not to exceed 10 percent, determine (*a*) the pressure at which reheating takes place, (*b*) the total rate of heat input in the boiler, and (*c*) the thermal efficiency of the cycle. Also, show the cycle on a *T-s* diagram with respect to saturation lines.

10–38 A steam power plant operates on the reheat Rankine cycle. Steam enters the high-pressure turbine at 12.5 MPa and 550°C at a rate of 7.7 kg/s and leaves at 2 MPa. Steam is then reheated at constant pressure to 450°C before it expands in the low-pressure turbine. The isentropic efficiencies of the turbine and the pump are 85 percent and 90 percent, respectively. Steam leaves the condenser as a saturated liquid. If the moisture content of the steam at the exit of the turbine is not to exceed 5 percent, determine (*a*) the condenser pressure, (*b*) the net power output, and (*c*) the thermal efficiency. *Answers:* (*a*) 9.73 kPa, (*b*) 10.2 MW, (*c*) 36.9 percent
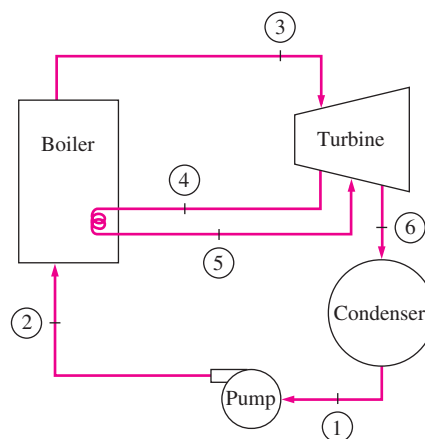

FIGURE P10–38

Regenerative Rankine Cycle

10–39C How do the following quantities change when the simple ideal Rankine cycle is modified with regeneration? Assume the mass flow rate through the boiler is the same.

Turbine work output:	(a) increases, (b) decreases, (c) remains the same
Heat supplied:	(a) increases, (b) decreases, (c) remains the same
Heat rejected:	(a) increases, (b) decreases, (c) remains the same
Moisture content at turbine exit:	(a) increases, (b) decreases, (c) remains the same

10–40C During a regeneration process, some steam is extracted from the turbine and is used to heat the liquid water leaving the pump. This does not seem like a smart thing to do since the extracted steam could produce some more work in the turbine. How do you justify this action?

10–41C How do open feedwater heaters differ from closed feedwater heaters?

10–42C Consider a simple ideal Rankine cycle and an ideal regenerative Rankine cycle with one open feedwater heater. The two cycles are very much alike, except the feedwater in the regenerative cycle is heated by extracting some steam just before it enters the turbine. How would you compare the efficiencies of these two cycles?

10–43C Devise an ideal regenerative Rankine cycle that has the same thermal efficiency as the Carnot cycle. Show the cycle on a T-s diagram.

10–44 A steam power plant operates on an ideal regenerative Rankine cycle. Steam enters the turbine at 6 MPa and 450°C and is condensed in the condenser at 20 kPa. Steam is extracted from the turbine at 0.4 MPa to heat the feedwater in an open feedwater heater. Water leaves the feedwater heater as a saturated liquid. Show the cycle on a T-s diagram, and determine (a) the net work output per kilogram of steam flowing through the boiler and (b) the thermal efficiency of the cycle. *Answers: (a)* 1017 kJ/kg, *(b)* 37.8 percent

10–45 Repeat Prob. 10–44 by replacing the open feedwater heater with a closed feedwater heater. Assume that the feedwater leaves the heater at the condensation temperature of the extracted steam and that the extracted steam leaves the heater as a saturated liquid and is pumped to the line carrying the feedwater.

10–46 A steam power plant operates on an ideal regenerative Rankine cycle with two open feedwater heaters. Steam enters the turbine at 10 MPa and 600°C and exhausts to the condenser at 5 kPa. Steam is extracted from the turbine at 0.6 and 0.2 MPa. Water leaves both feedwater heaters as a saturated liquid. The mass flow rate of steam through the boiler is 22 kg/s. Show the cycle on a T-s diagram, and determine (a) the net power output of the power plant and (b) the

thermal efficiency of the cycle. *Answers: (a)* 30.5 MW, *(b)* 47.1 percent

10–47 Consider an ideal steam regenerative Rankine cycle with two feedwater heaters, one closed and one open. Steam enters the turbine at 12.5 MPa and 550°C and exhausts to the condenser at 10 kPa. Steam is extracted from the turbine at 0.8 MPa for the closed feedwater heater and at 0.3 MPa for the open one. The feedwater is heated to the condensation temperature of the extracted steam in the closed feedwater heater. The extracted steam leaves the closed feedwater heater as a saturated liquid, which is subsequently throttled to the open feedwater heater. Show the cycle on a T-s diagram with respect to saturation lines, and determine (a) the mass flow rate of steam through the boiler for a net power output of 250 MW and (b) the thermal efficiency of the cycle.
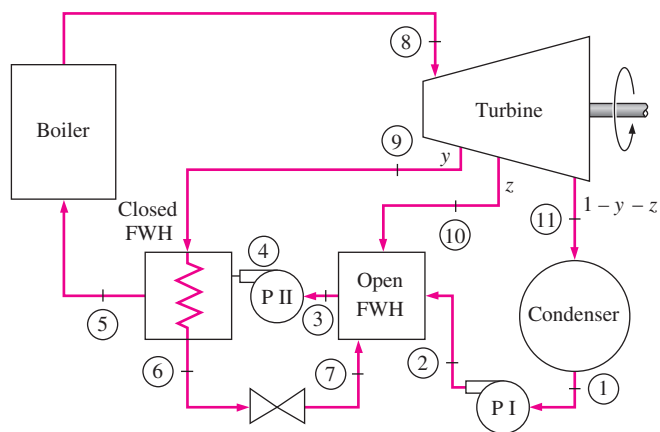

FIGURE P10–47

10–48 Reconsider Prob. 10–47. Using EES (or other) software, investigate the effects of turbine and pump efficiencies as they are varied from 70 percent to 100 percent on the mass flow rate and thermal efficiency. Plot the mass flow rate and the thermal efficiency as a function of turbine efficiency for pump efficiencies of 70, 85, and 100 percent, and discuss the results. Also plot the T-s diagram for turbine and pump efficiencies of 85 percent.

10–49 A steam power plant operates on an ideal reheat–regenerative Rankine cycle and has a net power output of 80 MW. Steam enters the high-pressure turbine at 10 MPa and 550°C and leaves at 0.8 MPa. Some steam is extracted at this pressure to heat the feedwater in an open feedwater heater. The rest of the steam is reheated to 500°C and is expanded in the low-pressure turbine to the condenser pressure of 10 kPa. Show the cycle on a T-s diagram with respect to saturation lines, and determine (a) the mass flow rate of steam through the boiler and (b) the thermal efficiency of the cycle. *Answers: (a)* 54.5 kg/s, *(b)* 44.4 percent

10–50 Repeat Prob. 10–49, but replace the open feedwater heater with a closed feedwater heater. Assume that the feed-

water leaves the heater at the condensation temperature of the extracted steam and that the extracted steam leaves the heater as a saturated liquid and is pumped to the line carrying the feedwater.
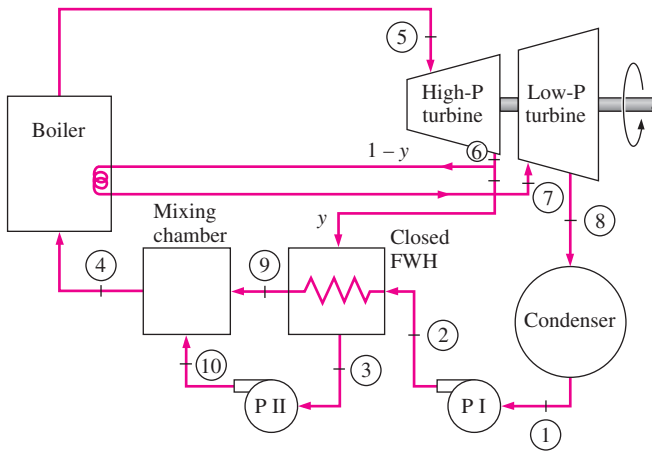

