Thermodynamics: An Engineering Approach
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CHAPTER 10
VAPOR AND COMBINED POWER CYCLES

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Objectives

- 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

THE CARNOT VAPOR CYCLE

The ***Carnot cycle*** is the most efficient cycle operating between two specified temperature limits

1-2 isothermal heat addition

2-3 isentropic expansion

3-4 isothermal heat rejection

4-1 isentropic compression

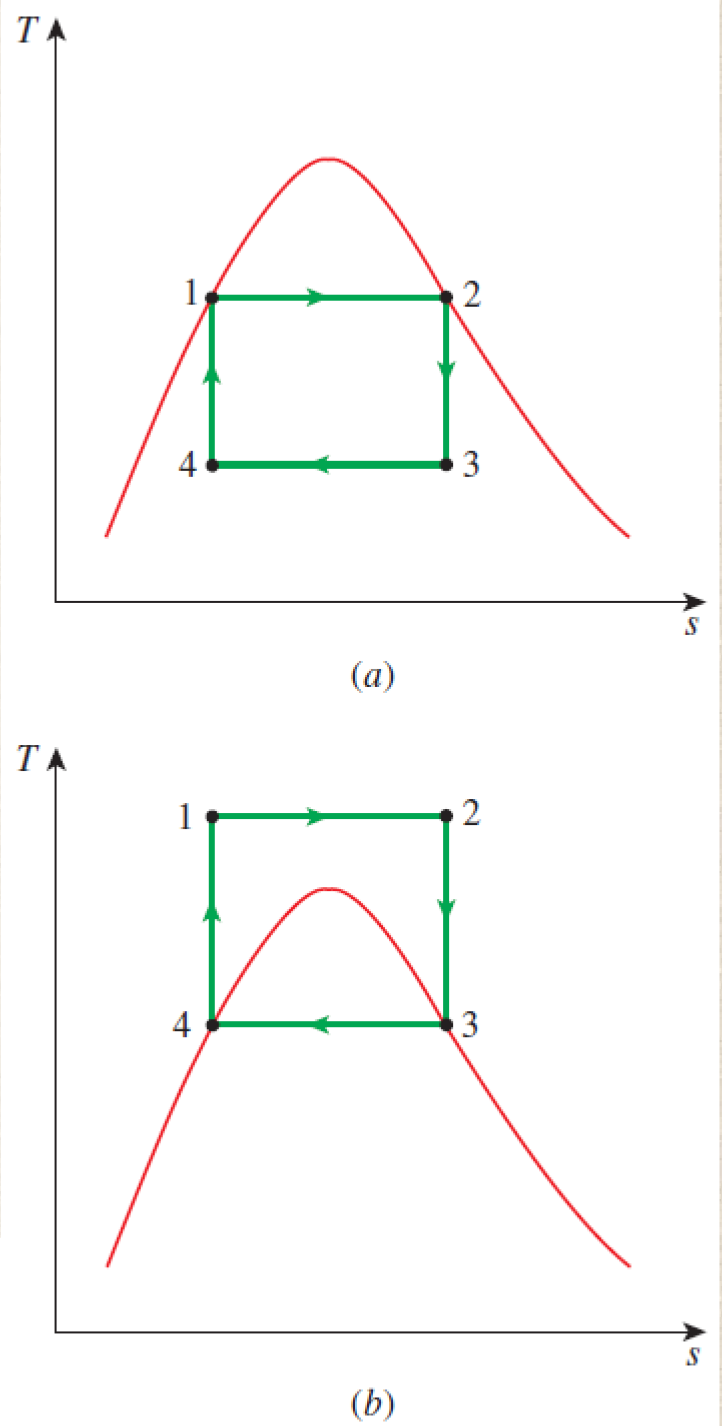
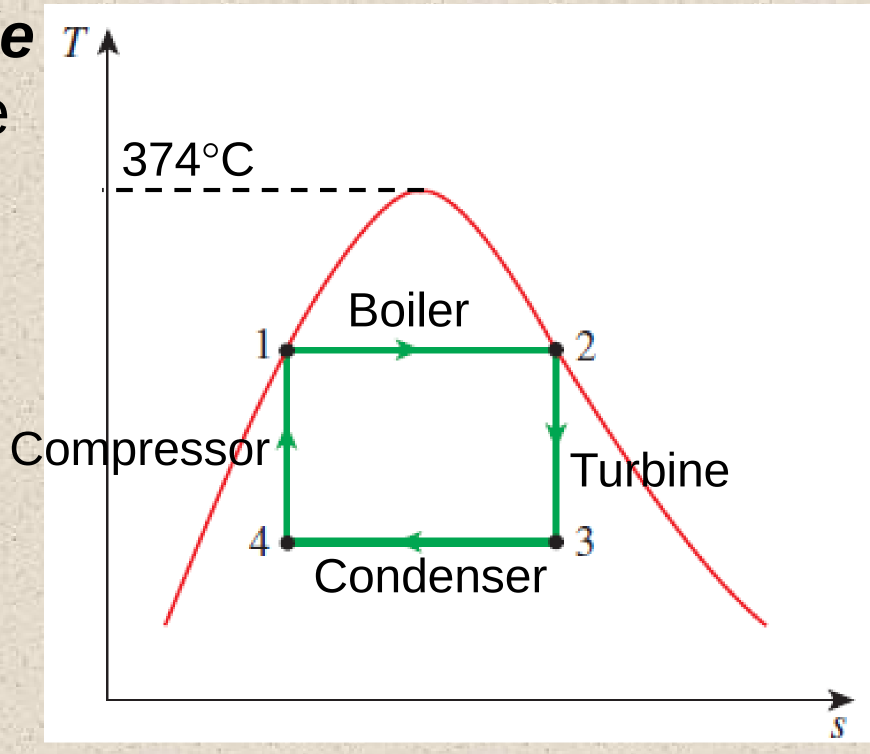


