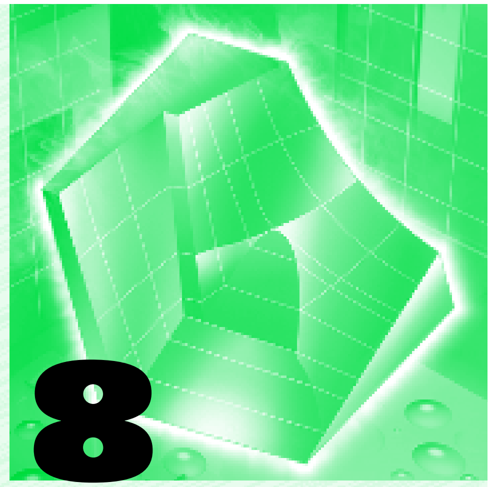


C H A P T E R 8

Vapor Power Systems

ENGINEERING CONTEXT An important engineering goal is to devise systems that accomplish desired types of energy conversion. The present chapter and the next are concerned with several types of power-generating systems, each of which produces a net power output from a fossil fuel, nuclear, or solar input. In these chapters, we describe some of the practical arrangements employed for power production and illustrate how such power plants can be modeled thermodynamically. The discussion is organized into three main areas of application: vapor power plants, gas turbine power plants, and internal combustion engines. These power systems, together with hydroelectric power plants, produce virtually all of the electrical and mechanical power used worldwide. The **objective** of the present chapter is to study *vapor* power plants in which the *working fluid* is alternately vaporized and condensed. Chapter 9 is concerned with gas turbines and internal combustion engines in which the working fluid remains a gas.

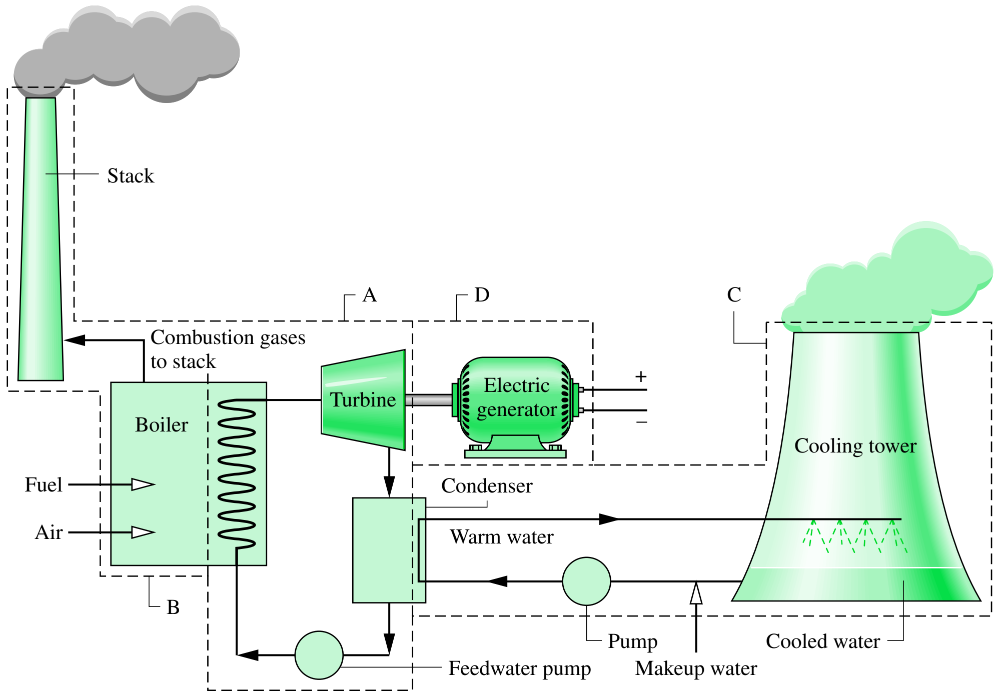
◀ **chapter objective**

8.1 Modeling Vapor Power Systems

The processes taking place in power-generating systems are sufficiently complicated that idealizations are required to develop thermodynamic models. Such modeling is an important initial step in engineering design. Although the study of simplified models generally leads only to qualitative conclusions about the performance of the corresponding actual devices, models often allow deductions about how changes in major operating parameters affect actual performance. They also provide relatively simple settings in which to discuss the functions and benefits of features intended to improve overall performance.

The vast majority of electrical generating plants are variations of vapor power plants in which water is the working fluid. The basic components of a simplified fossil-fuel vapor power plant are shown schematically in Fig. 8.1. To facilitate thermodynamic analysis, the overall plant can be broken down into the four major subsystems identified by the letters A through D on the diagram. The focus of our considerations in this chapter is subsystem A, where the important energy conversion from *heat to work* occurs. But first, let us briefly consider the other subsystems.

The function of subsystem B is to supply the energy required to vaporize the water passing through the boiler. In fossil-fuel plants, this is accomplished by heat transfer *to* the working fluid passing through tubes and drums in the boiler *from* the hot gases produced by the combustion of a fossil fuel. In nuclear plants, the origin of the energy is a controlled nuclear reaction taking place in an isolated reactor building. Pressurized water, a liquid metal,



▲ **Figure 8.1** Components of a simple vapor power plant.

or a gas such as helium can be used to transfer energy released in the nuclear reaction to the working fluid in specially designed heat exchangers. Solar power plants have receivers for concentrating and collecting solar radiation to vaporize the working fluid. Regardless of the energy source, the vapor produced in the boiler passes through a turbine, where it expands to a lower pressure. The shaft of the turbine is connected to an electric generator (subsystem D). The vapor leaving the turbine passes through the condenser, where it condenses on the outside of tubes carrying cooling water. The cooling water circuit comprises subsystem C. For the plant shown, the cooling water is sent to a cooling tower, where energy taken up in the condenser is rejected to the atmosphere. The cooling water is then recirculated through the condenser.

Concern for the environment and safety considerations govern what is allowable in the interactions between subsystems B and C and their surroundings. One of the major difficulties in finding a site for a vapor power plant is access to sufficient quantities of cooling water. For this reason and to minimize *thermal pollution* effects, most power plants now employ cooling towers. In addition to the question of cooling water, the safe processing and delivery of fuel, the control of pollutant discharges, and the disposal of wastes are issues that must be dealt with in both fossil-fueled and nuclear-fueled plants to ensure safety and operation with an acceptable level of environmental impact. Solar power plants are generally regarded as nonpolluting and safe but as yet are not widely used.

Returning now to subsystem A of Fig. 8.1, observe that each unit of mass periodically undergoes a thermodynamic cycle as the working fluid circulates through the series of four interconnected components. Accordingly, several concepts related to thermodynamic *power cycles* introduced in previous chapters are important for the present discussions. You will recall that the conservation of energy principle requires that the net work developed by a power cycle equals the net heat added. An important deduction from the second law is that the thermal efficiency, which indicates the extent to which the heat added is converted to a net work output, must be less than 100%. Previous discussions also have indicated that improved thermodynamic performance accompanies the reduction of irreversibilities. The extent to which irreversibilities can be reduced in power-generating systems depends on thermodynamic, economic, and other factors, however.

8.2 Analyzing Vapor Power Systems—Rankine Cycle

All of the fundamentals required for the thermodynamic analysis of power-generating systems already have been introduced. They include the conservation of mass and conservation of energy principles, the second law of thermodynamics, and thermodynamic data. These principles apply to individual plant components such as turbines, pumps, and heat exchangers as well as to the most complicated overall power plants. The object of this section is to introduce the ***Rankine cycle,*** which is a thermodynamic cycle that models the subsystem labeled A on Fig. 8.1. The presentation begins by considering the thermodynamic analysis of this subsystem.

Rankine cycle

▶ 8.2.1 Evaluating Principal Work and Heat Transfers

The principal work and heat transfers of subsystem A are illustrated in Fig. 8.2. In subsequent discussions, these energy transfers are taken to be *positive in the directions of the arrows.* The unavoidable stray heat transfer that takes place between the plant components and their surroundings is neglected here for simplicity. Kinetic and potential energy changes are also ignored. Each component is regarded as operating at steady state. Using the conservation of mass and conservation of energy principles together with these idealizations, we develop expressions for the energy transfers shown on Fig. 8.2 beginning at state 1 and proceeding through each component in turn.

METHODOLOGY UPDATE

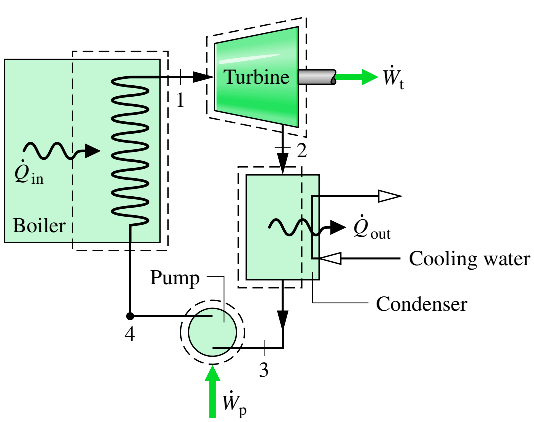
When analyzing vapor power cycles, we take energy transfers as positive in the directions of arrows on system schematics and write the energy balance accordingly.

TURBINE. Vapor from the boiler at state 1, having an elevated temperature and pressure, expands through the turbine to produce work and then is discharged to the condenser at state 2 with relatively low pressure. Neglecting heat transfer with the surroundings, the mass and energy rate balances for a control volume around the turbine reduce at steady state to give

$$0 = \cancelto{0}{\dot{Q}_{cv}} - \dot{W}_t + \dot{m}\left[h_1 - h_2 + \cancelto{0}{\frac{V_1^2 - V_2^2}{2}} + \cancelto{0}{g(z_1 - z_2)}\right]$$

or

$$\frac{\dot{W}_t}{\dot{m}} = h_1 - h_2 \tag{8.1}$$



◀ **Figure 8.2** Principal work and heat transfers of subsystem A.

where $\dot{m}$ denotes the mass flow rate of the working fluid, and $\dot{W}_t/\dot{m}$ is the rate at which work is developed per unit of mass of steam passing through the turbine. As noted above, kinetic and potential energy changes are ignored.

CONDENSER. In the condenser there is heat transfer from the vapor to cooling water flowing in a separate stream. The vapor condenses and the temperature of the cooling water increases. At steady state, mass and energy rate balances for a control volume enclosing the condensing side of the heat exchanger give

$$\frac{\dot{Q}_{\text{out}}}{\dot{m}} = h_2 - h_3 \tag{8.2}$$

where $\dot{Q}_{\text{out}}/\dot{m}$ is the rate at which energy is transferred by heat *from* the working fluid to the cooling water per unit mass of working fluid passing through the condenser. This energy transfer is positive in the direction of the arrow on Fig. 8.2.

PUMP. The liquid condensate leaving the condenser at 3 is pumped from the condenser into the higher pressure boiler. Taking a control volume around the pump and assuming no heat transfer with the surroundings, mass and energy rate balances give

$$\frac{\dot{W}_p}{\dot{m}} = h_4 - h_3 \tag{8.3}$$

where $\dot{W}_p/\dot{m}$ is the rate of power *input* per unit of mass passing through the pump. This energy transfer is positive in the direction of the arrow on Fig. 8.2.

feedwater

BOILER. The working fluid completes a cycle as the liquid leaving the pump at 4, called the boiler ***feedwater,*** is heated to saturation and evaporated in the boiler. Taking a control volume enclosing the boiler tubes and drums carrying the feedwater from state 4 to state 1, mass and energy rate balances give

$$\frac{\dot{Q}_{\text{in}}}{\dot{m}} = h_1 - h_4 \tag{8.4}$$

where $\dot{Q}_{\text{in}}/\dot{m}$ is the rate of heat transfer from the energy source into the working fluid per unit mass passing through the boiler.

PERFORMANCE PARAMETERS. The thermal efficiency gauges the extent to which the energy input to the working fluid passing through the boiler is converted to the *net* work output. Using the quantities and expressions just introduced, the ***thermal efficiency*** of the power cycle of Fig. 8.2 is

thermal efficiency

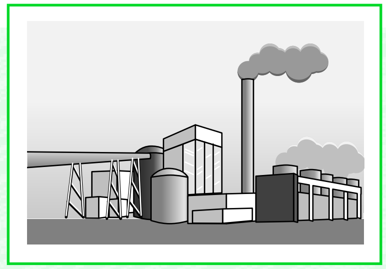
$$\eta = \frac{\dot{W}_t/\dot{m} - \dot{W}_p/\dot{m}}{\dot{Q}_{\text{in}}/\dot{m}} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} \tag{8.5a}$$

The net work output equals the net heat input. Thus, the thermal efficiency can be expressed alternatively as

$$\begin{aligned}\eta &= \frac{\dot{Q}_{\text{in}}/\dot{m} - \dot{Q}_{\text{out}}/\dot{m}}{\dot{Q}_{\text{in}}/\dot{m}} = 1 - \frac{\dot{Q}_{\text{out}}/\dot{m}}{\dot{Q}_{\text{in}}/\dot{m}} \\ &= 1 - \frac{(h_2 - h_3)}{(h_1 - h_4)} \end{aligned} \tag{8.5b}$$

Thermodynamics in the News...

Cleaner Is Better



The United States relies heavily upon abundant coal reserves to generate electric power, but these systems require a great deal of clean-up, experts say. Awareness of the health and environmental impacts of coal have led to increasingly-stringent regulations on coal-burning power plants. As a result, the search for new *clean-coal* technologies has intensified.