FIGURE P10–50

10–51E A steam power plant operates on an ideal reheat–regenerative Rankine cycle with one reheater and two open feedwater heaters. Steam enters the high-pressure turbine at 1500 psia and 1100°F and leaves the low-pressure turbine at 1 psia. Steam is extracted from the turbine at 250 and 40 psia, and it is reheated to 1000°F at a pressure of 140 psia. Water leaves both feedwater heaters as a saturated liquid. Heat is transferred to the steam in the boiler at a rate of 4×10^5 Btu/s. Show the cycle on a T-s diagram with respect to saturation lines, and determine (a) the mass flow rate of steam through the boiler, (b) the net power output of the plant, and (c) the thermal efficiency of the cycle.
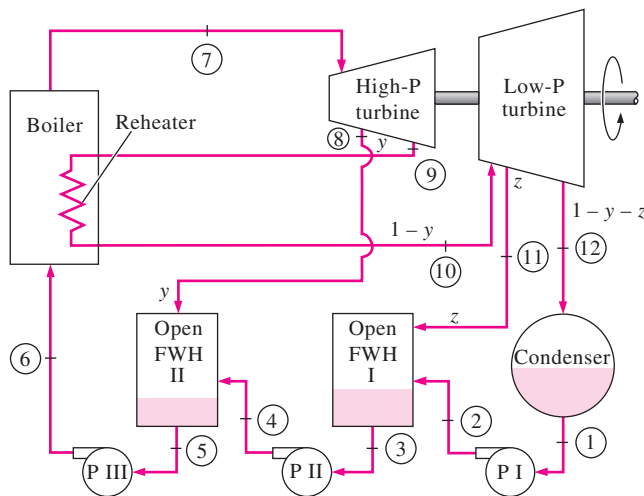

FIGURE P10–51E

10–52 A steam power plant operates on the reheat-regenerative Rankine cycle with a closed feedwater heater. Steam enters the turbine at 12.5 MPa and 550°C at a rate of 24 kg/s and is condensed in the condenser at a pressure of 20 kPa. Steam is reheated at 5 MPa to 550°C. Some steam is extracted from the low-pressure turbine at 1.0 MPa, is completely condensed in the closed feedwater heater, and pumped to 12.5 MPa before it mixes with the feedwater at the same pressure. Assuming an isentropic efficiency of 88 percent for both the turbine and the pump, determine (a) the temperature of the steam at the inlet of the closed feedwater heater, (b) the mass flow rate of the steam extracted from the turbine for the closed feedwater heater, (c) the net power output, and (d) the thermal efficiency. *Answers:* (a) 328°C, (b) 4.29 kg/s, (c) 28.6 MW, (d) 39.3 percent
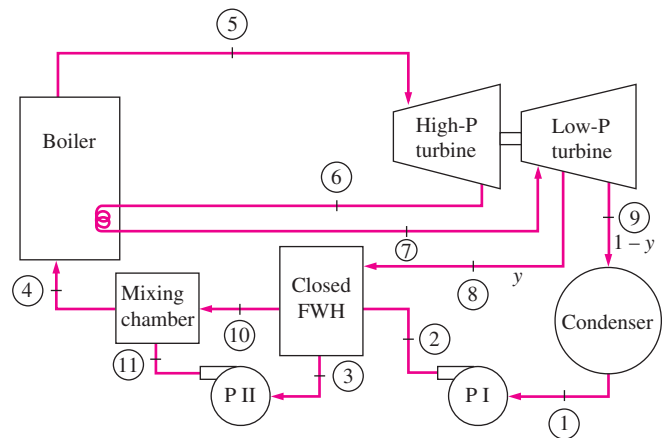

FIGURE P10–52

Second-Law Analysis of Vapor Power Cycles

10–53C How can the second-law efficiency of a simple ideal Rankine cycle be improved?

10–54 Determine the exergy destruction associated with each of the processes of the Rankine cycle described in Prob. 10–15, assuming a source temperature of 1500 K and a sink temperature of 290 K.

10–55 Determine the exergy destruction associated with each of the processes of the Rankine cycle described in Prob. 10–16, assuming a source temperature of 1500 K and a sink temperature of 290 K. *Answers:* 0, 1112 kJ/kg, 0, 172.3 kJ/kg

10–56 Determine the exergy destruction associated with the heat rejection process in Prob. 10–22. Assume a source temperature of 1500 K and a sink temperature of 290 K. Also, determine the exergy of the steam at the boiler exit. Take $P_0 = 100$ kPa.

10–57 Determine the exergy destruction associated with each of the processes of the reheat Rankine cycle described in Prob. 10–32. Assume a source temperature of 1800 K and a sink temperature of 300 K.

10–58 [EES] Reconsider Prob. 10–57. Using EES (or other) software, solve this problem by the diagram window data entry feature of EES. Include the effects of the turbine and pump efficiencies to evaluate the irreversibilities associated with each of the processes. Plot the cycle on a *T-s* diagram with respect to the saturation lines. Discuss the results of your parametric studies.

10–59 Determine the exergy destruction associated with the heat addition process and the expansion process in Prob. 10–34. Assume a source temperature of 1600 K and a sink temperature of 285 K. Also, determine the exergy of the steam at the boiler exit. Take $P_0 = 100$ kPa.   *Answers:* 1289 kJ/kg, 247.9 kJ/kg, 1495 kJ/kg

10–60 Determine the exergy destruction associated with the regenerative cycle described in Prob. 10–44. Assume a source temperature of 1500 K and a sink temperature of 290 K. *Answer:* 1155 kJ/kg

10–61 Determine the exergy destruction associated with the reheating and regeneration processes described in Prob. 10–49. Assume a source temperature of 1800 K and a sink temperature of 290 K.

10–62 The schematic of a single-flash geothermal power plant with state numbers is given in Fig. P10–62. Geothermal resource exists as saturated liquid at 230°C. The geothermal liquid is withdrawn from the production well at a rate of 230 kg/s and is flashed to a pressure of 500 kPa by an essentially isenthalpic flashing process where the resulting vapor is separated from the liquid in a separator and is directed to the turbine. The steam leaves the turbine at 10 kPa with a moisture content of 5 percent and enters the condenser where it is condensed; it is routed to a reinjection well along with the liquid coming off the separator. Determine (*a*) the power output of the turbine and the thermal efficiency of the plant, (*b*) the exergy of the geo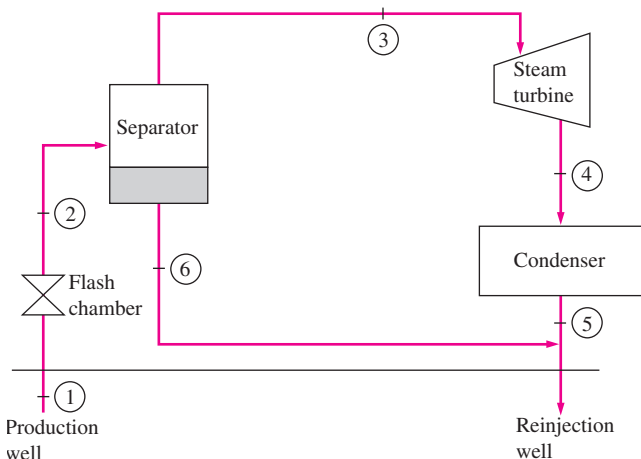thermal liquid at the exit of the flash chamber, and the exergy destructions and the second-law (exergetic) efficiencies for (*c*) the flash chamber, (*d*) the turbine, and (*e*) the entire plant.   *Answers:* (*a*) 10.8 MW, 0.053, (*b*) 17.3 MW, (*c*) 5.1 MW, 0.898, (*d*) 10.9 MW, 0.500, (*e*) 39.0 MW, 0.218

Cogeneration

10–63C How is the utilization factor ϵ_u for cogeneration plants defined? Could ϵ_u be unity for a cogeneration plant that does not produce any power?