FIGURE 10–1
T-s diagram of two Carnot vapor cycles.

THE CARNOT VAPOR CYCLE

Consider a steady flow ***Carnot cycle*** executed within the saturation dome of a pure substance (water)



Impracticalities with this cycle:

Process 1-2 Limiting the heat transfer processes to two-phase systems severely limits the maximum temperature that can be used in the cycle (374°C for water)

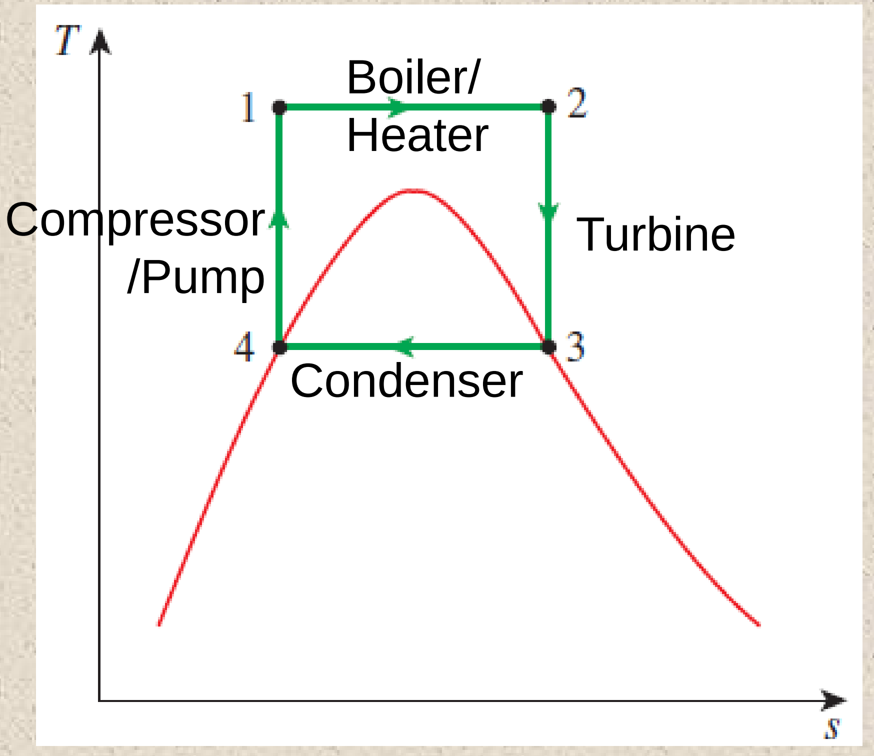
Process 2-3 The turbine cannot handle steam with a high moisture content (x < 0.9) because of the impingement of liquid droplets on the turbine blades causing erosion and wear

Process 4-1 It is not practical to design a compressor that handles two phases

THE CARNOT VAPOR CYCLE

This cycle is not suitable since it requires

- Isentropic compression to extremely high pressures
- Isothermal heat transfer at variable pressures



Carnot cycle cannot be approximated in actual devices and it is not a realistic model for vapor power cycles