According to industry sources, controlling particulate emissions and safely disposing of millions of tons of coal waste once were the main concerns. Sulfur dioxide removal then became an issue due to concern over acid rain. More recently, nitric oxide (NO_X), mercury, and fine particle (less than 3 microns) emissions were recognized as especially harmful. Strides have been made in developing more effective sulfur dioxide scrubbers and particulate capture devices, but stricter environmental standards demand new approaches.

One promising technology is fluidized bed combustion, where a powdered coal-limestone mixture churns in air to enhance the burning. The limestone removes some sulfur *during* combustion rather than waiting to remove the sulfur *after* combustion, as in conventional boilers. Nitric oxide formation is also less because of the relatively low temperatures of fluidized bed combustion. Another innovation is the integrated gasification combined-cycle plant, or IGCC, that promises to be cleaner and have higher thermal efficiency than conventional plants. In an IGCC, coal is converted to a cleaner-burning combustible gas that is used in a power plant combining gas and steam turbines.

Another parameter used to describe power plant performance is the ***back work ratio,*** or bwr, defined as the ratio of the pump work input to the work developed by the turbine. With Eqs. 8.1 and 8.3, the back work ratio for the power cycle of Fig. 8.2 is

back work ratio

$$\text{bwr} = \frac{\dot{W}_p/\dot{m}}{\dot{W}_t/\dot{m}} = \frac{(h_4 - h_3)}{(h_1 - h_2)} \tag{8.6}$$

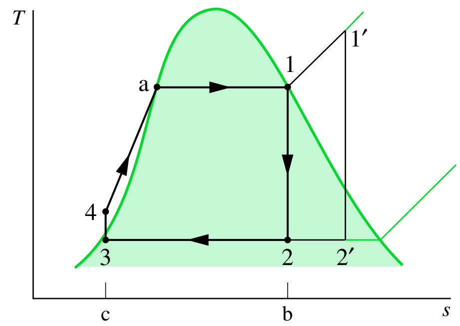
Examples to follow illustrate that the change in specific enthalpy for the expansion of vapor through the turbine is normally many times greater than the increase in enthalpy for the liquid passing through the pump. Hence, the back work ratio is characteristically quite low for vapor power plants.

Provided states 1 through 4 are fixed, Eqs. 8.1 through 8.6 can be applied to determine the thermodynamic performance of a simple vapor power plant. Since these equations have been developed from mass and energy rate balances, they apply equally for actual performance when irreversibilities are present and for idealized performance in the absence of such effects. It might be surmised that the irreversibilities of the various power plant components can affect overall performance, and this is the case. Even so, it is instructive to consider an idealized cycle in which irreversibilities are assumed absent, for such a cycle establishes an *upper limit* on the performance of the Rankine cycle. The ideal cycle also provides a simple setting in which to study various aspects of vapor power plant performance.

► 8.2.2 Ideal Rankine Cycle

If the working fluid passes through the various components of the simple vapor power cycle without irreversibilities, frictional pressure drops would be absent from the boiler and condenser, and the working fluid would flow through these components at constant pressure. Also, in the absence of irreversibilities and heat transfer with the surroundings, the processes through the turbine and pump would be isentropic. A cycle adhering to these idealizations is the ***ideal Rankine cycle*** shown in Fig. 8.3.

ideal Rankine cycle



◀ **Figure 8.3** Temperature–entropy diagram of the ideal Rankine cycle.

Referring to Fig. 8.3, we see that the working fluid undergoes the following series of internally reversible processes:

Process 1–2: Isentropic expansion of the working fluid through the turbine from saturated vapor at state 1 to the condenser pressure.

Process 2–3: Heat transfer *from* the working fluid as it flows at constant pressure through the condenser with saturated liquid at state 3.

Process 3–4: Isentropic compression in the pump to state 4 in the compressed liquid region.

Process 4–1: Heat transfer *to* the working fluid as it flows at constant pressure through the boiler to complete the cycle.

The ideal Rankine cycle also includes the possibility of superheating the vapor, as in cycle 1′–2′–3–4–1′. The importance of superheating is discussed in Sec. 8.3.

Since the ideal Rankine cycle consists of internally reversible processes, areas under the process lines of Fig. 8.3 can be interpreted as heat transfers per unit of mass flowing. Applying Eq. 6.51, area 1–b–c–4–a–1 represents the heat transfer to the working fluid passing through the boiler and area 2–b–c–3–2, is the heat transfer from the working fluid passing through the condenser, each per unit of mass flowing. The enclosed area 1–2–3–4–a–1 can be interpreted as the net heat input or, equivalently, the net work output, each per unit of mass flowing.

Because the pump is idealized as operating without irreversibilities, Eq. 6.53b can be invoked as an alternative to Eq. 8.3 for evaluating the pump work. That is

$$\left(\frac{\dot{W}_\text{p}}{\dot{m}}\right)_{\substack{\text{int}\\\text{rev}}} = \int_3^4 v\, dp \tag{8.7a}$$

where the minus sign has been dropped for consistency with the positive value for pump work in Eq. 8.3. The subscript "int rev" has been retained as a reminder that this expression is restricted to an internally reversible process through the pump. No such designation is required by Eq. 8.3, however, because it expresses the conservation of mass and energy principles and thus is not restricted to processes that are internally reversible.

METHODOLOGY UPDATE

For cycles, we modify the problem-solving methodology: The **Analysis** begins with a systematic evaluation of required property data at each numbered state. This reinforces what we know about the components, since given information and assumptions are required to fix the states.

Evaluation of the integral of Eq. 8.7a requires a relationship between the specific volume and pressure for the process. Because the specific volume of the liquid normally varies only slightly as the liquid flows from the inlet to the exit of the pump, a plausible approximation to the value of the integral can be had by taking the specific volume at the pump inlet, v_3, as constant for the process. Then

$$\left(\frac{\dot{W}_\text{p}}{\dot{m}}\right)_{\substack{\text{int}\\\text{rev}}} \approx v_3(p_4 - p_3) \tag{8.7b}$$

The next example illustrates the analysis of an ideal Rankine cycle.

EXAMPLE 8.1 Ideal Rankine Cycle

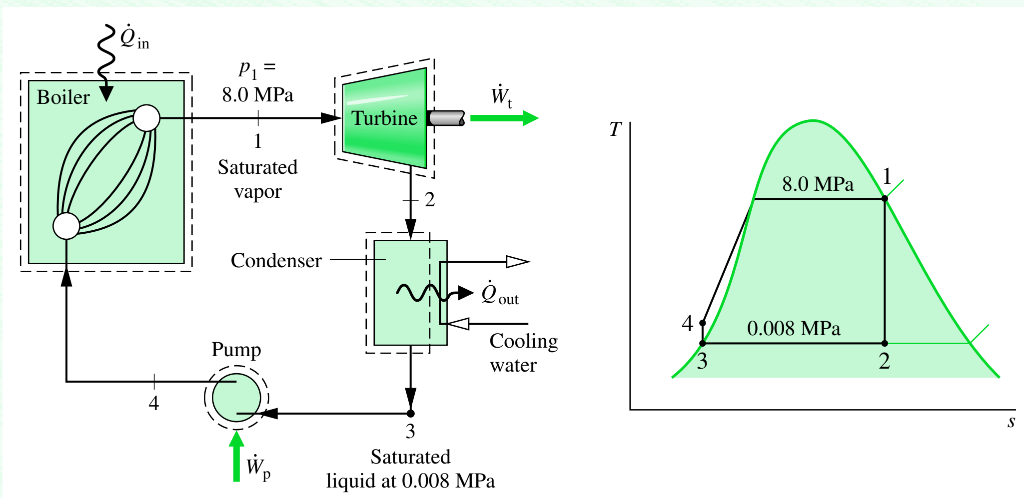
Steam is the working fluid in an ideal Rankine cycle. Saturated vapor enters the turbine at 8.0 MPa and saturated liquid exits the condenser at a pressure of 0.008 MPa. The *net* power output of the cycle is 100 MW. Determine for the cycle **(a)** the thermal efficiency, **(b)** the back work ratio, **(c)** the mass flow rate of the steam, in kg/h, **(d)** the rate of heat transfer, $\dot{Q}_{in}$, into the working fluid as it passes through the boiler, in MW, **(e)** the rate of heat transfer, $\dot{Q}_{out}$, from the condensing steam as it passes through the condenser, in MW, **(f)** the mass flow rate of the condenser cooling water, in kg/h, if cooling water enters the condenser at 15°C and exits at 35°C.

SOLUTION

Known: An ideal Rankine cycle operates with steam as the working fluid. The boiler and condenser pressures are specified, and the net power output is given.

Find: Determine the thermal efficiency, the back work ratio, the mass flow rate of the steam, in kg/h, the rate of heat transfer to the working fluid as it passes through the boiler, in MW, the rate of heat transfer from the condensing steam as it passes through the condenser, in MW, the mass flow rate of the condenser cooling water, which enters at 15°C and exits at 35°C.

Schematic and Given Data:



◀ **Figure E8.1**

Assumptions:

1. Each component of the cycle is analyzed as a control volume at steady state. The control volumes are shown on the accompanying sketch by dashed lines.
2. All processes of the working fluid are internally reversible.
3. The turbine and pump operate adiabatically.
4. Kinetic and potential energy effects are negligible.
5. Saturated vapor enters the turbine. Condensate exits the condenser as saturated liquid.

Analysis: To begin the analysis, we fix each of the principal states located on the accompanying schematic and *T–s* diagrams.
1 Starting at the inlet to the turbine, the pressure is 8.0 MPa and the steam is a saturated vapor, so from Table A-3, $h_1 = 2758.0$ kJ/kg and $s_1 = 5.7432$ kJ/kg · K.

State 2 is fixed by $p_2 = 0.008$ MPa and the fact that the specific entropy is constant for the adiabatic, internally reversible expansion through the turbine. Using saturated liquid and saturated vapor data from Table A-3, we find that the quality at state 2 is

$$x_2 = \frac{s_2 - s_f}{s_g - s_f} = \frac{5.7432 - 0.5926}{7.6361} = 0.6745$$

The enthalpy is then

$$h_2 = h_f + x_2 h_{fg} = 173.88 + (0.6745)2403.1$$
$$= 1794.8 \text{ kJ/kg}$$

State 3 is saturated liquid at 0.008 MPa, so $h_3 = 173.88$ kJ/kg.

State 4 is fixed by the boiler pressure p_4 and the specific entropy $s_4 = s_3$. The specific enthalpy h_4 can be found by interpolation in the compressed liquid tables. However, because compressed liquid data are relatively sparse, it is more convenient to solve Eq. 8.3 for h_4, using Eq. 8.7b to approximate the pump work. With this approach

$$h_4 = h_3 + \dot{W}_p/\dot{m} = h_3 + v_3(p_4 - p_3)$$

By inserting property values from Table A-3

$$h_4 = 173.88 \text{ kJ/kg} + (1.0084 \times 10^{-3} \text{ m}^3/\text{kg})(8.0 - 0.008)\text{MPa}\left|\frac{10^6 \text{ N/m}^2}{1 \text{ MPa}}\right|\left|\frac{1 \text{ kJ}}{10^3 \text{ N}\cdot\text{m}}\right|$$
$$= 173.88 + 8.06 = 181.94 \text{ kJ/kg}$$

(a) The *net* power developed by the cycle is

$$\dot{W}_{cycle} = \dot{W}_t - \dot{W}_p$$

Mass and energy rate balances for control volumes around the turbine and pump give, respectively

$$\frac{\dot{W}_t}{\dot{m}} = h_1 - h_2 \qquad \text{and} \qquad \frac{\dot{W}_p}{\dot{m}} = h_4 - h_3$$

where $\dot{m}$ is the mass flow rate of the steam. The rate of heat transfer to the working fluid as it passes through the boiler is determined using mass and energy rate balances as

$$\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4$$

The thermal efficiency is then

$$\eta = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_{in}} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$
$$= \frac{[(2758.0 - 1794.8) - (181.94 - 173.88)] \text{ kJ/kg}}{(2758.0 - 181.94) \text{ kJ/kg}}$$
$$= 0.371\ (37.1\%)$$

(b) The back work ratio is

2 $$\text{bwr} = \frac{\dot{W}_p}{\dot{W}_t} = \frac{h_4 - h_3}{h_1 - h_2} = \frac{(181.94 - 173.88) \text{ kJ/kg}}{(2758.0 - 1794.8) \text{ kJ/kg}}$$
$$= \frac{8.06}{963.2} = 8.37 \times 10^{-3}\ (0.84\%)$$

(c) The mass flow rate of the steam can be obtained from the expression for the net power given in part (a). Thus

$$\dot{m} = \frac{\dot{W}_{cycle}}{(h_1 - h_2) - (h_4 - h_3)}$$
$$= \frac{(100 \text{ MW})|10^3 \text{ kW/MW}||3600 \text{ s/h}|}{(963.2 - 8.06) \text{ kJ/kg}}$$
$$= 3.77 \times 10^5 \text{ kg/h}$$

(d) With the expression for $\dot{Q}_{in}$ from part (a) and previously determined specific enthalpy values

$$\begin{aligned}\dot{Q}_{in} &= \dot{m}(h_1 - h_4)\\ &= \frac{(3.77 \times 10^5 \text{ kg/h})(2758.0 - 181.94) \text{ kJ/kg}}{|3600 \text{ s/h}|\,|10^3 \text{ kW/MW}|}\\ &= 269.77 \text{ MW}\end{aligned}$$

(e) Mass and energy rate balances applied to a control volume enclosing the steam side of the condenser give

$$\begin{aligned}\dot{Q}_{out} &= \dot{m}(h_2 - h_3)\\ &= \frac{(3.77 \times 10^5 \text{ kg/h})(1794.8 - 173.88) \text{ kJ/kg}}{|3600 \text{ s/h}|\,|10^3 \text{ kW/MW}|}\\ &= 169.75 \text{ MW}\end{aligned}$$

❸ Note that the ratio of $\dot{Q}_{out}$ to $\dot{Q}_{in}$ is 0.629 (62.9%).