10–64C Consider a cogeneration plant for which the utilization factor is 1. Is the irreversibility associated with this cycle necessarily zero? Explain.

10–65C Consider a cogeneration plant for which the utilization factor is 0.5. Can the exergy destruction associated with this plant be zero? If yes, under what conditions?

10–66C What is the difference between cogeneration and regeneration?

10–67 Steam enters the turbine of a cogeneration plant at 7 MPa and 500°C. One-fourth of the steam is extracted from the turbine at 600-kPa pressure for process heating. The remaining steam continues to expand to 10 kPa. The extracted steam is then condensed and mixed with feedwater at constant pressure and the mixture is pumped to the boiler pressure of 7 MPa. The mass flow rate of steam through the boiler is 30 kg/s. Disregarding any pressure drops and heat losses in the piping, and assuming the turbine and the pump to be isentropic, determine the net power produced and the utilization factor of the plant.
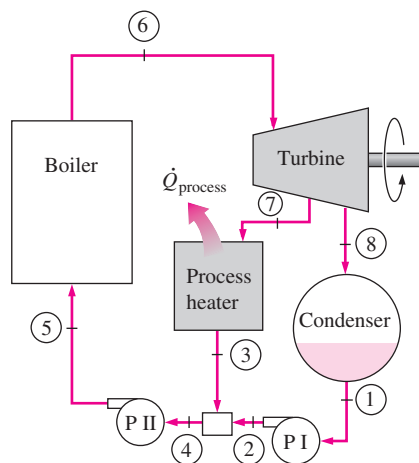

FIGURE P10–67

10–68E A large food-processing plant requires 2 lbm/s of saturated or slightly superheated steam at 80 psia, which is extracted from the turbine of a cogeneration plant. The boiler generates steam at 1000 psia and 1000°F at a rate of 5 lbm/s,

FIGURE P10–62

and the condenser pressure is 2 psia. Steam leaves the process heater as a saturated liquid. It is then mixed with the feedwater at the same pressure and this mixture is pumped to the boiler pressure. Assuming both the pumps and the turbine have isentropic efficiencies of 86 percent, determine (*a*) the rate of heat transfer to the boiler and (*b*) the power output of the cogeneration plant. *Answers:* (*a*) 6667 Btu/s, (*b*) 2026 kW

10–69   Steam is generated in the boiler of a cogeneration plant at 10 MPa and 450°C at a steady rate of 5 kg/s. In normal operation, steam expands in a turbine to a pressure of 0.5 MPa and is then routed to the process heater, where it supplies the process heat. Steam leaves the process heater as a saturated liquid and is pumped to the boiler pressure. In this mode, no steam passes through the condenser, which operates at 20 kPa.

(*a*) Determine the power produced and the rate at which process heat is supplied in this mode.

(*b*) Determine the power produced and the rate of process heat supplied if only 60 percent of the steam is routed to the process heater and the remainder is expanded to the condenser pressure.

10–70   Consider a cogeneration power plant modified with regeneration. Steam enters the turbine at 6 MPa and 450°C and expands to a pressure of 0.4 MPa. At this pressure, 60 percent of the steam is extracted from the turbine, and the remainder expands to 10 kPa. Part of the extracted steam is used to heat the feedwater in an open feedwater heater. The rest of the extracted steam is used for process heating and leaves the process heater as a saturated liquid at 0.4 MPa. It is subsequently mixed with the feedwater leaving the feedwater heater, and the mixture is pumped to the boiler pressure.
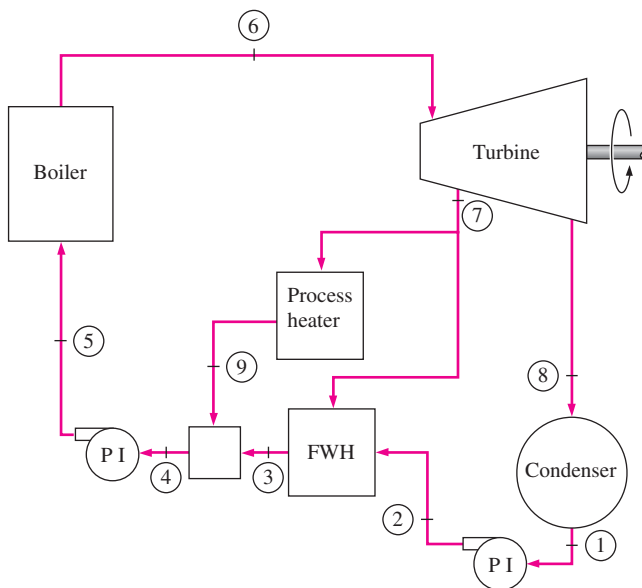
Assuming the turbines and the pumps to be isentropic, show the cycle on a *T-s* diagram with respect to saturation lines, and determine the mass flow rate of steam through the boiler for a net power output of 15 MW. *Answer:* 17.7 kg/s

10–71   [EES] Reconsider Prob. 10–70. Using EES (or other) software, investigate the effect of the extraction pressure for removing steam from the turbine to be used for the process heater and open feedwater heater on the required mass flow rate. Plot the mass flow rate through the boiler as a function of the extraction pressure, and discuss the results.

10–72E   Steam is generated in the boiler of a cogeneration plant at 600 psia and 800°F at a rate of 18 lbm/s. The plant is to produce power while meeting the process steam requirements for a certain industrial application. One-third of the steam leaving the boiler is throttled to a pressure of 120 psia and is routed to the process heater. The rest of the steam is expanded in an isentropic turbine to a pressure of 120 psia and is also routed to the process heater. Steam leaves the process heater at 240°F. Neglecting the pump work, determine (*a*) the net power produced, (*b*) the rate of process heat supply, and (*c*) the utilization factor of this plant.

10–73   A cogeneration plant is to generate power and 8600 kJ/s of process heat. Consider an ideal cogeneration steam plant. Steam enters the turbine from the boiler at 7 MPa and 500°C. One-fourth of the steam is extracted from the turbine at 600-kPa pressure for process heating. The remainder of the steam continues to expand and exhausts to the condenser at 10 kPa. The steam extracted for the process heater is condensed in the heater and mixed with the feedwater at 600 kPa. The mixture is pumped to the boiler pressure of 7 MPa. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the mass flow rate of steam that must be supplied by the boiler, (*b*) the net power produced by the plant, and (*c*) the utilization factor.
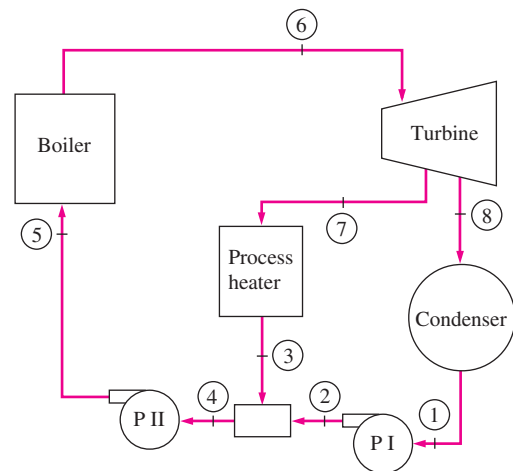

FIGURE P10–70

FIGURE P10–73

Combined Gas–Vapor Power Cycles

10–74C In combined gas–steam cycles, what is the energy source for the steam?