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* → **Rankine cycle**

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

The ideal Rankine cycle does not involve any internal irreversibilities

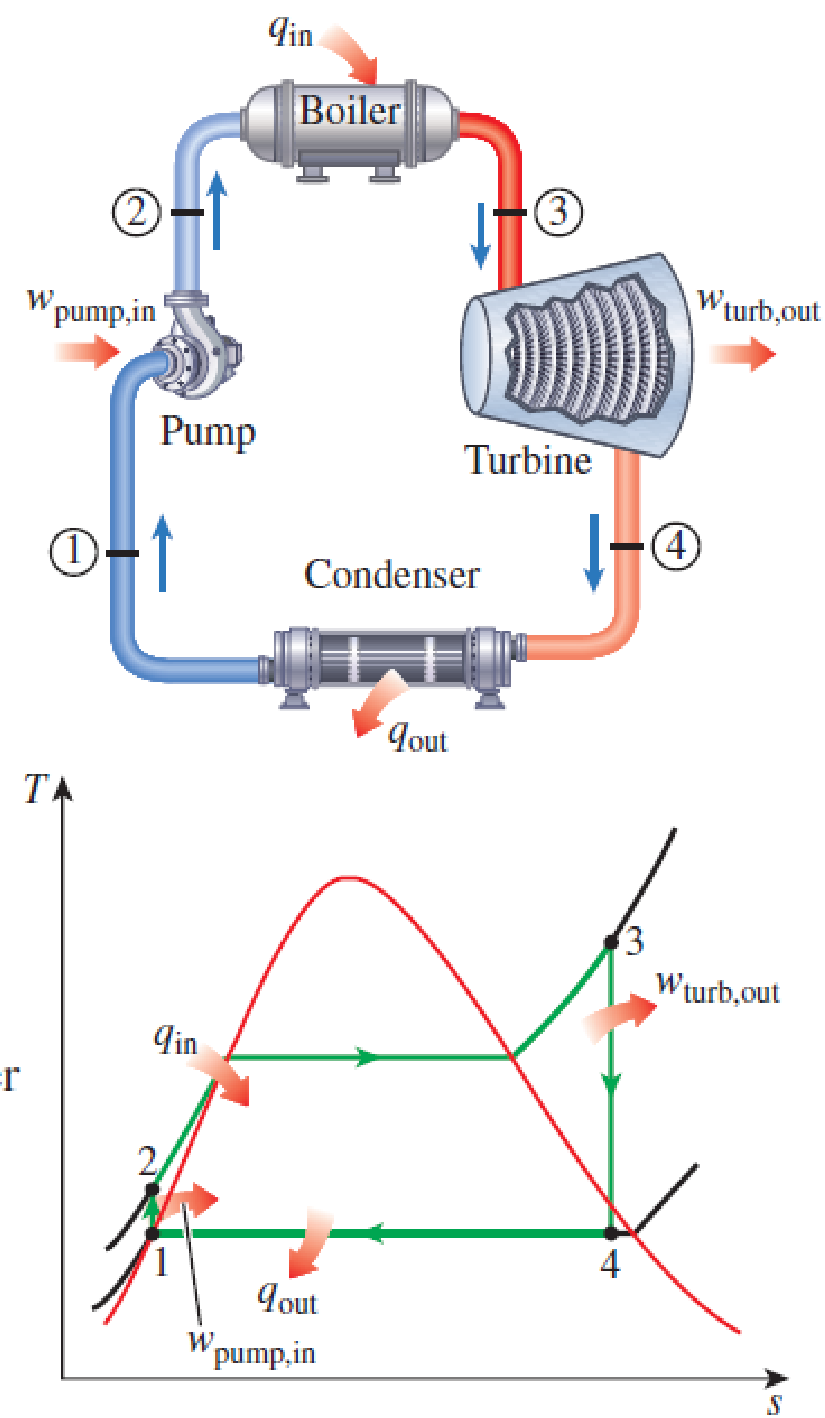
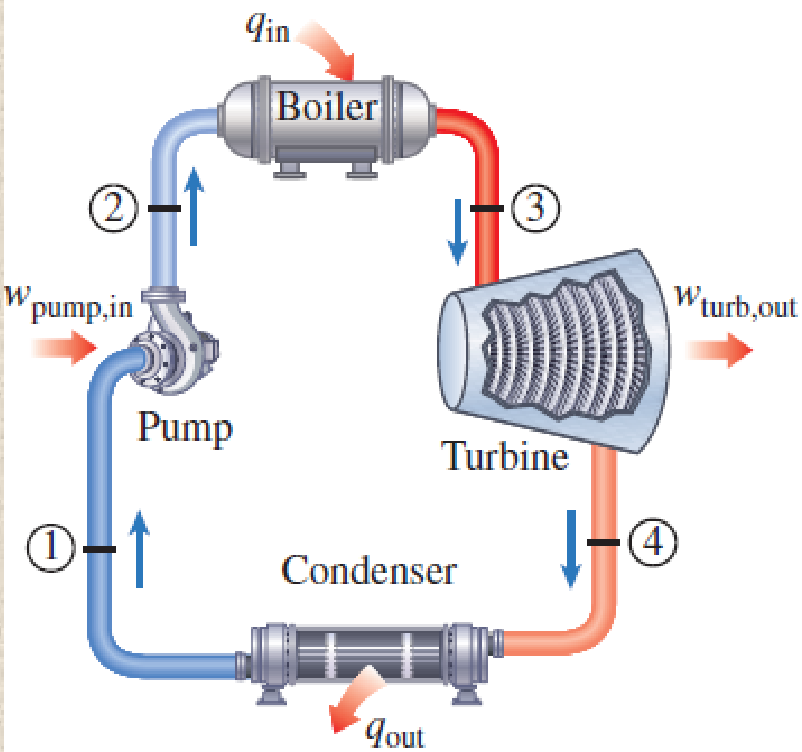