Alternatively, $\dot{Q}_{out}$ can be determined from an energy rate balance on the *overall* vapor power plant. At steady state, the net power developed equals the net rate of heat transfer to the plant

$$\dot{W}_{cycle} = \dot{Q}_{in} - \dot{Q}_{out}$$

Rearranging this expression and inserting values

$$\dot{Q}_{out} = \dot{Q}_{in} - \dot{W}_{cycle} = 269.77 \text{ MW} - 100 \text{ MW} = 169.77 \text{ MW}$$

The slight difference from the above value is due to round-off.

(f) Taking a control volume around the condenser, the mass and energy rate balances give at steady state

$$0 = \cancelto{0}{\dot{Q}_{cv}} - \cancelto{0}{\dot{W}_{cv}} + \dot{m}_{cw}(h_{cw,in} - h_{cw,out}) + \dot{m}(h_2 - h_3)$$

where $\dot{m}_{cw}$ is the mass flow rate of the cooling water. Solving for $\dot{m}_{cw}$

$$\dot{m}_{cw} = \frac{\dot{m}(h_2 - h_3)}{(h_{cw,out} - h_{cw,in})}$$

The numerator in this expression is evaluated in part (e). For the cooling water, $h \approx h_f(T)$, so with saturated liquid enthalpy values from Table A-2 at the entering and exiting temperatures of the cooling water

$$\dot{m}_{cw} = \frac{(169.75 \text{ MW})|10^3 \text{ kW/MW}|\,|3600 \text{ s/h}|}{(146.68 - 62.99) \text{ kJ/kg}} = 7.3 \times 10^6 \text{ kg/h}$$

❶ Note that a slightly revised problem-solving methodology is used in this example problem: We begin with a systematic evaluation of the specific enthalpy at each numbered state.

❷ Note that the back work ratio is relatively low for the Rankine cycle. In the present case, the work required to operate the pump is less than 1% of the turbine output.

❸ In this example, 62.9% of the energy added to the working fluid by heat transfer is subsequently discharged to the cooling water. Although considerable energy is carried away by the cooling water, its exergy is small because the water exits at a temperature only a few degrees greater than that of the surroundings. See Sec. 8.6 for further discussion.

▶ 8.2.3 Effects of Boiler and Condenser Pressures on the Rankine Cycle

In discussing Fig. 5.9 (Sec. 5.5.1), we observed that the thermal efficiency of power cycles tends to increase as the average temperature at which energy is added by heat transfer increases and/or the average temperature at which energy is rejected decreases. (For elaboration, see box.) Let us apply this idea to study the effects on performance of the ideal Rankine cycle of changes in the boiler and condenser pressures. Although these findings are obtained with reference to the ideal Rankine cycle, they also hold qualitatively for actual vapor power plants.

CONSIDERING THE EFFECT OF TEMPERATURE ON THERMAL EFFICIENCY

Since the ideal Rankine cycle consists entirely of internally reversible processes, an expression for thermal efficiency can be obtained in terms of *average* temperatures during the heat interaction processes. Let us begin the development of this expression by recalling that areas under the process lines of Fig. 8.3 can be interpreted as the heat transfer per unit of mass flowing through the respective components. For example, the total area 1–b–c–4–a–1 represents the heat transfer into the working fluid per unit of mass passing through the boiler. In symbols,

$$\left(\frac{\dot{Q}_{\text{in}}}{\dot{m}}\right)_{\substack{\text{int}\\\text{rev}}} = \int_4^1 T\,ds = \text{area } 1\text{–b–c–4–a–1}$$

The integral can be written in terms of an average temperature of heat addition, $\overline{T}_{\text{in}}$, as follows:

$$\left(\frac{\dot{Q}_{\text{in}}}{\dot{m}}\right)_{\substack{\text{int}\\\text{rev}}} = \overline{T}_{\text{in}}(s_1 - s_4)$$

where the overbar denotes *average*. Similarly, area 2–b–c–3–2 represents the heat transfer from the condensing steam per unit of mass passing through the condenser

$$\begin{aligned}\left(\frac{\dot{Q}_{\text{out}}}{\dot{m}}\right)_{\substack{\text{int}\\\text{rev}}} &= T_{\text{out}}(s_2 - s_3) = \text{area } 2\text{–b–c–3–2}\\ &= T_{\text{out}}(s_1 - s_4)\end{aligned}$$

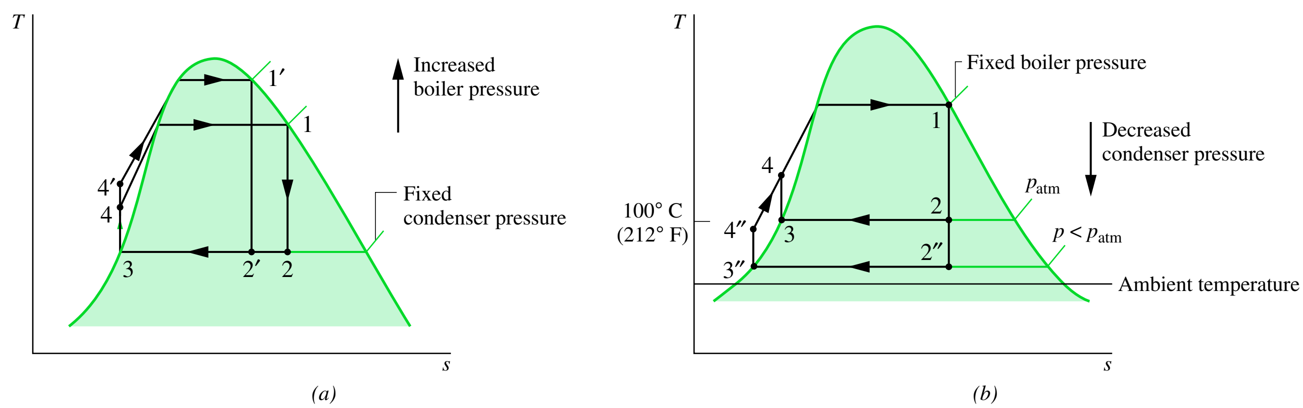
where T_{out} denotes the temperature on the steam side of the condenser of the ideal Rankine cycle pictured in Fig. 8.3. The thermal efficiency of the ideal Rankine cycle can be expressed in terms of these heat transfers as

$$\eta_{\text{ideal}} = 1 - \frac{(\dot{Q}_{\text{out}}/\dot{m})_{\substack{\text{int}\\\text{rev}}}}{(\dot{Q}_{\text{in}}/\dot{m})_{\substack{\text{int}\\\text{rev}}}} = 1 - \frac{T_{\text{out}}}{\overline{T}_{\text{in}}} \tag{8.8}$$

By the study of Eq. 8.8, we conclude that the thermal efficiency of the ideal cycle tends to increase as the average temperature at which energy is added by heat transfer increases and/or the temperature at which energy is rejected decreases. With similar reasoning, these conclusions can be shown to apply to the other ideal cycles considered in this chapter and the next.

Figure 8.4*a* shows two ideal cycles having the same condenser pressure but different boiler pressures. By inspection, the average temperature of heat addition is seen to be greater for the higher-pressure cycle 1′–2′–3′–4′–1′ than for cycle 1–2–3–4–1. It follows that increasing the boiler pressure of the ideal Rankine cycle tends to increase the thermal efficiency.

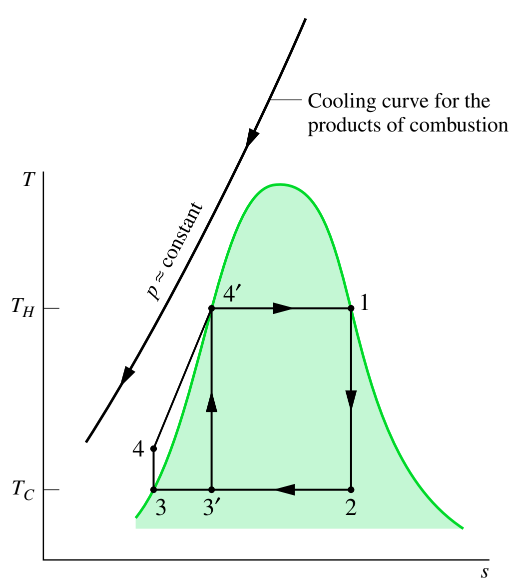
Figure 8.4*b* shows two cycles with the same boiler pressure but two different condenser pressures. One condenser operates at atmospheric pressure and the other at *less than* atmospheric pressure. The temperature of heat rejection for cycle 1–2–3–4–1 condensing at atmospheric pressure is 100°C (212°F). The temperature of heat rejection for the lower-pressure cycle 1–2″–3″–4″–1 is correspondingly lower, so this cycle has the greater thermal efficiency. It follows that decreasing the condenser pressure tends to increase the thermal efficiency.



▲ **Figure 8.4** Effects of varying operating pressures on the ideal Rankine cycle. (*a*) Effect of boiler pressure. (*b*) Effect of condenser pressure.

The lowest feasible condenser pressure is the saturation pressure corresponding to the ambient temperature, for this is the lowest possible temperature for heat rejection to the surroundings. The goal of maintaining the lowest practical turbine exhaust (condenser) pressure is a primary reason for including the condenser in a power plant. Liquid water at atmospheric pressure could be drawn into the boiler by a pump, and steam could be discharged directly to the atmosphere at the turbine exit. However, by including a condenser in which the steam side is operated at a pressure *below atmospheric,* the turbine has a lower-pressure region in which to discharge, resulting in a significant increase in net work and thermal efficiency. The addition of a condenser also allows the working fluid to flow in a closed loop. This arrangement permits continual circulation of the working fluid, so purified water that is less corrosive than tap water can be used economically.

COMPARISON WITH CARNOT CYCLE. Referring to Fig. 8.5, the ideal Rankine cycle 1–2–3–4–4′–1 has a lower thermal efficiency than the Carnot cycle 1–2–3′–4′–1 having the same maximum temperature T_H and minimum temperature T_C because the average temperature between 4 and 4′ is less than T_H. Despite the greater thermal efficiency of the Carnot



◀ **Figure 8.5** Illustration used to compare the ideal Rankine cycle with the Carnot cycle.

cycle, it has two shortcomings as a model for the simple vapor power cycle. First, the heat passing to the working fluid of a vapor power plant is usually obtained from hot products of combustion cooling at approximately constant pressure. To exploit fully the energy released on combustion, the hot products should be cooled as much as possible. The first portion of the heating process of the Rankine cycle shown in Fig. 8.5, Process 4–4′, is achieved by cooling the combustion products *below* the maximum temperature T_H. With the Carnot cycle, however, the combustion products would be cooled *at the most* to T_H. Thus, a smaller portion of the energy released on combustion would be used. The second shortcoming of the Carnot vapor power cycle involves the pumping process. Note that the state 3′ of Fig. 8.5 is a two-phase liquid–vapor mixture. Significant practical problems are encountered in developing pumps that handle two-phase mixtures, as would be required by Carnot cycle 1–2–3′–4′–1. It is far easier to condense the vapor completely and handle only liquid in the pump, as is done in the Rankine cycle. Pumping from 3 to 4 and constant-pressure heating without work from 4 to 4′ are processes that can be closely achieved in practice.

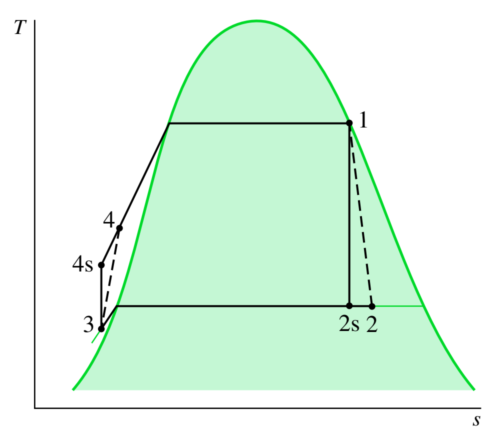
▶ 8.2.4 Principal Irreversibilities and Losses

Irreversibilities and losses are associated with each of the four subsystems shown in Fig. 8.1. Some of these effects have a more pronounced influence on performance than others. Let us consider the irreversibilities and losses associated with the Rankine cycle.

TURBINE. The principal irreversibility experienced by the working fluid is associated with the expansion through the turbine. Heat transfer from the turbine to the surroundings represents a loss, but since it is usually of secondary importance, this loss is ignored in subsequent discussions. As illustrated by Process 1–2 of Fig. 8.6, an actual adiabatic expansion through the turbine is accompanied by an increase in entropy. The work developed per unit of mass in this process is less than for the corresponding isentropic expansion 1–2s. The isentropic turbine efficiency η_t introduced in Sec. 6.8 allows the effect of irreversibilities within the turbine to be accounted for in terms of the actual and isentropic work amounts. Designating the states as in Fig. 8.6, the isentropic turbine efficiency is

$$\eta_t = \frac{(\dot{W}_t/\dot{m})}{(\dot{W}_t/\dot{m})_s} = \frac{h_1 - h_2}{h_1 - h_{2s}} \tag{8.9}$$

where the numerator is the actual work developed per unit of mass passing through the turbine and the denominator is the work for an isentropic expansion from the turbine inlet state



◀ **Figure 8.6** Temperature–entropy diagram showing the effects of turbine and pump irreversibilities.

to the turbine exhaust pressure. Irreversibilities within the turbine significantly reduce the net power output of the plant.