10–75C Why is the combined gas–steam cycle more efficient than either of the cycles operated alone?

10–76 The gas-turbine portion of a combined gas–steam power plant has a pressure ratio of 16. Air enters the compressor at 300 K at a rate of 14 kg/s and is heated to 1500 K in the combustion chamber. The combustion gases leaving the gas turbine are used to heat the steam to 400°C at 10 MPa in a heat exchanger. The combustion gases leave the heat exchanger at 420 K. The steam leaving the turbine is condensed at 15 kPa. Assuming all the compression and expansion processes to be isentropic, determine (*a*) the mass flow rate of the steam, (*b*) the net power output, and (*c*) the thermal efficiency of the combined cycle. For air, assume constant specific heats at room temperature. *Answers:* (*a*) 1.275 kg/s, (*b*) 7819 kW, (*c*) 66.4 percent

10–77 Consider a combined gas–steam power plant that has a net power output of 450 MW. The pressure ratio of the gas-turbine cycle is 14. Air enters the compressor at 300 K and the turbine at 1400 K. The combustion gases leaving the gas turbine are used to heat the steam at 8 MPa to 400°C in a heat exchanger. The combustion gases leave the heat exchanger at 460 K. An open feedwater heater incorporated with the steam cycle operates at a pressure of 0.6 MPa. The condenser pressure is 20 kPa. Assuming all the compression and expansion processes to be isentropic, determine (*a*) the mass flow rate ratio of air to steam, (*b*) the required rate of heat input in the combustion chamber, and (*c*) the thermal efficiency of the combined cycle.

10–78 Reconsider Prob. 10–77. Using EES (or other) software, study the effects of the gas cycle pressure ratio as it is varied from 10 to 20 on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency. Plot your results as functions of gas cycle pressure ratio, and discuss the results.

10–79 Repeat Prob. 10–77 assuming isentropic efficiencies of 100 percent for the pump, 82 percent for the compressor, and 86 percent for the gas and steam turbines.

10–80 Reconsider Prob. 10–79. Using EES (or other) software, study the effects of the gas cycle pressure ratio as it is varied from 10 to 20 on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency. Plot your results as functions of gas cycle pressure ratio, and discuss the results.

10–81 Consider a combined gas–steam power cycle. The topping cycle is a simple Brayton cycle that has a pressure ratio of 7. Air enters the compressor at 15°C at a rate of 10 kg/s and the gas turbine at 950°C. The bottoming cycle is a reheat Rankine cycle between the pressure limits of 6 MPa and 10 kPa. Steam is heated in a heat exchanger at a rate of 1.15 kg/s by the exhaust gases leaving the gas turbine and the exhaust gases leave the heat exchanger at 200°C. Steam leaves the high-pressure turbine at 1.0 MPa and is reheated to 400°C in the heat exchanger before it expands in the low-pressure turbine. Assuming 80 percent isentropic efficiency for all pumps and turbine, determine (*a*) the moisture content at the exit of the low-pressure turbine, (*b*) the steam temperature at the inlet of the high-pressure turbine, (*c*) the net power output and the thermal efficiency of the combined plant.
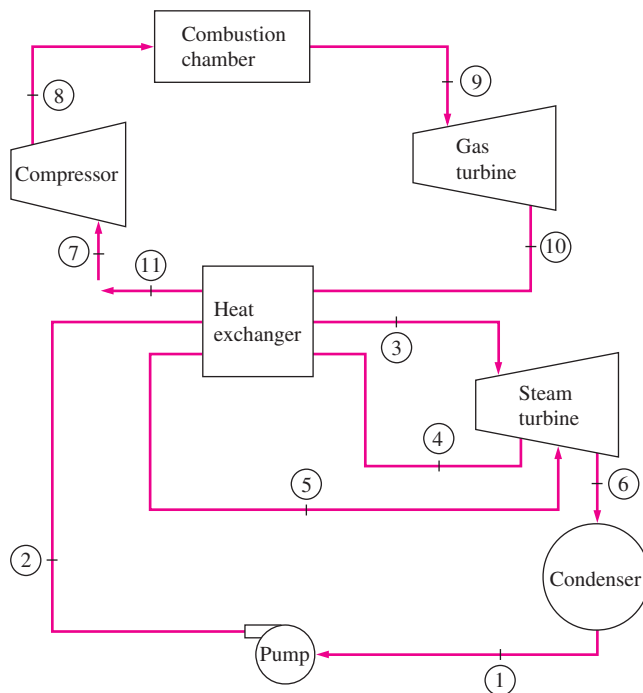

FIGURE P10–81

Special Topic: Binary Vapor Cycles

10–82C What is a binary power cycle? What is its purpose?

10–83C By writing an energy balance on the heat exchanger of a binary vapor power cycle, obtain a relation for the ratio of mass flow rates of two fluids in terms of their enthalpies.

10–84C Why is steam not an ideal working fluid for vapor power cycles?

10–85C Why is mercury a suitable working fluid for the topping portion of a binary vapor cycle but not for the bottoming cycle?

10–86C What is the difference between the binary vapor power cycle and the combined gas–steam power cycle?

Review Problems

10–87 Show that the thermal efficiency of a combined gas–steam power plant η_{cc} can be expressed as

$$\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s$$

where $\eta_g = W_g/Q_{in}$ and $\eta_s = W_s/Q_{g,out}$ are the thermal efficiencies of the gas and steam cycles, respectively. Using this relation, determine the thermal efficiency of a combined power cycle that consists of a topping gas-turbine cycle with an efficiency of 40 percent and a bottoming steam-turbine cycle with an efficiency of 30 percent.

10–88 It can be shown that the thermal efficiency of a combined gas–steam power plant η_{cc} can be expressed in terms of the thermal efficiencies of the gas- and the steam-turbine cycles as

$$\eta_{cc} = \eta_g + \eta_s - \eta_g\eta_s$$

Prove that the value of η_{cc} is greater than either of η_g or η_s. That is, the combined cycle is more efficient than either of the gas-turbine or steam-turbine cycles alone.

10–89 Consider a steam power plant operating on the ideal Rankine cycle with reheat between the pressure limits of 25 MPa and 10 kPa with a maximum cycle temperature of 600°C and a moisture content of 8 percent at the turbine exit. For a reheat temperature of 600°C, determine the reheat pressures of the cycle for the cases of (a) single and (b) double reheat.

10–90E The Stillwater geothermal power plant in Nevada, which started full commercial operation in 1986, is designed to operate with seven identical units. Each of these seven units consists of a pair of power cycles, labeled Level I and Level II, operating on the simple Rankine cycle using an organic fluid as the working fluid.

The heat source for the plant is geothermal water (brine) entering the vaporizer (boiler) of Level I of each unit at 325°F at a rate of 384,286 lbm/h and delivering 22.79 MBtu/h ("M" stands for "million"). The organic fluid that enters the vaporizer at 202.2°F at a rate of 157,895 lbm/h leaves it at 282.4°F and 225.8 psia as saturated vapor. This saturated vapor expands in the turbine to 95.8°F and 19.0 psia and produces 1271 kW of electric power. About 200 kW of this power is used by the pumps, the auxiliaries, and the six fans of the condenser. Subsequently, the organic working fluid is condensed in an air-cooled condenser by air that enters the condenser at 55°F at a rate of 4,195,100 lbm/h and leaves at 84.5°F. The working fluid is pumped and then preheated in a preheater to 202.2°F by absorbing 11.14 MBtu/h of heat from the geothermal water (coming from the vaporizer of Level II) entering the preheater at 211.8°F and leaving at 154.0°F.