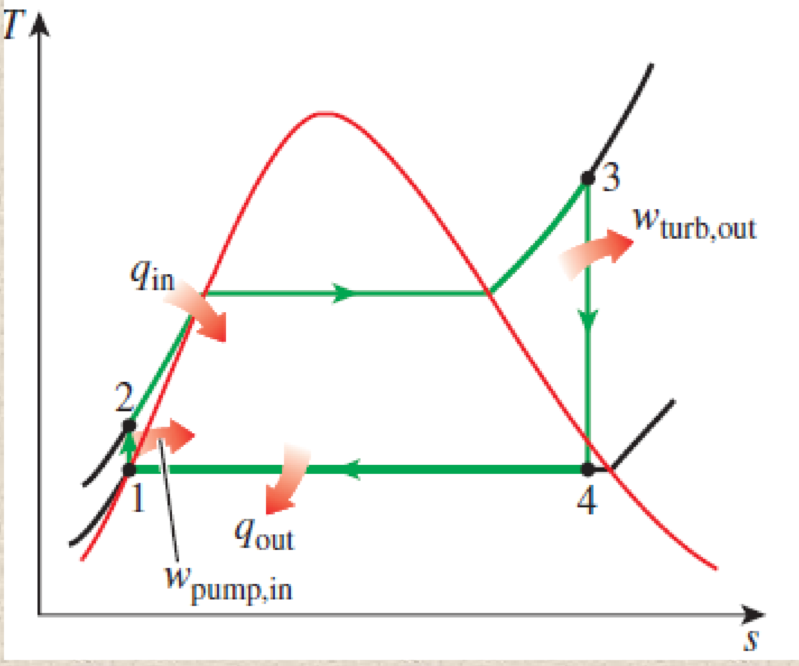
FIGURE 10–2

The simple ideal Rankine cycle.

RANKINE CYCLE

- Water enters the ***pump*** at state 1 as saturated liquid
- Water enters the ***boiler*** as a compressed liquid at state 2 and leaves as a superheated vapor at state 3
- Superheated vapor at state 3 enters the ***turbine*** and exits at state 4 where it enters the ***condenser***
- At state 4, steam is usually a saturated liquid-vapor mixture with a high quality (x > 0.9)
- Steam leave the ***condenser*** as saturated liquid (state 1), and enters the ***pump***, thus completing the cycle





Energy Analysis of the Ideal Rankine Cycle

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

Steady-flow energy equation (neglect ke, pe)