PUMP. The work input to the pump required to overcome frictional effects also reduces the net power output of the plant. In the absence of heat transfer to the surroundings, there would be an increase in entropy across the pump. Process 3–4 of Fig. 8.6 illustrates the actual pumping process. The work input for this process is *greater* than for the corresponding isentropic process 3–4s. The isentropic pump efficiency η_p introduced in Sec. 6.8 allows the effect of irreversibilities within the pump to be accounted for in terms of the actual and isentropic work amounts. Designating the states as in Fig. 8.6, the isentropic pump efficiency is

$$\eta_p = \frac{(\dot{W}_p/\dot{m})_s}{(\dot{W}_p/\dot{m})} = \frac{h_{4s} - h_3}{h_4 - h_3} \tag{8.10}$$

In this expression, the pump work for the isentropic process appears in the numerator. The actual pump work, being the larger quantity, is the denominator. Because the pump work is so much less than the turbine work, irreversibilities in the pump have a much smaller impact on the net work of the cycle than do irreversibilities in the turbine.

OTHER NONIDEALITIES. The turbine and pump irreversibilities mentioned above are *internal* irreversibilities experienced by the working fluid as it flows around the closed loop of the Rankine cycle. The most significant sources of irreversibility for a fossil-fueled vapor power plant, however, are associated with the combustion of the fuel and the subsequent heat transfer from the hot combustion products to the cycle working fluid. These effects occur in the surroundings of the subsystem labeled A on Fig. 8.1 and thus are *external* irreversibilities for the Rankine cycle. These irreversibilities are considered further in Sec. 8.6 and Chap. 13 using the exergy concept.

Another effect that occurs in the surroundings is the energy discharge to the cooling water as the working fluid condenses. Although considerable energy is carried away by the cooling water, its *utility* is extremely limited. For condensers in which steam condenses near the ambient temperature, the cooling water experiences a temperature rise of only a few degrees over the temperature of the surroundings in passing through the condenser and thus has limited usefulness. Accordingly, the significance of this loss is far less than suggested by the magnitude of the energy transferred to the cooling water. The utility of condenser cooling water is considered further in Sec. 8.6 using the exergy concept.

In addition to the foregoing, there are several other sources of nonideality. For example, stray heat transfers from the outside surfaces of the plant components have detrimental effects on performance, since such losses reduce the extent of conversion from heat input to work output. Frictional effects resulting in pressure drops are sources of internal irreversibility as the working fluid flows through the boiler, condenser, and piping connecting the various components. Detailed thermodynamic analyses would account for these effects. For simplicity, however, they are ignored in the subsequent discussions. Thus, Fig. 8.6 shows no pressure drops for flow through the boiler and condenser or between plant components. Another effect on performance is suggested by the placement of state 3 on Fig. 8.6. At this state, the temperature of the working fluid exiting the condenser would be lower than the saturation temperature corresponding to the condenser pressure. This is disadvantageous because a greater heat transfer would be required in the boiler to bring the water to saturation.

In the next example, the ideal Rankine cycle of Example 8.1 is modified to include the effects of irreversibilities in the turbine and pump.

EXAMPLE 8.2 Rankine Cycle with Irreversibilities

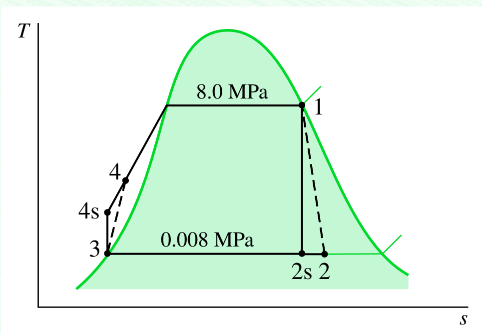
Reconsider the vapor power cycle of Example 8.1, but include in the analysis that the turbine and the pump each have an isentropic efficiency of 85%. Determine for the modified cycle **(a)** the thermal efficiency, **(b)** the mass flow rate of steam, in kg/h, for a net power output of 100 MW, **(c)** the rate of heat transfer $\dot{Q}_{in}$ into the working fluid as it passes through the boiler, in MW, **(d)** the rate of heat transfer $\dot{Q}_{out}$ from the condensing steam as it passes through the condenser, in MW, **(e)** the mass flow rate of the condenser cooling water, in kg/h, if cooling water enters the condenser at 15°C and exits as 35°C. Discuss the effects on the vapor cycle of irreversibilities within the turbine and pump.

SOLUTION

Known: A vapor power cycle operates with steam as the working fluid. The turbine and pump both have efficiencies of 85%.

Find: Determine the thermal efficiency, the mass flow rate, in kg/h, the rate of heat transfer to the working fluid as it passes through the boiler, in MW, the heat transfer rate from the condensing steam as it passes through the condenser, in MW, and the mass flow rate of the condenser cooling water, in kg/h. Discuss.

Schematic and Given Data:



Assumptions:

1. Each component of the cycle is analyzed as a control volume at steady state.

2. The working fluid passes through the boiler and condenser at constant pressure. Saturated vapor enters the turbine. The condensate is saturated at the condenser exit.

3. The turbine and pump each operate adiabatically with an efficiency of 85%.

4. Kinetic and potential energy effects are negligible.

◀ **Figure E8.2**

Analysis: Owing to the presence of irreversibilities during the expansion of the steam through the turbine, there is an increase in specific entropy from turbine inlet to exit, as shown on the accompanying *T–s* diagram. Similarly, there is an increase in specific entropy from pump inlet to exit. Let us begin the analysis by fixing each of the principal states. State 1 is the same as in Example 8.1, so $h_1 = 2758.0$ kJ/kg and $s_1 = 5.7432$ kJ/kg · K.

The specific enthalpy at the turbine exit, state 2, can be determined using the turbine efficiency

$$\eta_t = \frac{\dot{W}_t/\dot{m}}{(\dot{W}_t/\dot{m})_s} = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

where h_{2s} is the specific enthalpy at state 2s on the accompanying *T–s* diagram. From the solution to Example 8.1, $h_{2s} =$ 1794.8 kJ/kg. Solving for h_2 and inserting known values

$$\begin{aligned} h_2 &= h_1 - \eta_t(h_1 - h_{2s}) \\ &= 2758 - 0.85(2758 - 1794.8) = 1939.3\,\text{kJ/kg} \end{aligned}$$

State 3 is the same as in Example 8.1, so $h_3 = 173.88$ kJ/kg.

To determine the specific enthalpy at the pump exit, state 4, reduce mass and energy rate balances for a control volume around the pump to obtain $\dot{W}_p/\dot{m} = h_4 - h_3$. On rearrangement, the specific enthalpy at state 4 is

$$h_4 = h_3 + \dot{W}_p/\dot{m}$$

To determine h_4 from this expression requires the pump work, which can be evaluated using the pump efficiency η_p, as follows. By definition

$$\eta_p = \frac{(\dot{W}_p/\dot{m})_s}{(\dot{W}_p/\dot{m})}$$

The term $(\dot{W}_p/\dot{m})_s$ can be evaluated using Eq. 8.7b. Then solving for $\dot{W}_p/\dot{m}$ results in

$$\frac{\dot{W}_p}{\dot{m}} = \frac{v_3(p_4 - p_3)}{\eta_p}$$

The numerator of this expression was determined in the solution to Example 8.1. Accordingly,

$$\frac{\dot{W}_p}{\dot{m}} = \frac{8.06\,\text{kJ/kg}}{0.85} = 9.48\ \text{kJ/kg}$$

The specific enthalpy at the pump exit is then

$$h_4 = h_3 + \dot{W}_p/\dot{m} = 173.88 + 9.48 = 183.36\ \text{kJ/kg}$$

(a) The net power developed by the cycle is

$$\dot{W}_{\text{cycle}} = \dot{W}_t - \dot{W}_p = \dot{m}[(h_1 - h_2) - (h_4 - h_3)]$$

The rate of heat transfer to the working fluid as it passes through the boiler is

$$\dot{Q}_{\text{in}} = \dot{m}(h_1 - h_4)$$

Thus, the thermal efficiency is

$$\eta = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

Inserting values

$$\eta = \frac{(2758 - 1939.3) - 9.48}{2758 - 183.36} = 0.314\ (31.4\%)$$

(b) With the net power expression of part (a), the mass flow rate of the steam is

$$\dot{m} = \frac{\dot{W}_{\text{cycle}}}{(h_1 - h_2) - (h_4 - h_3)}$$
$$= \frac{(100\,\text{MW})|3600\ \text{s/h}|\,|10^3\,\text{kW/MW}|}{(818.7 - 9.48)\ \text{kJ/kg}} = 4.449 \times 10^5\,\text{kg/h}$$

(c) With the expression for $\dot{Q}_{\text{in}}$ from part (a) and previously determined specific enthalpy values

$$\dot{Q}_{\text{in}} = \dot{m}(h_1 - h_4)$$
$$= \frac{(4.449 \times 10^5\ \text{kg/h})(2758 - 183.36)\ \text{kJ/kg}}{|3600\ \text{s/h}|\,|10^3\,\text{kW/MW}|} = 318.2\,\text{MW}$$

(d) The rate of heat transfer from the condensing steam to the cooling water is

$$\dot{Q}_{\text{out}} = \dot{m}(h_2 - h_3)$$
$$= \frac{(4.449 \times 10^5\ \text{kg/h})(1939.3 - 173.88)\ \text{kJ/kg}}{|3600\ \text{s/h}|\,|10^3\,\text{kW/MW}|} = 218.2\,\text{MW}$$

(e) The mass flow rate of the cooling water can be determined from

$$\dot{m}_{\text{cw}} = \frac{\dot{m}(h_2 - h_3)}{(h_{\text{cw,out}} - h_{\text{cw,in}})}$$
$$= \frac{(218.2\,\text{MW})|10^3\ \text{kW/MW}|\,|3600\ \text{s/h}|}{(146.68 - 62.99)\,\text{kJ/kg}} = 9.39 \times 10^6\,\text{kg/h}$$

The effect of irreversibilities within the turbine and pump can be gauged by comparing the present values with their counterparts in Example 8.1. In this example, the turbine work per unit of mass is less and the pump work per unit of mass is greater than in Example 8.1. The thermal efficiency in the present case is less than in the ideal case of the previous example. For a fixed net power output (100 MW), the smaller net work output per unit mass in the present case dictates a greater mass flow rate of steam. The magnitude of the heat transfer to the cooling water is greater in this example than in Example 8.1; consequently, a greater mass flow rate of cooling water would be required.

8.3 Improving Performance—Superheat and Reheat

The representations of the vapor power cycle considered thus far do not depict actual vapor power plants faithfully, for various modifications are usually incorporated to improve overall performance. In this section we consider two cycle modifications known as *superheat* and *reheat*. Both features are normally incorporated into vapor power plants.

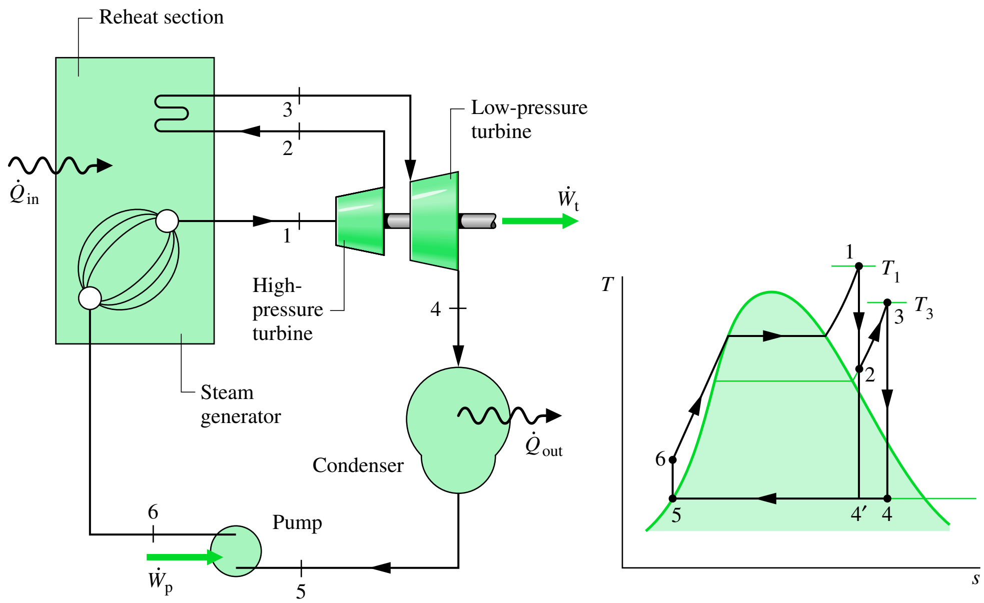
Let us begin the discussion by noting that an increase in the boiler pressure or a decrease in the condenser pressure may result in a reduction of the steam quality at the exit of the turbine. This can be seen by comparing states 2′ and 2″ of Figs. 8.4*a* and 8.4*b* to the corresponding state 2 of each diagram. If the quality of the mixture passing through the turbine becomes too low, the impact of liquid droplets in the flowing liquid–vapor mixture can erode the turbine blades, causing a decrease in the turbine efficiency and an increased need for maintenance. Accordingly, common practice is to maintain at least 90% quality ($x \geqq 0.9$) at the turbine exit. The cycle modifications known as *superheat* and *reheat* permit advantageous operating pressures in the boiler and condenser and yet offset the problem of low quality of the turbine exhaust.