Taking the average specific heat of the geothermal water to be 1.03 Btu/lbm · °F, determine (a) the exit temperature of the geothermal water from the vaporizer, (b) the rate of heat

rejection from the working fluid to the air in the condenser, (c) the mass flow rate of the geothermal water at the preheater, and (d) the thermal efficiency of the Level I cycle of this geothermal power plant. *Answers:* (a) 267.4°F, (b) 29.7 MBtu/h, (c) 187,120 lbm/h, (d) 10.8 percent
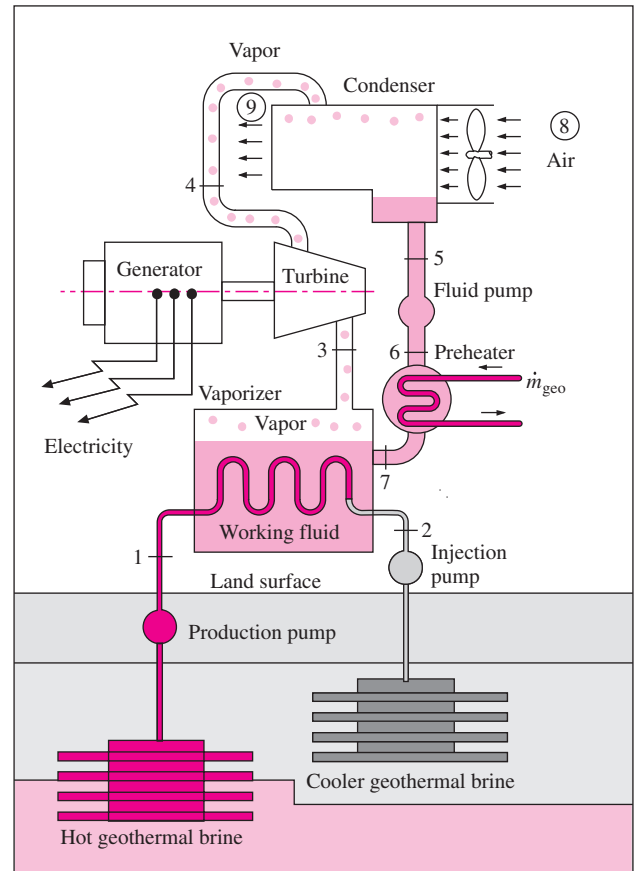

FIGURE P10–90E

Schematic of a binary geothermal power plant.

Courtesy of ORMAT Energy Systems, Inc.

10–91 Steam enters the turbine of a steam power plant that operates on a simple ideal Rankine cycle at a pressure of 6 MPa, and it leaves as a saturated vapor at 7.5 kPa. Heat is transferred to the steam in the boiler at a rate of 40,000 kJ/s. Steam is cooled in the condenser by the cooling water from a nearby river, which enters the condenser at 15°C. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (a) the turbine inlet temperature, (b) the net power output and thermal efficiency, and (c) the minimum mass flow rate of the cooling water required.

10–92 A steam power plant operates on an ideal Rankine cycle with two stages of reheat and has a net power output of

120 MW. Steam enters all three stages of the turbine at 500°C. The maximum pressure in the cycle is 15 MPa, and the minimum pressure is 5 kPa. Steam is reheated at 5 MPa the first time and at 1 MPa the second time. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the thermal efficiency of the cycle and (*b*) the mass flow rate of the steam. *Answers:* (*a*) 45.5 percent, (*b*) 64.4 kg/s

10–93 Consider a steam power plant that operates on a regenerative Rankine cycle and has a net power output of 150 MW. Steam enters the turbine at 10 MPa and 500°C and the condenser at 10 kPa. The isentropic efficiency of the turbine is 80 percent, and that of the pumps is 95 percent. Steam is extracted from the turbine at 0.5 MPa to heat the feedwater in an open feedwater heater. Water leaves the feedwater heater as a saturated liquid. Show the cycle on a *T-s* diagram, and determine (*a*) the mass flow rate of steam through the boiler and (*b*) the thermal efficiency of the cycle. Also, determine the exergy destruction associated with the regeneration process. Assume a source temperature of 1300 K and a sink temperature of 303 K.
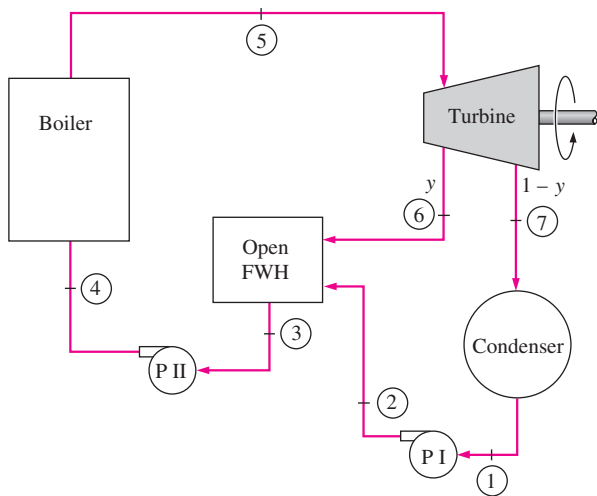

FIGURE P10–93

10–94 Repeat Prob. 10–93 assuming both the pump and the turbine are isentropic.

10–95 Consider an ideal reheat–regenerative Rankine cycle with one open feedwater heater. The boiler pressure is 10 MPa, the condenser pressure is 15 kPa, the reheater pressure is 1 MPa, and the feedwater pressure is 0.6 MPa. Steam enters both the high- and low-pressure turbines at 500°C. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the fraction of steam extracted for regeneration and (*b*) the thermal efficiency of the cycle. *Answers:* (*a*) 0.144, (*b*) 42.1 percent

10–96 Repeat Prob. 10–95 assuming an isentropic efficiency of 84 percent for the turbines and 100 percent for the pumps.

10–97 A steam power plant operates on an ideal reheat–regenerative Rankine cycle with one reheater and two feedwater heaters, one open and one closed. Steam enters the high-pressure turbine at 15 MPa and 600°C and the low-pressure turbine at 1 MPa and 500°C. The condenser pressure is 5 kPa. Steam is extracted from the turbine at 0.6 MPa for the closed feedwater heater and at 0.2 MPa for the open feedwater heater. In the closed feedwater heater, the feedwater is heated to the condensation temperature of the extracted steam. The extracted steam leaves the closed feedwater heater as a saturated liquid, which is subsequently throttled to the open feedwater heater. Show the cycle on a *T-s* diagram with respect to saturation lines. Determine (*a*) the fraction of steam extracted from the turbine for the open feedwater heater, (*b*) the thermal efficiency of the cycle, and (*c*) the net power output for a mass flow rate of 42 kg/s through the boiler.
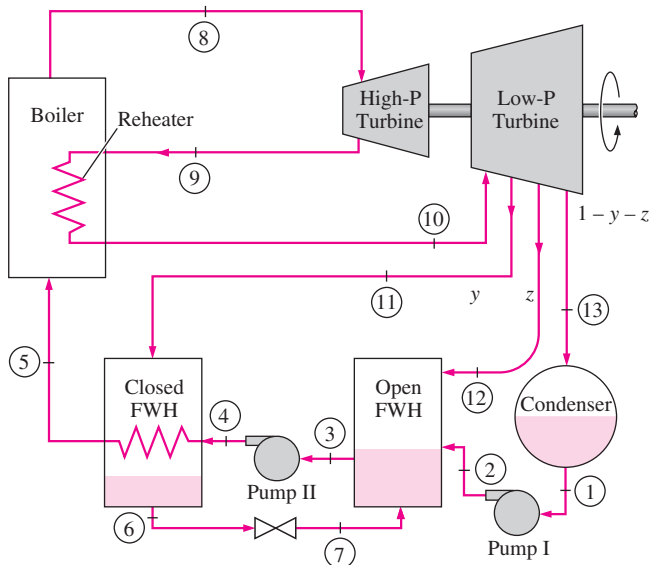