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

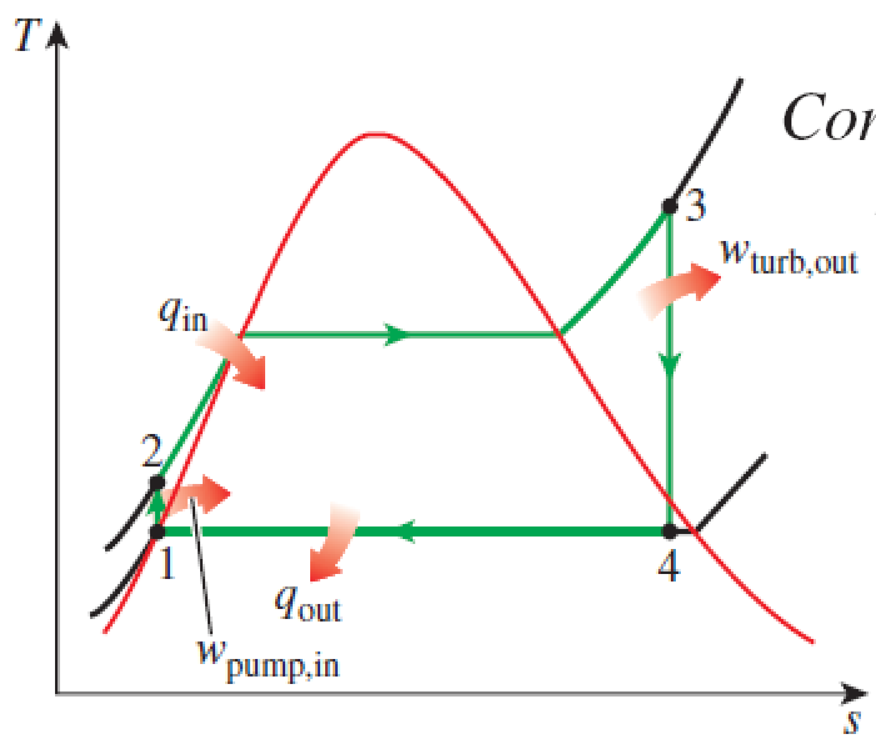
$$w_{pump,in} = v(P_2 - P_1)$$

$$h_1 = h_{f\,@\,P_1} \quad \text{and} \quad v \cong v_1 = v_{f\,@\,P_1}$$

$$Boiler\ (w = 0): \qquad q_{in} = h_3 - h_2$$

$$Turbine\ (q = 0): \qquad w_{turb,out} = h_3 - h_4$$

$$Condenser\ (w = 0): \qquad q_{out} = h_4 - h_1$$



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

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

The thermal efficiency can 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

EXAMPLE: 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.

State 1:
$$\left.\begin{array}{l} P_1 = 75 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \quad \begin{array}{l} h_1 = h_{f\,@\,75\text{ kPa}} = 384.44 \text{ kJ/kg} \\ v_1 = v_{f\,@\,75\text{ kPa}} = 0.001037 \text{ m}^3\text{/kg} \end{array}$$

State 2:
$$P_2 = 3 \text{ MPa}$$
$$s_2 = s_1$$

$$w_{\text{pump,in}} = v_1(P_2 - P_1)$$
$$= 3.03 \text{ kJ/kg}$$

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

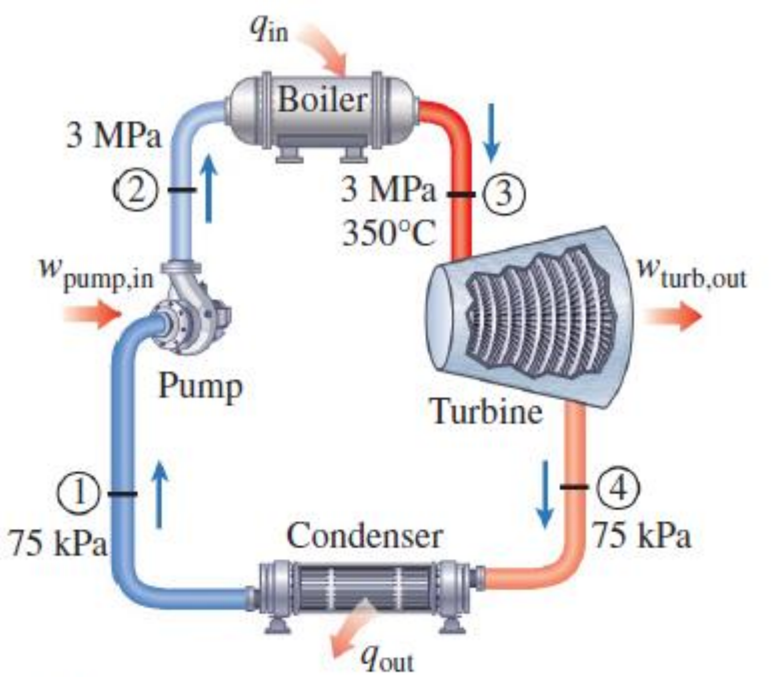
State 4:

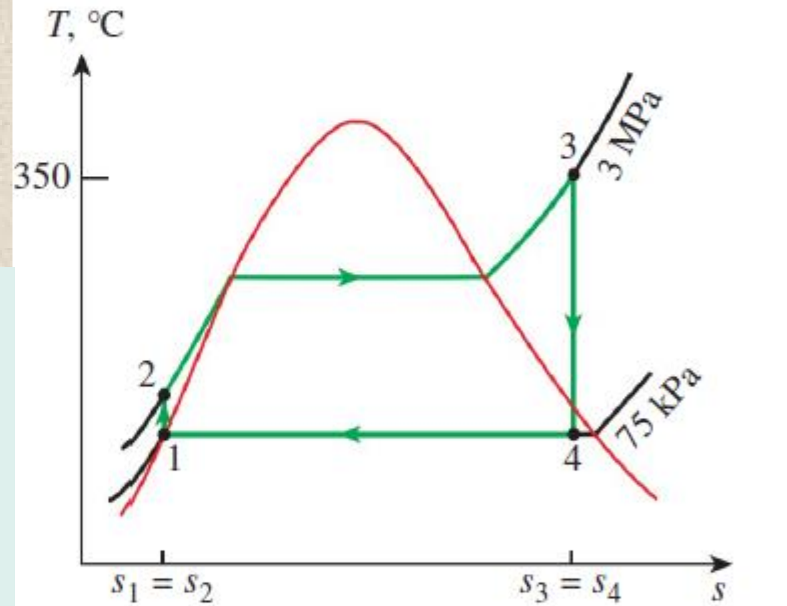
$P_4 = 75$ kPa (sat. mixture)