superheat

SUPERHEAT. First, let us consider ***superheat.*** As we are not limited to having saturated vapor at the turbine inlet, further energy can be added by heat transfer to the steam, bringing it to a superheated vapor condition at the turbine inlet. This is accomplished in a separate heat exchanger called a superheater. The combination of boiler and superheater is referred to as a *steam generator*. Figure 8.3 shows an ideal Rankine cycle with superheated vapor at the turbine inlet: cycle 1′–2′–3–4–1′. The cycle with superheat has a higher average temperature of heat addition than the cycle without superheating (cycle 1–2–3–4–1), so the thermal efficiency is higher. Moreover, the quality at turbine exhaust state 2′ is greater than at state 2, which would be the turbine exhaust state without superheating. Accordingly, superheating also tends to alleviate the problem of low steam quality at the turbine exhaust. With sufficient superheating, the turbine exhaust state may even fall in the superheated vapor region.

reheat

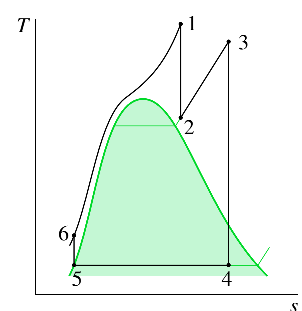
REHEAT. A further modification normally employed in vapor power plants is ***reheat.*** With reheat, a power plant can take advantage of the increased efficiency that results with higher boiler pressures and yet avoid low-quality steam at the turbine exhaust. In the ideal reheat cycle shown in Fig. 8.7, the steam does not expand to the condenser pressure in a single stage. The steam expands through a first-stage turbine (Process 1–2) to some pressure between the steam generator and condenser pressures. The steam is then reheated in the steam generator (Process 2–3). Ideally, there would be no pressure drop as the steam is reheated. After reheating, the steam expands in a second-stage turbine to the condenser pressure (Process 3–4). The principal advantage of reheat is to increase the quality of the steam at the turbine exhaust. This can be seen from the *T–s* diagram of Fig. 8.7 by comparing state 4 with



▲ **Figure 8.7** Ideal reheat cycle.

state 4′, the turbine exhaust state without reheating. When computing the thermal efficiency of a reheat cycle, it is necessary to account for the work output of both turbine stages as well as the total heat addition occurring in the vaporization/superheating and reheating processes. This calculation is illustrated in Example 8.3.

SUPERCRITICAL CYCLE. The temperature of the steam entering the turbine is restricted by metallurgical limitations imposed by the materials used to fabricate the superheater, reheater, and turbine. High pressure in the steam generator also requires piping that can withstand great stresses at elevated temperatures. Although these factors limit the gains that can be realized through superheating and reheating, improved materials and methods of fabrication have permitted significant increases over the years in the maximum allowed cycle temperatures and steam generator pressures, with corresponding increases in thermal efficiency. This has progressed to the extent that vapor power plants can be designed to operate with steam generator pressures exceeding the critical pressure of water 22.1 MPa, and turbine inlet temperatures exceeding 600°C. Figure 8.8 shows an ideal reheat cycle with a supercritical steam generator pressure. Observe that no phase change occurs during the heat addition process from 6 to 1.



▲ **Figure 8.8** Supercritical ideal reheat cycle.

In the next example, the ideal Rankine cycle of Example 8.1 is modified to include superheat and reheat.

EXAMPLE 8.3 Ideal Reheat Cycle

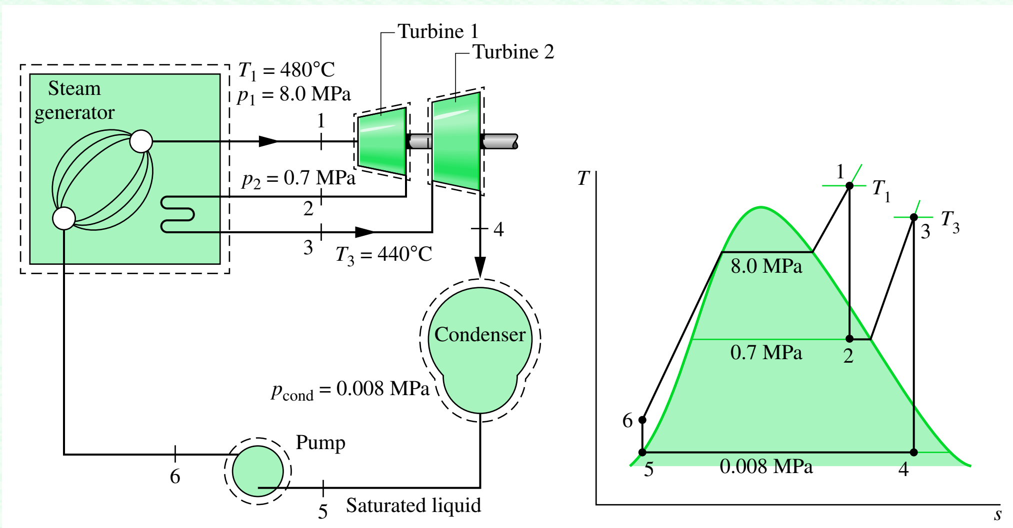
Steam is the working fluid in an ideal Rankine cycle with superheat and reheat. Steam enters the first-stage turbine at 8.0 MPa, 480°C, and expands to 0.7 MPa. It is then reheated to 440°C before entering the second-stage turbine, where it expands to the condenser pressure of 0.008 MPa. The *net* power output is 100 MW. Determine **(a)** the thermal efficiency of the cycle, **(b)** the mass flow rate of steam, in kg/h, **(c)** the rate of heat transfer $\dot{Q}_{out}$ from the condensing steam as it passes through the condenser, in MW. Discuss the effects of reheat on the vapor power cycle.

SOLUTION

Known: An ideal reheat cycle operates with steam as the working fluid. Operating pressures and temperatures are specified, and the net power output is given.

Find: Determine the thermal efficiency, the mass flow rate of the steam, in kg/h, and the heat transfer rate from the condensing steam as it passes through the condenser, in MW. Discuss.

Schematic and Given Data:



◀ **Figure E8.3**

Assumptions:

1. Each component in the cycle is analyzed as a control volume at steady state. The control volumes are shown on the accompanying sketch by dashed lines.
2. All processes of the working fluid are internally reversible.
3. The turbine and pump operate adiabatically.
4. Condensate exits the condenser as saturated liquid.
5. Kinetic and potential energy effects are negligible.

Analysis: To begin, we fix each of the principal states. Starting at the inlet to the first turbine stage, the pressure is 8.0 MPa and the temperature is 480°C, so the steam is a superheated vapor. From Table A-4, $h_1 = 3348.4$ kJ/kg and $s_1 = 6.6586$ kJ/kg · K.

State 2 is fixed by $p_2 = 0.7$ MPa and $s_2 = s_1$ for the isentropic expansion through the first-stage turbine. Using saturated liquid and saturated vapor data from Table A-3, the quality at state 2 is

$$x_2 = \frac{s_2 - s_f}{s_g - s_f} = \frac{6.6586 - 1.9922}{6.708 - 1.9922} = 0.9895$$

The specific enthalpy is then

$$\begin{aligned} h_2 &= h_f + x_2 h_{fg} \\ &= 697.22 + (0.9895)2066.3 = 2741.8 \text{ kJ/kg} \end{aligned}$$

State 3 is superheated vapor with $p_3 = 0.7$ MPa and $T_3 = 440°C$, so from Table A-4, $h_3 = 3353.3$ kJ/kg and $s_3 = 7.7571$ kJ/kg · K.

To fix state 4, use $p_4 = 0.008$ MPa and $s_4 = s_3$ for the isentropic expansion through the second-stage turbine. With data from Table A-3, the quality at state 4 is

$$x_4 = \frac{s_4 - s_f}{s_g - s_f} = \frac{7.7571 - 0.5926}{8.2287 - 0.5926} = 0.9382$$

The specific enthalpy is

$$h_4 = 173.88 + (0.9382)2403.1 = 2428.5 \text{ kJ/kg}$$

State 5 is saturated liquid at 0.008 MPa, so $h_5 = 173.88$ kJ/kg. Finally, the state at the pump exit is the same as in Example 8.1, so $h_6 = 181.94$ kJ/kg.

(a) The *net* power developed by the cycle is

$$\dot{W}_{\text{cycle}} = \dot{W}_{t1} + \dot{W}_{t2} - \dot{W}_p$$

Mass and energy rate balances for the two turbine stages and the pump reduce to give, respectively

$$\begin{aligned} &\text{Turbine 1:} && \dot{W}_{t1}/\dot{m} = h_1 - h_2 \\ &\text{Turbine 2:} && \dot{W}_{t2}/\dot{m} = h_3 - h_4 \\ &\text{Pump:} && \dot{W}_p/\dot{m} = h_6 - h_5 \end{aligned}$$

where $\dot{m}$ is the mass flow rate of the steam.

The total rate of heat transfer to the working fluid as it passes through the boiler–superheater and reheater is

$$\frac{\dot{Q}_{\text{in}}}{\dot{m}} = (h_1 - h_6) + (h_3 - h_2)$$

Using these expressions, the thermal efficiency is

$$\begin{aligned} \eta &= \frac{(h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)}{(h_1 - h_6) + (h_3 - h_2)} \\ &= \frac{(3348.4 - 2741.8) + (3353.3 - 2428.5) - (181.94 - 173.88)}{(3348.4 - 181.94) + (3353.3 - 2741.8)} \\ &= \frac{606.6 + 924.8 - 8.06}{3166.5 + 611.5} = \frac{1523.3 \text{ kJ/kg}}{3778 \text{ kJ/kg}} = 0.403\,(40.3\%) \end{aligned}$$

(b) The mass flow rate of the steam can be obtained with the expression for net power given in part (a).

$$\begin{aligned} \dot{m} &= \frac{\dot{W}_{\text{cycle}}}{(h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)} \\ &= \frac{(100 \text{ MW})|3600 \text{ s/h}|\,|10^3 \text{ kW/MW}|}{(606.6 + 924.8 - 8.06) \text{ kJ/kg}} = 2.363 \times 10^5 \text{ kg/h} \end{aligned}$$

(c) The rate of heat transfer from the condensing steam to the cooling water is

$$\begin{aligned} \dot{Q}_{\text{out}} &= \dot{m}(h_4 - h_5) \\ &= \frac{2.363 \times 10^5 \text{ kg/h}\,(2428.5 - 173.88) \text{ kJ/kg}}{|3600 \text{ s/h}|\,|10^3 \text{ kW/MW}|} = 148 \text{ MW} \end{aligned}$$

To see the effects of reheat, we compare the present values with their counterparts in Example 8.1. With superheat and reheat, the thermal efficiency is increased over that of the cycle of Example 8.1. For a specified net power output (100 MW), a larger thermal efficiency means that a smaller mass flow rate of steam is required. Moreover, with a greater thermal efficiency the rate of heat transfer to the cooling water is also less, resulting in a reduced demand for cooling water. With reheating, the steam quality at the turbine exhaust is substantially increased over the value for the cycle of Example 8.1.

The following example illustrates the effect of turbine irreversibilities on the ideal reheat cycle of Example 8.3.

EXAMPLE 8.4 Reheat Cycle with Turbine Irreversibility

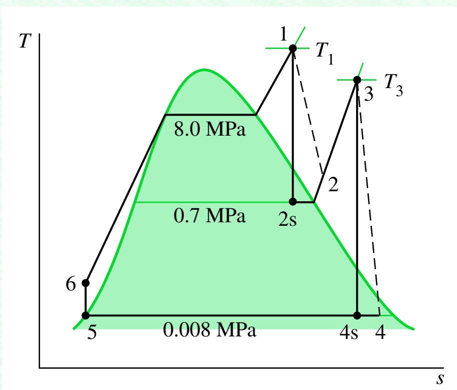
Reconsider the reheat cycle of Example 8.3, but include in the analysis that each turbine stage has the same isentropic efficiency. **(a)** If $\eta_t = 85\%$, determine the thermal efficiency. **(b)** Plot the thermal efficiency versus turbine stage efficiency ranging from 85 to 100%.

SOLUTION

Known: A reheat cycle operates with steam as the working fluid. Operating pressures and temperatures are specified. Each turbine stage has the same isentropic efficiency.

Find: If $\eta_t = 85\%$, determine the thermal efficiency. Also, plot the thermal efficiency versus turbine stage efficiency ranging from 85 to 100%.

Schematic and Given Data:



Assumptions:

1. As in Example 8.3, each component is analyzed as a control volume at steady state.
2. Except for the two turbine stages, all processes are internally reversible.
3. The turbine and pump operate adiabatically.
4. The condensate exits the condenser as saturated liquid.
5. Kinetic and potential energy effects are negligible.

◄ **Figure E8.4*a***

Analysis:

(a) From the solution to Example 8.3, the following specific enthalpy values are known, in kJ/kg: $h_1 = 3348.4$, $h_{2s} = 2741.8$, $h_3 = 3353.3$, $h_{4s} = 2428.5$, $h_5 = 173.88$, $h_6 = 181.94$.