FIGURE P10–97

10–98 Consider a cogeneration power plant that is modified with reheat and that produces 3 MW of power and supplies 7 MW of process heat. Steam enters the high-pressure turbine at 8 MPa and 500°C and expands to a pressure of 1 MPa. At this pressure, part of the steam is extracted from the turbine and routed to the process heater, while the remainder is reheated to 500°C and expanded in the low-pressure turbine to the condenser pressure of 15 kPa. The condensate from the condenser is pumped to 1 MPa and is mixed with the extracted steam, which leaves the process heater as a com-

pressed liquid at 120°C. The mixture is then pumped to the boiler pressure. Assuming the turbine to be isentropic, show the cycle on a *T-s* diagram with respect to saturation lines, and disregarding pump work, determine (*a*) the rate of heat input in the boiler and (*b*) the fraction of steam extracted for process heating.
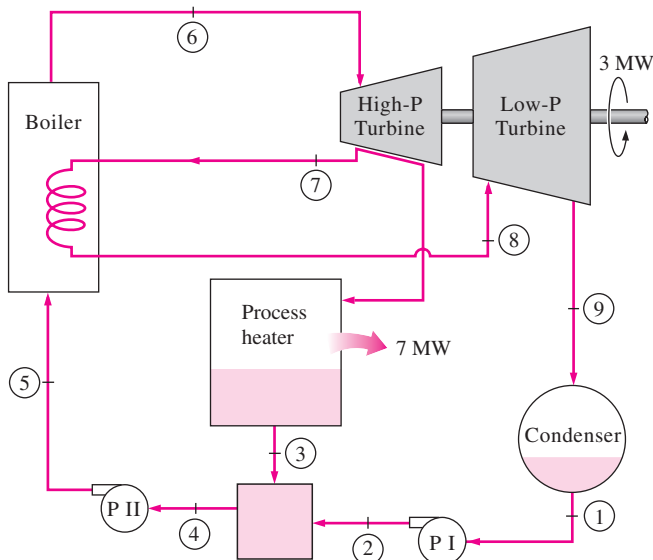

FIGURE P10–98

10–99 The gas-turbine cycle of a combined gas–steam power plant has a pressure ratio of 8. Air enters the compressor at 290 K and the turbine at 1400 K. The combustion gases leaving the gas turbine are used to heat the steam at 15 MPa to 450°C in a heat exchanger. The combustion gases leave the heat exchanger at 247°C. Steam expands in a high-pressure turbine to a pressure of 3 MPa and is reheated in the combustion chamber to 500°C before it expands in a low-pressure turbine to 10 kPa. The mass flow rate of steam is 30 kg/s. Assuming all the compression and expansion processes to be isentropic, determine (*a*) the mass flow rate of air in the gas-turbine cycle, (*b*) the rate of total heat input, and (*c*) the thermal efficiency of the combined cycle.
Answers: (*a*) 263 kg/s, (*b*) 2.80 × 10^5 kJ/s, (*c*) 55.6 percent

10–100 Repeat Prob. 10–99 assuming isentropic efficiencies of 100 percent for the pump, 80 percent for the compressor, and 85 percent for the gas and steam turbines.

10–101 Starting with Eq. 10–20, show that the exergy destruction associated with a simple ideal Rankine cycle can be expressed as $i = q_{in}(\eta_{th,Carnot} - \eta_{th})$, where η_{th} is efficiency of the Rankine cycle and $\eta_{th,Carnot}$ is the efficiency of the Carnot cycle operating between the same temperature limits.

10–102 Steam is to be supplied from a boiler to a high-pressure turbine whose isentropic efficiency is 75 percent at conditions to be determined. The steam is to leave the high-pressure turbine as a saturated vapor at 1.4 MPa, and the turbine is to produce 1 MW of power. Steam at the turbine exit is extracted at a rate of 1000 kg/min and routed to a process heater while the rest of the steam is supplied to a low-pressure turbine whose isentropic efficiency is 60 percent. The low-pressure turbine allows the steam to expand to 10 kPa pressure and produces 0.8 MW of power. Determine the temperature, pressure, and the flow rate of steam at the inlet of the high-pressure turbine.

10–103 A textile plant requires 4 kg/s of saturated steam at 2 MPa, which is extracted from the turbine of a cogeneration plant. Steam enters the turbine at 8 MPa and 500°C at a rate of 11 kg/s and leaves at 20 kPa. The extracted steam leaves the process heater as a saturated liquid and mixes with the feedwater at constant pressure. The mixture is pumped to the boiler pressure. Assuming an isentropic efficiency of 88 percent for both the turbine and the pumps, determine (*a*) the rate of process heat supply, (*b*) the net power output, and (*c*) the utilization factor of the plant. *Answers:* (*a*) 8.56 MW, (*b*) 8.60 MW, (*c*) 53.8 percent
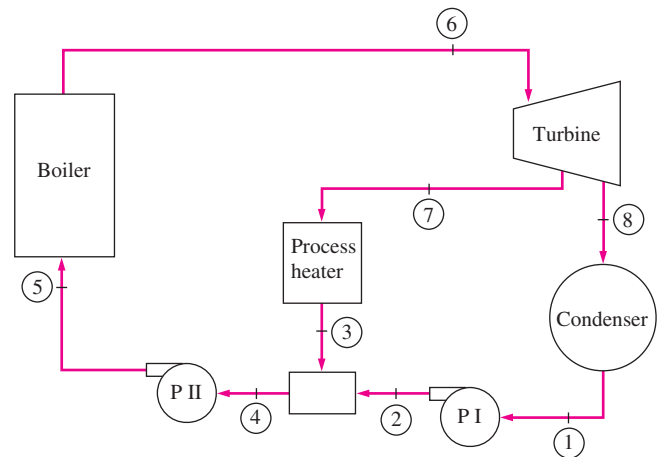

FIGURE P10–103

10–104 [EES] Using EES (or other) software, investigate the effect of the condenser pressure on the performance of a simple ideal Rankine cycle. Turbine inlet conditions of steam are maintained constant at 5 MPa and 500°C while the condenser pressure is varied from 5 to 100 kPa. Determine the thermal efficiency of the cycle and plot it against the condenser pressure, and discuss the results.

10–105 [EES] Using EES (or other) software, investigate the effect of the boiler pressure on the performance of a simple ideal Rankine cycle. Steam enters the turbine at 500°C and exits at 10 kPa. The boiler pressure is

varied from 0.5 to 20 MPa. Determine the thermal efficiency of the cycle and plot it against the boiler pressure, and discuss the results.

10–106 [EES] Using EES (or other) software, investigate the effect of superheating the steam on the performance of a simple ideal Rankine cycle. Steam enters the turbine at 3 MPa and exits at 10 kPa. The turbine inlet temperature is varied from 250 to 1100°C. Determine the thermal efficiency of the cycle and plot it against the turbine inlet temperature, and discuss the results.

10–107 [EES] Using EES (or other) software, investigate the effect of reheat pressure on the performance of an ideal Rankine cycle. The maximum and minimum pressures in the cycle are 15 MPa and 10 kPa, respectively, and steam enters both stages of the turbine at 500°C. The reheat pressure is varied from 12.5 to 0.5 MPa. Determine the thermal efficiency of the cycle and plot it against the reheat pressure, and discuss the results.