$$s_4 = s_3$$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 1.2132}{6.2426} = 0.8861$$

$$h_4 = h_f + x_4 h_{fg} = 384.44 + 0.8861(2278.0) = 2403.0 \text{ kJ/kg}$$

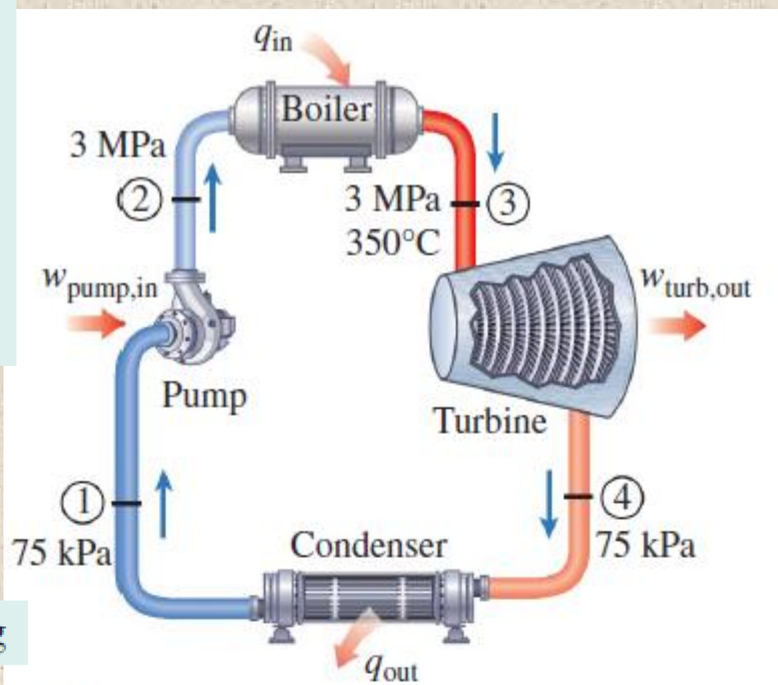




$$q_{in} = h_3 - h_2 = (3116.1 - 387.47)\ \text{kJ/kg} = 2728.6\ \text{kJ/kg}$$

$$q_{out} = h_4 - h_1 = (2403.0 - 384.44)\ \text{kJ/kg} = 2018.6\ \text{kJ/kg}$$

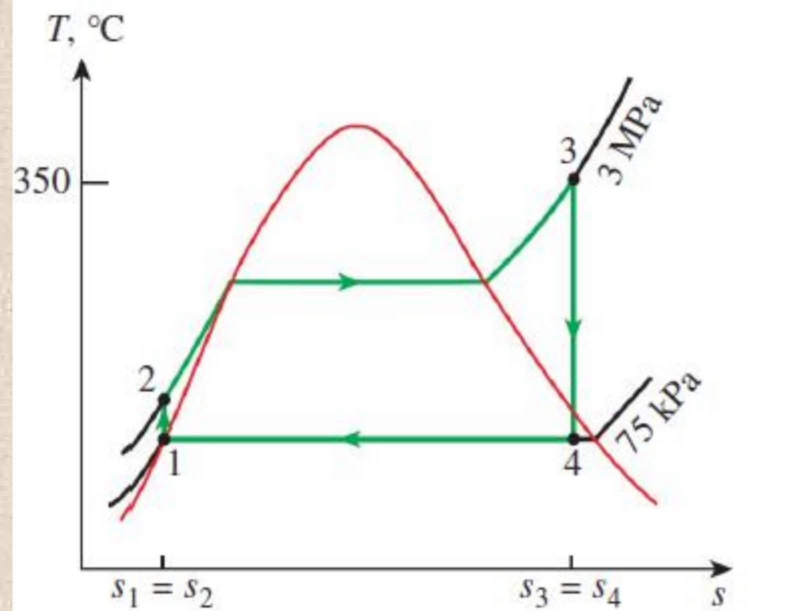
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{2018.6\ \text{kJ/kg}}{2728.6\ \text{kJ/kg}} = \mathbf{0.260\ \text{or}\ 26.0\%}$$