The specific enthalpy at the exit of the first-stage turbine, h_2, can be determined by solving the expression for the turbine efficiency to obtain

$$\begin{aligned} h_2 &= h_1 - \eta_t(h_1 - h_{2s}) \\ &= 3348.4 - 0.85(3348.4 - 2741.8) = 2832.8 \text{ kJ/kg} \end{aligned}$$

The specific enthalpy at the exit of the second-stage turbine can be found similarly:

$$\begin{aligned} h_4 &= h_3 - \eta_t(h_3 - h_{4s}) \\ &= 3353.3 - 0.85(3353.3 - 2428.5) = 2567.2 \text{ kJ/kg} \end{aligned}$$

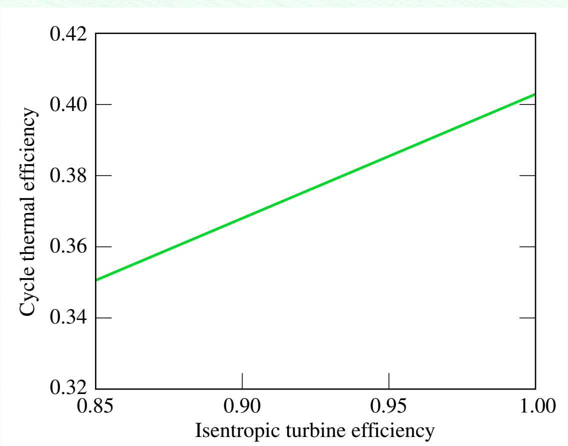
The thermal efficiency is then

1
$$\begin{aligned} \eta &= \frac{(h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)}{(h_1 - h_6) + (h_3 - h_2)} \\ &= \frac{(3348.4 - 2832.8) + (3353.3 - 2567.2) - (181.94 - 173.88)}{(3348.4 - 181.94) + (3353.3 - 2832.8)} \\ &= \frac{1293.6 \text{ kJ/kg}}{3687.0 \text{ kJ/kg}} = 0.351\ (35.1\%) \end{aligned}$$

(b) The *IT* code for the solution follows, where etat1 is η_{t1}, etat2 is η_{t2}, eta is η, Wnet = $\dot{W}_{net}/\dot{m}$, and Qin = $\dot{Q}_{in}/\dot{m}$.

```
// Fix the states

T1 = 480 // °C
p1 = 80 // bar
h1 = h_PT("Water/Steam", p1, T1)
s1 = s_PT("Water/Steam", p1, T1)

p2 = 7 // bar
h2s = h_Ps("Water/Steam", p2, s1)
etat1 = 0.85
h2 = h1 – etat1 * (h1 – h2s)

T3 = 440 // °C
p3 = p2
h3 = h_PT("Water/Steam", p3, T3)
s3 = s_PT("Water/Steam", p3, T3)

p4 = 0.08 // bar
h4s = h_Ps("Water/Steam", p4, s3)
etat2 = etat1
h4 = h3 – etat2 * (h3 – h4s)

p5 = p4
h5 = hsat_Px("Water/Steam", p5, 0) // kJ/kg
v5 = vsat_Px("Water/Steam", p5, 0) // m3/kg

p6 = p1
h6 = h5 + v5 * (p6 – p5) * 100 // The 100 in this expression is a unit conversion factor.

// Calculate thermal efficiency
Wnet = (h1 – h2) + (h3 – h4) – (h6 – h5)
Qin = (h1 – h6) + (h3 – h2)
eta = Wnet/Qin
```

Using the **Explore** button, sweep eta from 0.85 to 1.0 in steps of 0.01. Then, using the **Graph** button, obtain the following plot:



◄ **Figure E8.4*b***

From Fig. E8.4*b*, we see that the cycle thermal efficiency increases from 0.351 to 0.403 as turbine stage efficiency increases from 0.85 to 1.00, as expected based on the results of Examples 8.3(a) and 8.4(a). Turbine isentropic efficiency is seen to have a significant effect on cycle thermal efficiency.

1 Owing to the irreversibilities present in the turbine stages, the net work per unit of mass developed in the present case is significantly less than in the case of Example 8.3. The thermal efficiency is also considerably less.

8.4 Improving Performance—Regenerative Vapor Power Cycle

regeneration

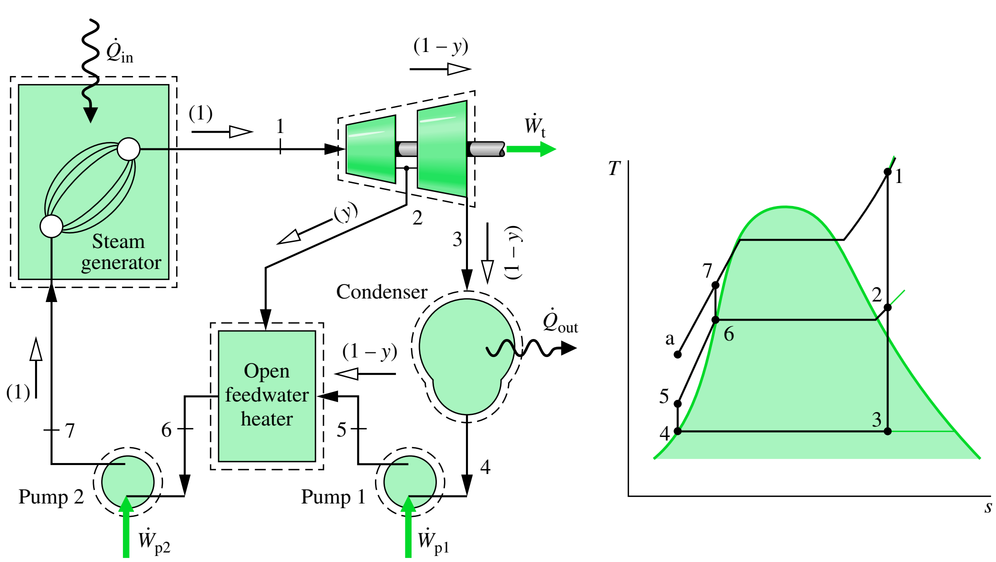
Another commonly used method for increasing the thermal efficiency of vapor power plants is *regenerative feedwater heating,* or simply **regeneration.** This is the subject of the present section.

To introduce the principle underlying regenerative feedwater heating, consider Fig. 8.3 once again. In cycle 1–2–3–4–a–1, the working fluid would enter the boiler as a compressed liquid at state 4 and be heated while in the liquid phase to state a. With regenerative feedwater heating, the working fluid would enter the boiler at a state *between* 4 and a. As a result, the average temperature of heat addition would be increased, thereby tending to increase the thermal efficiency.

▶ 8.4.1 Open Feedwater Heaters

open feedwater heater

Let us consider how regeneration can be accomplished using an **open feedwater heater,** a direct contact-type heat exchanger in which streams at different temperatures mix to form a stream at an intermediate temperature. Shown in Fig. 8.9 are the schematic diagram and the associated T–s diagram for a regenerative vapor power cycle having one open feedwater heater. For this cycle, the working fluid passes isentropically through the turbine stages and pumps, and the flow through the steam generator, condenser, and feedwater heater takes place with no pressure drop in any of these components. Steam enters the first-stage turbine at state 1 and expands to state 2, where a fraction of the total flow is *extracted,* or *bled,* into an open feedwater heater operating at the extraction pressure, p_2. The rest of the steam expands through the second-stage turbine to state 3. This portion of the total flow is condensed to saturated liquid, state 4, and then pumped to the extraction pressure and introduced into the feedwater heater at state 5. A single mixed stream exits the feedwater heater at state 6. For the case shown in Fig. 8.9, the mass flow rates of the streams entering the feedwater heater are chosen so that the stream exiting the feedwater heater is a saturated liquid at the extraction pressure. The liquid at state 6 is then pumped to the steam generator pressure and enters the steam generator at state 7. Finally, the working fluid is heated from state 7 to state 1 in the steam generator.



▲ **Figure 8.9** Regenerative vapor power cycle with one open feedwater heater.

Referring to the *T–s* diagram of the cycle, note that the heat addition would take place from state 7 to state 1, rather than from state a to state 1, as would be the case without regeneration. Accordingly, the amount of energy that must be supplied from the combustion of a fossil fuel, or another source, to vaporize and superheat the steam would be reduced. This is the desired outcome. Only a portion of the total flow expands through the second-stage turbine (Process 2–3), however, so less work would be developed as well. In practice, operating conditions are chosen so that the reduction in heat added more than offsets the decrease in net work developed, resulting in an increased thermal efficiency in regenerative power plants.

CYCLE ANALYSIS. Consider next the thermodynamic analysis of the regenerative cycle illustrated in Fig. 8.9. An important initial step in analyzing any regenerative vapor cycle is the evaluation of the mass flow rates through each of the components. Taking a single control volume enclosing both turbine stages, the mass rate balance reduces at steady state to

$$\dot{m}_2 + \dot{m}_3 = \dot{m}_1$$

where $\dot{m}_1$ is the rate at which mass enters the first-stage turbine at state 1, $\dot{m}_2$ is the rate at which mass is extracted and exits at state 2, and $\dot{m}_3$ is the rate at which mass exits the second-stage turbine at state 3. Dividing by $\dot{m}_1$ places this on the basis of a *unit of mass* passing through the first-stage turbine

$$\frac{\dot{m}_2}{\dot{m}_1} + \frac{\dot{m}_3}{\dot{m}_1} = 1$$

Denoting the fraction of the total flow extracted at state 2 by y ($y = \dot{m}_2/\dot{m}_1$), the fraction of the total flow passing through the second-stage turbine is

$$\frac{\dot{m}_3}{\dot{m}_1} = 1 - y \tag{8.11}$$

The fractions of the total flow at various locations are indicated on Fig. 8.9.

The fraction y can be determined by applying the conservation of mass and conservation of energy principles to a control volume around the feedwater heater. Assuming no heat transfer between the feedwater heater and its surroundings and ignoring kinetic and potential energy effects, the mass and energy rate balances reduce at steady state to give

$$0 = yh_2 + (1 - y)h_5 - h_6$$

Solving for y

$$y = \frac{h_6 - h_5}{h_2 - h_5} \tag{8.12}$$

Equation 8.12 allows the fraction y to be determined when states 2, 5, and 6 are fixed.

Expressions for the principal work and heat transfers of the regenerative cycle can be determined by applying mass and energy rate balances to control volumes around the individual components. Beginning with the turbine, the total work is the sum of the work developed by each turbine stage. Neglecting kinetic and potential energy effects and assuming no heat transfer with the surroundings, we can express the total turbine work on the basis of a unit of mass passing through the first-stage turbine as

$$\frac{\dot{W}_t}{\dot{m}_1} = (h_1 - h_2) + (1 - y)(h_2 - h_3) \tag{8.13}$$

The total pump work is the sum of the work required to operate each pump individually. On the basis of a unit of mass passing through the first-stage turbine, the total pump work is

$$\frac{\dot{W}_p}{\dot{m}_1} = (h_7 - h_6) + (1 - y)(h_5 - h_4) \tag{8.14}$$

The energy added by heat transfer to the working fluid passing through the steam generator, per unit of mass expanding through the first-stage turbine, is

$$\frac{\dot{Q}_{in}}{\dot{m}_1} = h_1 - h_7 \tag{8.15}$$

and the energy rejected by heat transfer to the cooling water is

$$\frac{\dot{Q}_{out}}{\dot{m}_1} = (1 - y)(h_3 - h_4) \tag{8.16}$$

The following example illustrates the analysis of a regenerative cycle with one open feedwater heater, including the evaluation of properties at state points around the cycle and the determination of the fractions of the total flow at various locations.

EXAMPLE 8.5 Regenerative Cycle with Open Feedwater Heater

Consider a regenerative vapor power cycle with one open feedwater heater. Steam enters the turbine at 8.0 MPa, 480°C and expands to 0.7 MPa, where some of the steam is extracted and diverted to the open feedwater heater operating at 0.7 MPa. The remaining steam expands through the second-stage turbine to the condenser pressure of 0.008 MPa. Saturated liquid exits the open feedwater heater at 0.7 MPa. The isentropic efficiency of each turbine stage is 85% and each pump operates isentropically. If the net power output of the cycle is 100 MW, determine **(a)** the thermal efficiency and **(b)** the mass flow rate of steam entering the first turbine stage, in kg/h.

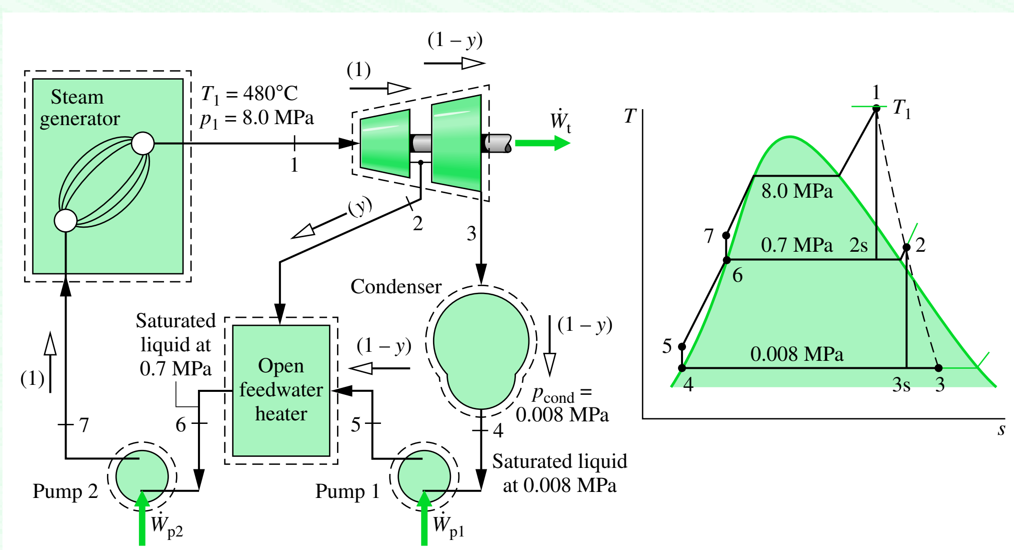
SOLUTION

Known: A regenerative vapor power cycle operates with steam as the working fluid. Operating pressures and temperatures are specified; the efficiency of each turbine stage and the net power output are also given.