10–108 [EES] Using EES (or other) software, investigate the effect of number of reheat stages on the performance of an ideal Rankine cycle. The maximum and minimum pressures in the cycle are 15 MPa and 10 kPa, respectively, and steam enters all stages of the turbine at 500°C. For each case, maintain roughly the same pressure ratio across each turbine stage. Determine the thermal efficiency of the cycle and plot it against the number of reheat stages 1, 2, 4, and 8, and discuss the results.

10–109 [EES] Using EES (or other) software, investigate the effect of extraction pressure on the performance of an ideal regenerative Rankine cycle with one open feedwater heater. Steam enters the turbine at 15 MPa and 600°C and the condenser at 10 kPa. Determine the thermal efficiency of the cycle, and plot it against extraction pressures of 12.5, 10, 7, 5, 2, 1, 0.5, 0.1, and 0.05 MPa, and discuss the results.

10–110 [EES] Using EES (or other) software, investigate the effect of the number of regeneration stages on the performance of an ideal regenerative Rankine cycle. Steam enters the turbine at 15 MPa and 600°C and the condenser at 5 kPa. For each case, maintain about the same temperature difference between any two regeneration stages. Determine the thermal efficiency of the cycle, and plot it against the number of regeneration stages for 1, 2, 3, 4, 5, 6, 8, and 10 regeneration stages.

Fundamentals of Engineering (FE) Exam Problems

10–111 Consider a steady-flow Carnot cycle with water as the working fluid executed under the saturation dome between the pressure limits of 8 MPa and 20 kPa. Water changes from saturated liquid to saturated vapor during the heat addition process. The net work output of this cycle is

(*a*) 494 kJ/kg   (*b*) 975 kJ/kg   (*c*) 596 kJ/kg
(*d*) 845 kJ/kg   (*e*) 1148 kJ/kg

10–112 A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 3 MPa, with a turbine inlet temperature of 600°C. Disregarding the pump work, the cycle efficiency is

(*a*) 24 percent   (*b*) 37 percent   (*c*) 52 percent
(*d*) 63 percent   (*e*) 71 percent

10–113 A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 5 MPa, with a turbine inlet temperature of 600°C. The mass fraction of steam that condenses at the turbine exit is

(*a*) 6 percent   (*b*) 9 percent   (*c*) 12 percent
(*d*) 15 percent   (*e*) 18 percent

10–114 A steam power plant operates on the simple ideal Rankine cycle between the pressure limits of 10 kPa and 10 MPa, with a turbine inlet temperature of 600°C. The rate of heat transfer in the boiler is 800 kJ/s. Disregarding the pump work, the power output of this plant is

(*a*) 243 kW   (*b*) 284 kW   (*c*) 508 kW
(*d*) 335 kW   (*e*) 800 kW

10–115 Consider a combined gas-steam power plant. Water for the steam cycle is heated in a well-insulated heat exchanger by the exhaust gases that enter at 800 K at a rate of 60 kg/s and leave at 400 K. Water enters the heat exchanger at 200°C and 8 MPa and leaves at 350°C and 8 MPa. If the exhaust gases are treated as air with constant specific heats at room temperature, the mass flow rate of water through the heat exchanger becomes

(*a*) 11 kg/s   (*b*) 24 kg/s   (*c*) 46 kg/s
(*d*) 53 kg/s   (*e*) 60 kg/s

10–116 An ideal reheat Rankine cycle operates between the pressure limits of 10 kPa and 8 MPa, with reheat occurring at 4 MPa. The temperature of steam at the inlets of both turbines is 500°C, and the enthalpy of steam is 3185 kJ/kg at the exit of the high-pressure turbine, and 2247 kJ/kg at the exit of the low-pressure turbine. Disregarding the pump work, the cycle efficiency is

(*a*) 29 percent   (*b*) 32 percent   (*c*) 36 percent
(*d*) 41 percent   (*e*) 49 percent

10–117 Pressurized feedwater in a steam power plant is to be heated in an ideal open feedwater heater that operates at a pressure of 0.5 MPa with steam extracted from the turbine. If the enthalpy of feedwater is 252 kJ/kg and the enthalpy of extracted steam is 2665 kJ/kg, the mass fraction of steam extracted from the turbine is

(*a*) 4 percent   (*b*) 10 percent   (*c*) 16 percent
(*d*) 27 percent   (*e*) 12 percent

10–118 Consider a steam power plant that operates on the regenerative Rankine cycle with one open feedwater heater. The enthalpy of the steam is 3374 kJ/kg at the turbine inlet, 2797 kJ/kg at the location of bleeding, and 2346 kJ/kg at the

turbine exit. The net power output of the plant is 120 MW, and the fraction of steam bled off the turbine for regeneration is 0.172. If the pump work is negligible, the mass flow rate of steam at the turbine inlet is

(*a*) 117 kg/s   (*b*) 126 kg/s   (*c*) 219 kg/s
(*d*) 268 kg/s   (*e*) 679 kg/s

10–119 Consider a simple ideal Rankine cycle. If the condenser pressure is lowered while keeping turbine inlet state the same,
(*a*) the turbine work output will decrease.
(*b*) the amount of heat rejected will decrease.
(*c*) the cycle efficiency will decrease.
(*d*) the moisture content at turbine exit will decrease.
(*e*) the pump work input will decrease.

10–120 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. If the steam is superheated to a higher temperature,
(*a*) the turbine work output will decrease.
(*b*) the amount of heat rejected will decrease.
(*c*) the cycle efficiency will decrease.
(*d*) the moisture content at turbine exit will decrease.
(*e*) the amount of heat input will decrease.

10–121 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. If the cycle is modified with reheating,
(*a*) the turbine work output will decrease.
(*b*) the amount of heat rejected will decrease.
(*c*) the pump work input will decrease.
(*d*) the moisture content at turbine exit will decrease.
(*e*) the amount of heat input will decrease.

10–122 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. If the cycle is modified with regeneration that involves one open feedwater heater (select the correct statement per unit mass of steam flowing through the boiler),
(*a*) the turbine work output will decrease.
(*b*) the amount of heat rejected will increase.
(*c*) the cycle thermal efficiency will decrease.
(*d*) the quality of steam at turbine exit will decrease.
(*e*) the amount of heat input will increase.

10–123 Consider a cogeneration power plant modified with regeneration. Steam enters the turbine at 6 MPa and 450°C at a rate of 20 kg/s and expands to a pressure of 0.4 MPa. At this pressure, 60 percent of the steam is extracted from the turbine, and the remainder expands to a pressure of 10 kPa. Part of the extracted steam is used to heat feedwater in an open feedwater heater. The rest of the extracted steam is used for process heating and leaves the process heater as a saturated liquid at 0.4 MPa. It is subsequently mixed with the feedwater leaving the feedwater heater, and the mixture is pumped to the boiler pressure. The steam in the condenser

is cooled and condensed by the cooling water from a nearby river, which enters the adiabatic condenser at a rate of 463 kg/s.