$$w_{turb,out} = h_3 - h_4 = (3116.1 - 2403.0)\ \text{kJ/kg} = 713.1\ \text{kJ/kg}$$

$$w_{net} = w_{turb,out} - w_{pump,in} = (713.1 - 3.03)\ \text{kJ/kg} = 710.1\ \text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{710.0\ \text{kJ/kg}}{2728.6\ \text{kJ/kg}} = \mathbf{0.260\ \text{or}\ 26.0\%}$$

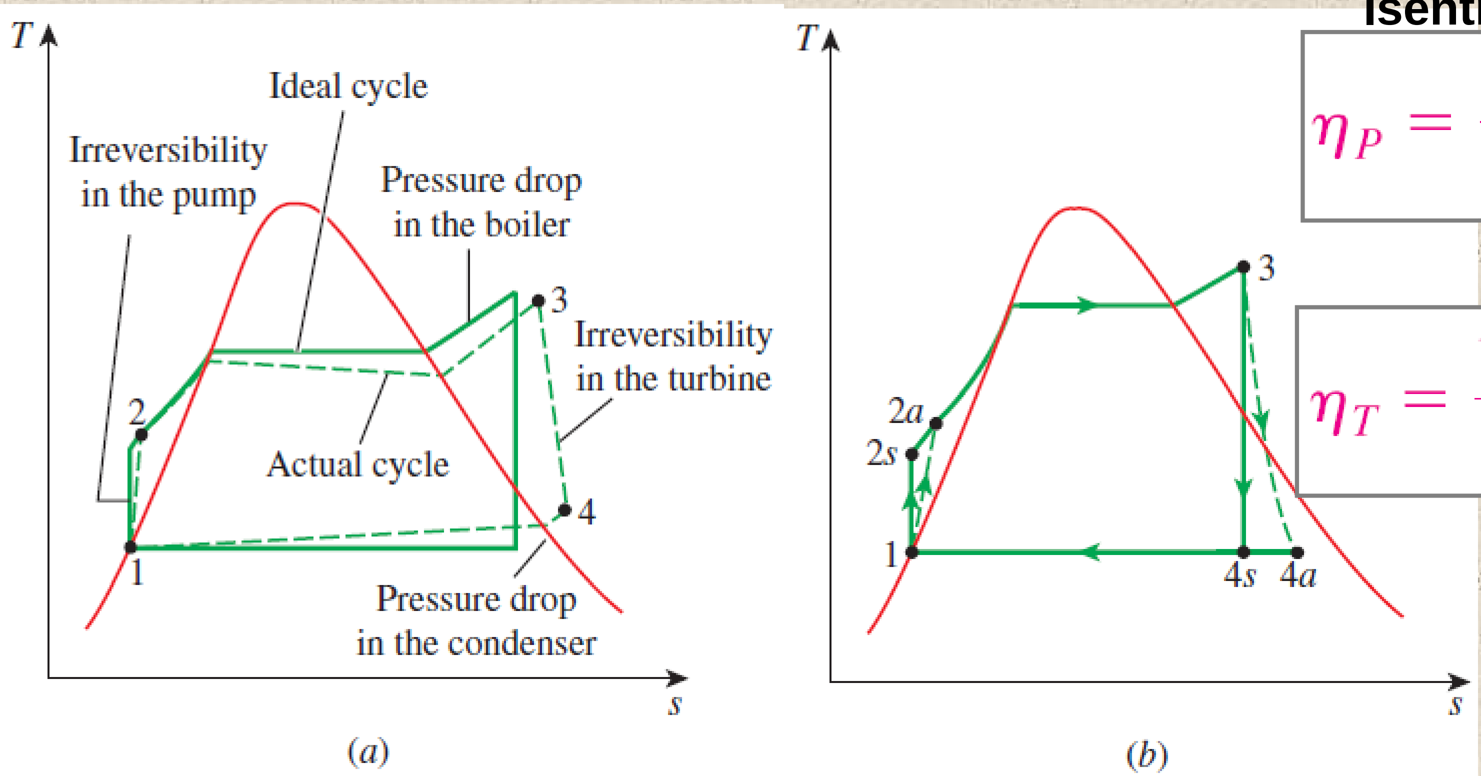


back work ratio ($r_{bw} = w_{in}/w_{out}$) = 3.03/713.1 = 0.004

DEVIATION OF ACTUAL VAPOR POWER CYCLES FROM IDEALIZED ONES

The actual vapor power cycle differs from the ideal Rankine cycle as a result of irreversibilities in various components