Find: Determine the thermal efficiency and the mass flow rate into the turbine, in kg/h.

Schematic and Given Data:

1



◀ Figure E8.5

Assumptions:

1. Each component in the cycle is analyzed as a steady-state control volume. The control volumes are shown in the accompanying sketch by dashed lines.
2. All processes of the working fluid are internally reversible, except for the expansions through the two turbine stages and mixing in the open feedwater heater.
3. The turbines, pumps, and feedwater heater operate adiabatically.
4. Kinetic and potential energy effects are negligible.
5. Saturated liquid exits the open feedwater heater, and saturated liquid exits the condenser.

Analysis: The specific enthalpy at states 1 and 4 can be read from the steam tables. The specific enthalpy at state 2 is evaluated in the solution to Example 8.4. The specific entropy at state 2 can be obtained from the steam tables using the known values of enthalpy and pressure at this state. In summary, $h_1 = 3348.4$ kJ/kg, $h_2 = 2832.8$ kJ/kg, $s_2 = 6.8606$ kJ/kg · K, $h_4 = 173.88$ kJ/kg.

The specific enthalpy at state 3 can be determined using the efficiency of the second-stage turbine

$$h_3 = h_2 - \eta_t(h_2 - h_{3s})$$

With $s_{3s} = s_2$, the quality at state 3s is $x_{3s} = 0.8208$; using this, we get $h_{3s} = 2146.3$ kJ/kg. Hence

$$h_3 = 2832.8 - 0.85(2832.8 - 2146.3) = 2249.3 \text{ kJ/kg}$$

State 6 is saturated liquid at 0.7 MPa. Thus, $h_6 = 697.22$ kJ/kg.

Since the pumps are assumed to operate with no irreversibilities, the specific enthalpy values at states 5 and 7 can be determined as

$$\begin{aligned} h_5 &= h_4 + v_4(p_5 - p_4) \\ &= 173.88 + (1.0084 \times 10^{-3})(\text{m}^3/\text{kg})(0.7 - 0.008)\text{ MPa}\left|\frac{10^6 \text{ N/m}^2}{1 \text{ MPa}}\right|\left|\frac{1 \text{ kJ}}{10^3 \text{ N}\cdot\text{m}}\right| = 174.6 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} h_7 &= h_6 + v_6(p_7 - p_6) \\ &= 697.22 + (1.1080 \times 10^{-3})(8.0 - 0.7)|10^3| = 705.3 \text{ kJ/kg} \end{aligned}$$

Applying mass and energy rate balances to a control volume enclosing the open heater, we find the fraction y of the flow extracted at state 2 from

$$y = \frac{h_6 - h_5}{h_2 - h_5} = \frac{697.22 - 174.6}{2832.8 - 174.6} = 0.1966$$

(a) On the basis of a unit of mass passing through the first-stage turbine, the total turbine work output is

$$\begin{aligned} \frac{\dot{W}_t}{\dot{m}_1} &= (h_1 - h_2) + (1 - y)(h_2 - h_3) \\ &= (3348.4 - 2832.8) + (0.8034)(2832.8 - 2249.3) = 984.4 \text{ kJ/kg} \end{aligned}$$

The total pump work per unit of mass passing through the first-stage turbine is

$$\begin{aligned} \frac{\dot{W}_p}{\dot{m}_1} &= (h_7 - h_6) + (1 - y)(h_5 - h_4) \\ &= (705.3 - 697.22) + (0.8034)(174.6 - 173.88) = 8.7 \text{ kJ/kg} \end{aligned}$$

The heat added in the steam generator per unit of mass passing through the first-stage turbine is

$$\frac{\dot{Q}_{in}}{\dot{m}_1} = h_1 - h_7 = 3348.4 - 705.3 = 2643.1 \text{ kJ/kg}$$

The thermal efficiency is then

$$\eta = \frac{\dot{W}_t/\dot{m}_1 - \dot{W}_p/\dot{m}_1}{\dot{Q}_{in}/\dot{m}_1} = \frac{984.4 - 8.7}{2643.1} = 0.369 \; (36.9\%)$$

(b) The mass flow rate of the steam entering the turbine, $\dot{m}_1$, can be determined using the given value for the net power output, 100 MW. Since

$$\dot{W}_{\text{cycle}} = \dot{W}_{\text{t}} - \dot{W}_{\text{p}}$$

and

$$\frac{\dot{W}_{\text{t}}}{\dot{m}_1} = 984.4 \text{ kJ/kg} \qquad \text{and} \qquad \frac{\dot{W}_{\text{p}}}{\dot{m}_1} = 8.7 \text{ kJ/kg}$$

it follows that

$$\dot{m}_1 = \frac{(100 \text{ MW})|3600 \text{ s/h}|}{(984.4 - 8.7) \text{ kJ/kg}} \left| \frac{10^3 \text{ kJ/s}}{1 \text{ MW}} \right| = 3.69 \times 10^5 \text{ kg/h}$$

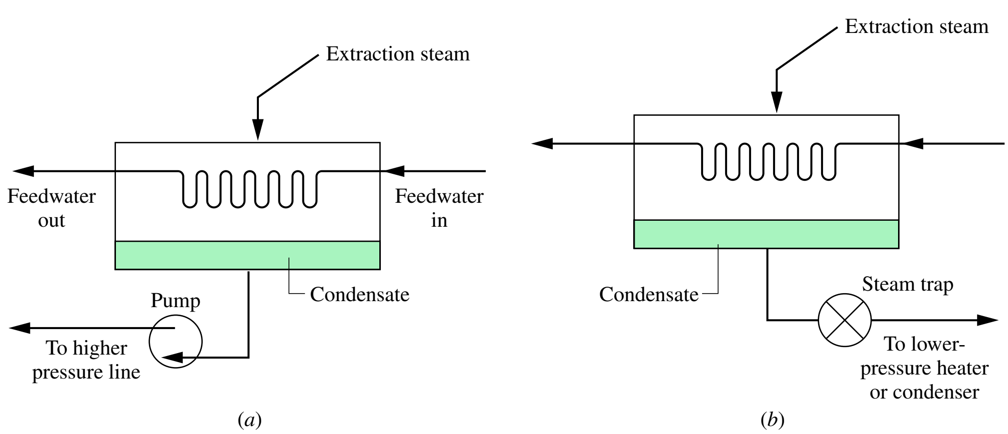
❶ Note that the fractions of the total flow at various locations are labeled on the figure.

▶ 8.4.2 Closed Feedwater Heaters

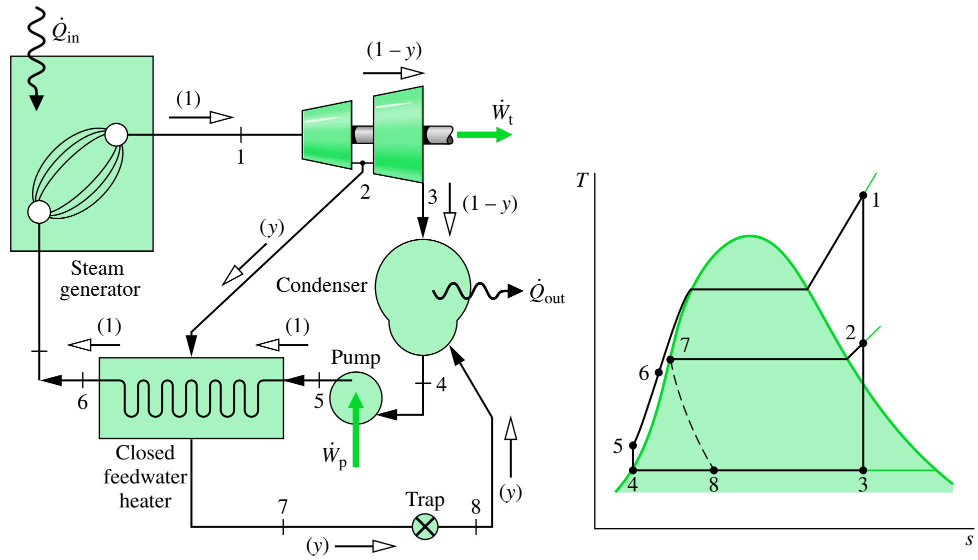
closed feedwater heater

Regenerative feedwater heating also can be accomplished with ***closed feedwater heaters.*** Closed heaters are shell-and-tube-type recuperators in which the feedwater temperature increases as the extracted steam condenses on the outside of the tubes carrying the feedwater. Since the two streams do not mix, they can be at different pressures. The diagrams of Fig. 8.10 show two different schemes for removing the condensate from closed feedwater heaters. In Fig. 8.10*a*, this is accomplished by means of a pump whose function is to pump the condensate forward to a higher-pressure point in the cycle. In Fig. 8.10*b*, the condensate is allowed to pass through a *trap* into a feedwater heater operating at a lower pressure or into the condenser. A trap is a type of valve that permits only liquid to pass through to a region of lower pressure.

A regenerative vapor power cycle having one closed feedwater heater with the condensate trapped into the condenser is shown schematically in Fig. 8.11. For this cycle, the working fluid passes isentropically through the turbine stages and pumps, and there are no pressure drops accompanying the flow through the other components. The T–s diagram shows the principal states of the cycle. The total steam flow expands through the first-stage turbine from state 1 to state 2. At this location, a fraction of the flow is bled into the closed feedwater heater, where it condenses. Saturated liquid at the extraction pressure exits the feedwater heater at state 7. The condensate is then trapped into the condenser, where it is reunited with



▲ **Figure 8.10** Examples of closed feedwater heaters.



▲ **Figure 8.11** Regenerative vapor power cycle with one closed feedwater heater.

the portion of the total flow passing through the second-stage turbine. The expansion from state 7 to state 8 through the trap is irreversible, so it is shown by a dashed line on the *T–s* diagram. The total flow exiting the condenser as saturated liquid at state 4 is pumped to the steam generator pressure and enters the feedwater heater at state 5. The temperature of the feedwater is increased in passing through the feedwater heater. The feedwater then exits at state 6. The cycle is completed as the working fluid is heated in the steam generator at constant pressure from state 6 to state 1. Although the closed heater shown on the figure operates with no pressure drop in either stream, there is a source of irreversibility due to the stream-to-stream temperature differences.

CYCLE ANALYSIS. The schematic diagram of the cycle shown in Fig. 8.11 is labeled with the fractions of the total flow at various locations. This is usually helpful in analyzing such cycles. The fraction of the total flow extracted, y, can be determined by applying the conservation of mass and conservation of energy principles to a control volume around the closed heater. Assuming no heat transfer between the feedwater heater and its surroundings and neglecting kinetic and potential energy effects, the mass and energy rate balances reduce at steady state to give

$$0 = y(h_2 - h_7) + (h_5 - h_6)$$

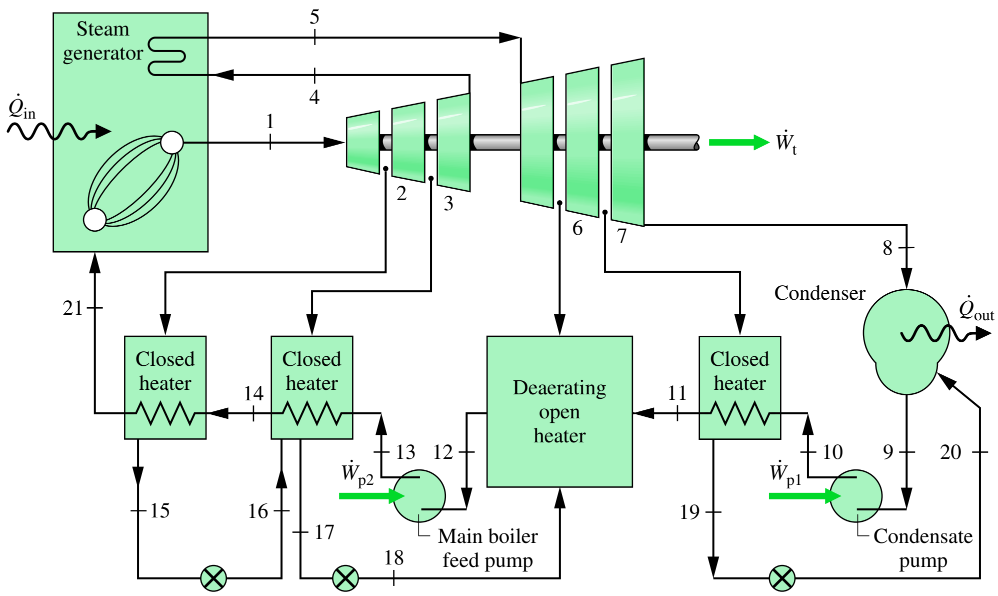
Solving for y

$$y = \frac{h_6 - h_5}{h_2 - h_7} \tag{8.17}$$

The principal work and heat transfers are evaluated as discussed previously.

▶ 8.4.3 Multiple Feedwater Heaters

The thermal efficiency of the regenerative cycle can be increased by incorporating several feedwater heaters at suitably chosen pressures. The number of feedwater heaters used is based on economic considerations, since incremental increases in thermal efficiency achieved with each additional heater must justify the added capital costs (heater, piping, pumps, etc.). Power



▲ **Figure 8.12** Example of a power plant layout.

plant designers use computer programs to simulate the thermodynamic and economic performance of different designs to help them decide on the number of heaters to use, the types of heaters, and the pressures at which they should operate.