1. The total power output of the turbine is

(*a*) 17.0 MW   (*b*) 8.4 MW   (*c*) 12.2 MW
(*d*) 20.0 MW   (*e*) 3.4 MW

2. The temperature rise of the cooling water from the river in the condenser is

(*a*) 8.0°C   (*b*) 5.2°C   (*c*) 9.6°C
(*d*) 12.9°C   (*e*) 16.2°C

3. The mass flow rate of steam through the process heater is

(*a*) 1.6 kg/s   (*b*) 3.8 kg/s   (*c*) 5.2 kg/s
(*d*) 7.6 kg/s   (*e*) 10.4 kg/s

4. The rate of heat supply from the process heater per unit mass of steam passing through it is

(*a*) 246 kJ/kg   (*b*) 893 kJ/kg   (*c*) 1344 kJ/kg
(*d*) 1891 kJ/kg   (*e*) 2060 kJ/kg

5. The rate of heat transfer to the steam in the boiler is

(*a*) 26.0 MJ/s   (*b*) 53.8 MJ/s   (*c*) 39.5 MJ/s
(*d*) 62.8 MJ/s   (*e*) 125.4 MJ/s

FIGURE P10–123

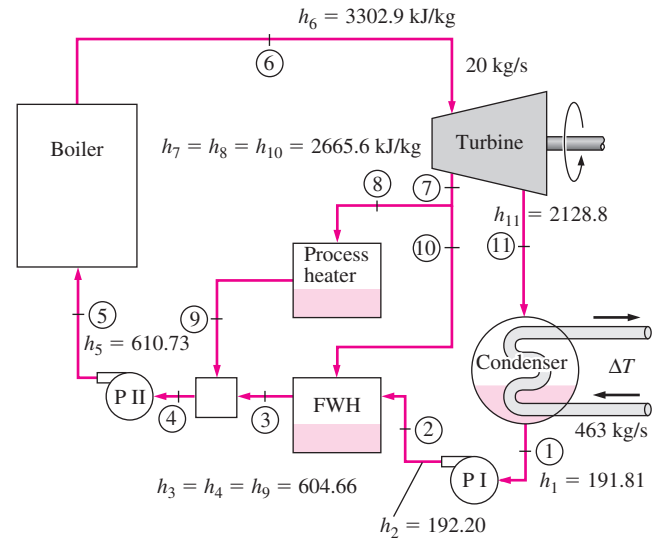### Design and Essay Problems

10–124 Design a steam power cycle that can achieve a cycle thermal efficiency of at least 40 percent under the conditions that all turbines have isentropic efficiencies of 85 percent and all pumps have isentropic efficiencies of 60 percent. Prepare

an engineering report describing your design. Your design report must include, but is not limited to, the following:

(*a*) Discussion of various cycles attempted to meet the goal as well as the positive and negative aspects of your design.

(*b*) System figures and *T-s* diagrams with labeled states and temperature, pressure, enthalpy, and entropy information for your design.

(*c*) Sample calculations.

10–125 Contact your power company and obtain information on the thermodynamic aspects of their most recently built power plant. If it is a conventional power plant, find out why it is preferred over a highly efficient combined power plant.

10–126 Several geothermal power plants are in operation in the United States and more are being built since the heat source of a geothermal plant is hot geothermal water, which is "free energy." An 8-MW geothermal power plant is being considered at a location where geothermal water at 160°C is available. Geothermal water is to serve as the heat source for a closed Rankine power cycle with refrigerant-134a as the working fluid. Specify suitable temperatures and pressures for the cycle, and determine the thermal efficiency of the cycle. Justify your selections.

10–127 A 10-MW geothermal power plant is being considered at a site where geothermal water at 230°C is available. Geothermal water is to be flashed into a chamber to a lower pressure where part of the water evaporates. The liquid is returned to the ground while the vapor is used to drive the steam turbine. The pressures at the turbine inlet and the turbine exit are to remain above 200 kPa and 8 kPa, respectively. High-pressure flash chambers yield a small amount of steam with high exergy whereas lower-pressure flash chambers yield considerably more steam but at a lower exergy. By tryi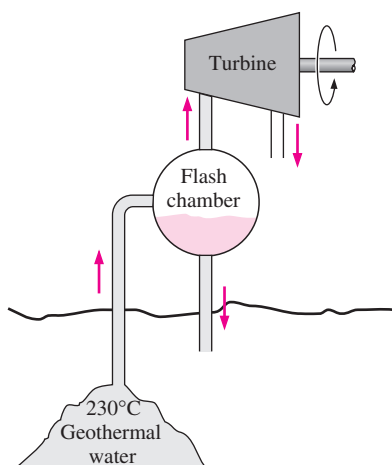ng several pressures, determine the optimum pressure of the flash chamber to maximize the power production per unit mass of geothermal water withdrawn. Also, determine the thermal efficiency for each case assuming 10 percent of the power produced is used to drive the pumps and other auxiliary equipment.

10–128 A natural gas–fired furnace in a textile plant is used to provide steam at 130°C. At times of high demand, the furnace supplies heat to the steam at a rate of 30 MJ/s. The plant also uses up to 6 MW of electrical power purchased from the local power company. The plant management is considering converting the existing process plant into a cogeneration plant to meet both their process-heat and power requirements. Your job is to come up with some designs. Designs based on a gas turbine or a steam turbine are to be considered. First decide whether a system based on a gas turbine or a steam turbine will best serve the purpose, considering the cost and the complexity. Then propose your design for the cogeneration plant complete with pressures and temperatures and the mass flow rates. Show that the proposed design meets the power and process-heat requirements of the plant.

10–129E A photographic equipment manufacturer uses a flow of 64,500 lbm/h of steam in its manufacturing process. Presently the spent steam at 3.8 psig and 224°F is exhausted to the atmosphere. Do the preliminary design of a system to use the energy in the waste steam economically. If electricity is produced, it can be generated about 8000 h/yr and its value is $0.05/kWh. If the energy is used for space heating, the value is also $0.05/kWh, but it can only be used about 3000 h/yr (only during the "heating season"). If the steam is condensed and the liquid H_2O is recycled through the process, its value is $0.50/100 gal. Make all assumptions as realistic as possible. Sketch the system you propose. Make a separate list of required components and their specifications (capacity, efficiency, etc.). The final result will be the calculated annual dollar value of the energy use plan (actually a *saving* because it will replace electricity or heat and/or water that would otherwise have to be purchased).

10–130 Design the condenser of a steam power plant that has a thermal efficiency of 40 percent and generates 10 MW of net electric power. Steam enters the condenser as saturated vapor at 10 kPa, and it is to be condensed outside horizontal tubes through which cooling water from a nearby river flows. The temperature rise of the cooling water is limited to 8°C, and the velocity of the cooling water in the pipes is limited to 6 m/s to keep the pressure drop at an acceptable level. From prior experience, the average heat flux based on the outer surface of the tubes can be taken to be 12,000 W/m². Specify the pipe diameter, total pipe length, and the arrangement of the pipes to minimize the condenser volume.

10–131 Water-cooled steam condensers are commonly used in steam power plants. Obtain information about water-cooled steam condensers by doing a literature search on the topic and

FIGURE P10–127

also by contacting some condenser manufacturers. In a report, describe the various types, the way they are designed, the limitation on each type, and the selection criteria.

10–132 Steam boilers have long been used to provide process heat as well as to generate power. Write an essay on the history of steam boilers and the evolution of modern supercritical steam power plants. What was the role of the American Society of Mechanical Engineers in this development?

10–133 The technology for power generation using geothermal energy is well established, and numerous geothermal power plants throughout the world are currently generating electricity economically. Binary geothermal plants utilize a volatile secondary fluid such as isobutane, n-pentane, and R-114 in a closed loop. Consider a binary geothermal plant with R-114 as the working fluid that is flowing at a rate of 600 kg/s. The R-114 is vaporized in a boiler at 115°C by the geothermal fluid that enters at 165°C, and is condensed at 30°C outside the tubes by cooling water that enters the tubes at 18°C. Based on prior experience, the average heat flux based on the outer surface of the tubes can be taken to be 4600 W/m^2. The enthalpy of vaporization of R-114 at 30°C is $h_{fg} = 121.5$ kJ/kg.

Specify (*a*) the length, diameter, and number of tubes and their arrangement in the condenser to minimize overall volume of the condenser; (*b*) the mass flow rate of cooling water; and (*c*) the flow rate of make-up water needed if a cooling tower is used to reject the waste heat from the cooling water. The liquid velocity is to remain under 6 m/s and the length of the tubes is limited to 8 m.