Figure 8.12 shows the layout of a power plant with three closed feedwater heaters and one open heater. Power plants with multiple feedwater heaters ordinarily have at least one open feedwater heater operating at a pressure greater than atmospheric pressure so that oxygen and other dissolved gases can be vented from the cycle. This procedure, known as ***deaeration,*** is needed to maintain the purity of the working fluid in order to minimize corrosion. Actual power plants have many of the same basic features as the one shown in the figure.

deaeration

In analyzing regenerative vapor power cycles with multiple feedwater heaters, it is good practice to base the analysis on a unit of mass entering the first-stage turbine. To clarify the quantities of matter flowing through the various plant components, the fractions of the total flow removed at each extraction point and the fraction of the total flow remaining at each state point in the cycle should be labeled on a schematic diagram of the cycle. The fractions extracted are determined from mass and energy rate balances for control volumes around each of the feedwater heaters, starting with the highest-pressure heater and proceeding to each lower-pressure heater in turn. This procedure is used in the next example that involves a reheat–regenerative vapor power cycle with two feedwater heaters, one open feedwater heater and one closed feedwater heater.

EXAMPLE 8.6 Reheat–Regenerative Cycle with Two Feedwater Heaters

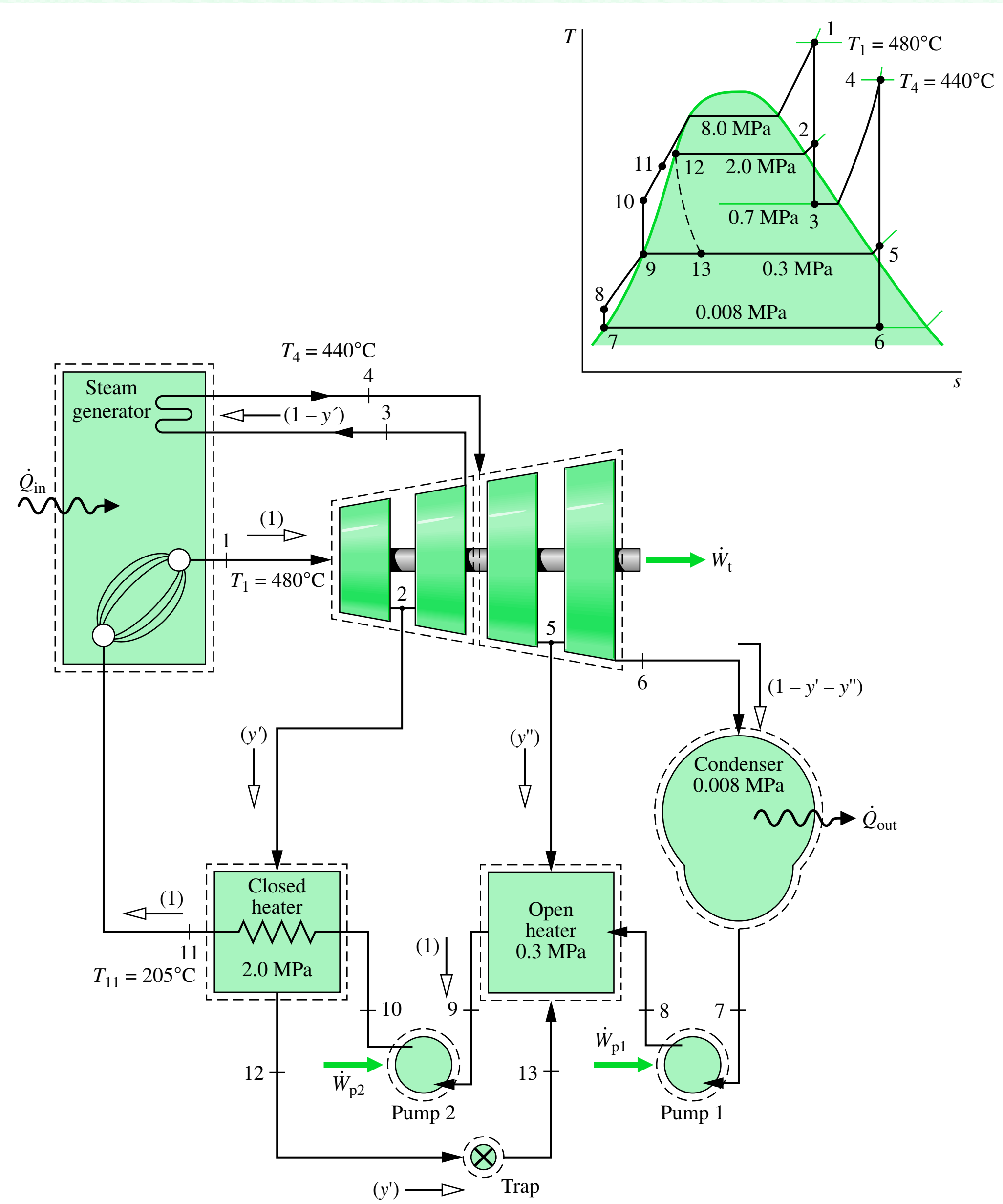
Consider a reheat–regenerative vapor power cycle with two feedwater heaters, a closed feedwater heater and an open feedwater heater. Steam enters the first turbine at 8.0 MPa, 480°C and expands to 0.7 MPa. The steam is reheated to 440°C before entering the second turbine, where it expands to the condenser pressure of 0.008 MPa. Steam is extracted from the first turbine at 2 MPa and fed to the closed feedwater heater. Feedwater leaves the closed heater at 205°C and 8.0 MPa, and condensate exits as saturated liquid at 2 MPa. The condensate is trapped into the open feedwater heater. Steam extracted from the second turbine at 0.3 MPa is also fed into the open feedwater heater, which operates at 0.3 MPa. The stream exiting the open feedwater heater is saturated liquid at 0.3 MPa. The *net* power output of the cycle is 100 MW. There is no stray heat transfer from any component to its surroundings. If the working fluid experiences no irreversibilities as it passes through the turbines, pumps, steam generator, reheater, and condenser, determine **(a)** the thermal efficiency, **(b)** the mass flow rate of the steam entering the first turbine, in kg/h.

SOLUTION

Known: A reheat–regenerative vapor power cycle operates with steam as the working fluid. Operating pressures and temperatures are specified, and the net power output is given.

Find: Determine the thermal efficiency and the mass flow rate entering the first turbine, in kg/h.

Schematic and Given Data:



◀ **Figure E8.6**

Assumptions:

1. Each component in the cycle is analyzed as a control volume at steady state. The control volumes are shown on the accompanying sketch by dashed lines.
2. There is no stray heat transfer from any component to its surroundings.
3. The working fluid undergoes internally reversible processes as it passes through the turbines, pumps, steam generator, reheater, and condenser.
4. The expansion through the trap is a *throttling* process.
5. Kinetic and potential energy effects are negligible.
6. Condensate exits the closed heater as a saturated liquid at 2 MPa. Feedwater exits the open heater as a saturated liquid at 0.3 MPa. Condensate exits the condenser as a saturated liquid.

Analysis: Let us determine the specific enthalpies at the principal states of the cycle. State 1 is the same as in Example 8.3, so $h_1 = 3348.4$ kJ/kg and $s_1 = 6.6586$ kJ/kg · K.

State 2 is fixed by $p_2 = 2.0$ MPa and the specific entropy s_2, which is the same as that of state 1. Interpolating in Table A-4, we get $h_2 = 2963.5$ kJ/kg. The state at the exit of the first turbine is the same as at the exit of the first turbine of Example 8.3, so $h_3 = 2741.8$ kJ/kg.

State 4 is superheated vapor at 0.7 MPa, 440°C. From Table A-4, $h_4 = 3353.3$ kJ/kg and $s_4 = 7.7571$ kJ/kg · K. Interpolating in Table A-4 at $p_5 = 0.3$ MPa and $s_5 = s_4 = 7.7571$ kJ/kg · K, the enthalpy at state 5 is $h_5 = 3101.5$ kJ/kg.

Using $s_6 = s_4$, the quality at state 6 is found to be $x_6 = 0.9382$. So

$$\begin{aligned} h_6 &= h_f + x_6 h_{fg} \\ &= 173.88 + (0.9382)2403.1 = 2428.5 \text{ kJ/kg} \end{aligned}$$

At the condenser exit, $h_7 = 173.88$ kJ/kg. The specific enthalpy at the exit of the first pump is

$$\begin{aligned} h_8 &= h_7 + v_7(p_8 - p_7) \\ &= 173.88 + (1.0084)(0.3 - 0.008) = 174.17 \text{ kJ/kg} \end{aligned}$$

The required unit conversions were considered in previous examples.

The liquid leaving the open feedwater heater at state 9 is saturated liquid at 0.3 MPa. The specific enthalpy is $h_9 = 561.47$ kJ/kg. The specific enthalpy at the exit of the second pump is

$$\begin{aligned} h_{10} &= h_9 + v_9(p_{10} - p_9) \\ &= 561.47 + (1.0732)(8.0 - 0.3) = 569.73 \text{ kJ/kg} \end{aligned}$$

The condensate leaving the closed heater is saturated at 2 MPa. From Table A-3, $h_{12} = 908.79$ kJ/kg. The fluid passing through the trap undergoes a throttling process, so $h_{13} = 908.79$ kJ/kg.

The specific enthalpy of the feedwater exiting the closed heater at 8.0 MPa and 205°C is found using Eq. 3.13 as

$$\begin{aligned} h_{11} &= h_f + v_f(p_{11} - p_{sat}) \\ &= 875.1 + (1.1646)(8.0 - 1.73) = 882.4 \text{ kJ/kg} \end{aligned}$$

where h_f and v_f are the saturated liquid specific enthalpy and specific volume at 205°C, respectively, and p_{sat} is the saturation pressure in MPa at this temperature. Alternatively, h_{11} can be found from Table A-5.

The schematic diagram of the cycle is labeled with the fractions of the total flow into the turbine that remain at various locations. The fractions of the total flow diverted to the closed heater and open heater, respectively, are $y' = \dot{m}_2/\dot{m}_1$ and $y'' = \dot{m}_5/\dot{m}_1$, where $\dot{m}_1$ denotes the mass flow rate entering the first turbine.

The fraction y' can be determined by application of mass and energy rate balances to a control volume enclosing the closed heater. The result is

$$y' = \frac{h_{11} - h_{10}}{h_2 - h_{12}} = \frac{882.4 - 569.73}{2963.5 - 908.79} = 0.1522$$

The fraction y'' can be determined by application of mass and energy rate balances to a control volume enclosing the open heater, resulting in

$$0 = y''h_5 + (1 - y' - y'')h_8 + y'h_{13} - h_9$$

Solving for y''

$$y'' = \frac{(1 - y')h_8 + y'h_{13} - h_9}{h_8 - h_5}$$

$$= \frac{(0.8478)174.17 + (0.1522)908.79 - 561.47}{174.17 - 3101.5}$$

$$= 0.0941$$

(a) The following work and heat transfer values are expressed on the basis of a unit mass entering the first turbine. The work developed by the first turbine per unit of mass entering is the sum

$$\frac{\dot{W}_{t1}}{\dot{m}_1} = (h_1 - h_2) + (1 - y')(h_2 - h_3)$$

$$= (3348.4 - 2963.5) + (0.8478)(2963.5 - 2741.8)$$

$$= 572.9 \text{ kJ/kg}$$

Similarly, for the second turbine

$$\frac{\dot{W}_{t2}}{\dot{m}_1} = (1 - y')(h_4 - h_5) + (1 - y' - y'')(h_5 - h_6)$$

$$= (0.8478)(3353.3 - 3101.5) + (0.7537)(3101.5 - 2428.5)$$

$$= 720.7 \text{ kJ/kg}$$

For the first pump

$$\frac{\dot{W}_{p1}}{\dot{m}_1} = (1 - y' - y'')(h_8 - h_7)$$

$$= (0.7537)(174.17 - 173.88) = 0.22 \text{ kJ/kg}$$

and for the second pump

$$\frac{\dot{W}_{p2}}{\dot{m}_1} = (h_{10} - h_9)$$

$$= 569.73 - 561.47 = 8.26 \text{ kJ/kg}$$

The total heat added is the sum of the energy added by heat transfer during boiling/superheating and reheating. When expressed on the basis of a unit of mass entering the first turbine, this is

$$\frac{\dot{Q}_{in}}{\dot{m}_1} = (h_1 - h_{11}) + (1 - y')(h_4 - h_3)$$

$$= (3348.4 - 882.4) + (0.8478)(3353.3 - 2741.8)$$

$$= 2984.4 \text{ kJ/kg}$$

With the foregoing values, the thermal efficiency is

$$\eta = \frac{\dot{W}_{t1}/\dot{m}_1 + \dot{W}_{t2}/\dot{m}_1 - \dot{W}_{p1}/\dot{m}_1 - \dot{W}_{p2}/\dot{m}_1}{\dot{Q}_{in}/\dot{m}_1}$$

$$= \frac{572.9 + 720.7 - 0.22 - 8.26}{2984.4} = 0.431 \; (43.1\%)$$

(b) The mass flow rate entering the first turbine can be determined using the given value of the net power output. Thus

$$\dot{m}_1 = \frac{\dot{W}_{cycle}}{\dot{W}_{t1}/\dot{m}_1 + \dot{W}_{t2}/\dot{m}_1 - \dot{W}_{p1}/\dot{m}_1 - \dot{W}_{p2}/\dot{m}_1}$$

1 $$= \frac{(100 \text{ MW})|3600 \text{ s/h}|\,|10^3 \text{ kW/MW}|}{1285.1 \text{ kJ/kg}} = 2.8 \times 10^5 \text{ kg/h}$$

1 Compared to the corresponding values determined for the simple Rankine cycle of Example 8.1, the thermal efficiency of the present regenerative cycle is substantially greater and the mass flow rate is considerably less.