Fluid friction and heat loss to the surroundings are the two common sources of irreversibilities



Isentropic efficiencies

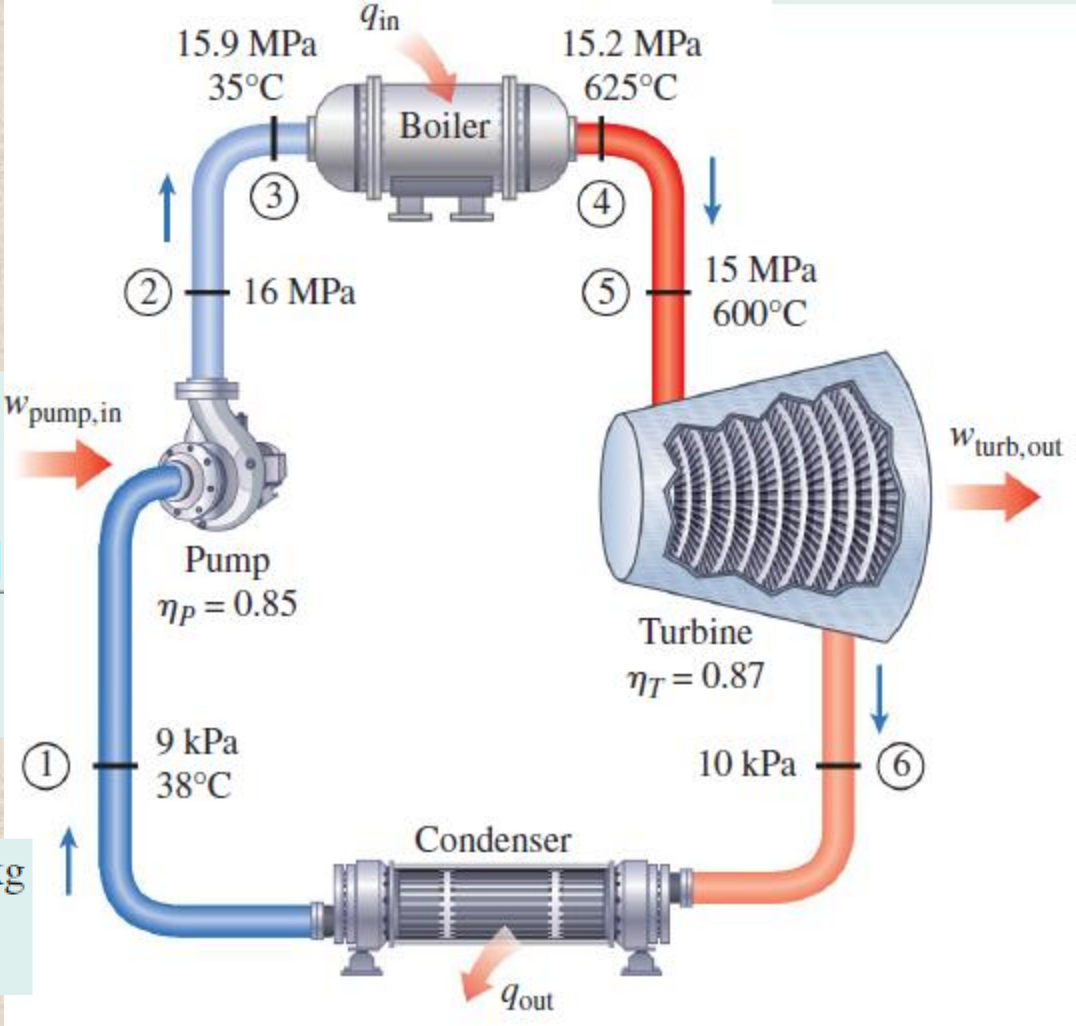
$$\eta_P = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

(*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.

EXAMPLE: AN ACTUAL STEAM POWER CYCLE

A steam power plant operates on the cycle shown. If the isentropic efficiency of the turbine is 87 percent and the isentropic efficiency of the pump is 85 percent, determine the thermal efficiency of the cycle



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

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

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

$$w_{turb,out} = \eta_T w_{s,turb,out}$$

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

$$= 1277.0 \text{ kJ/kg}$$

$$q_{in} = h_4 - h_3 = (3647.6 - 160.1) \text{ kJ/kg} = 3487.5 \text{ kJ/kg}$$

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

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

HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

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*
- *and/or decrease the average temperature at which heat is rejected from the working fluid in the condenser*

The average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection

Lowering the Condenser Pressure (*Lowers $T_{low,avg}$*)

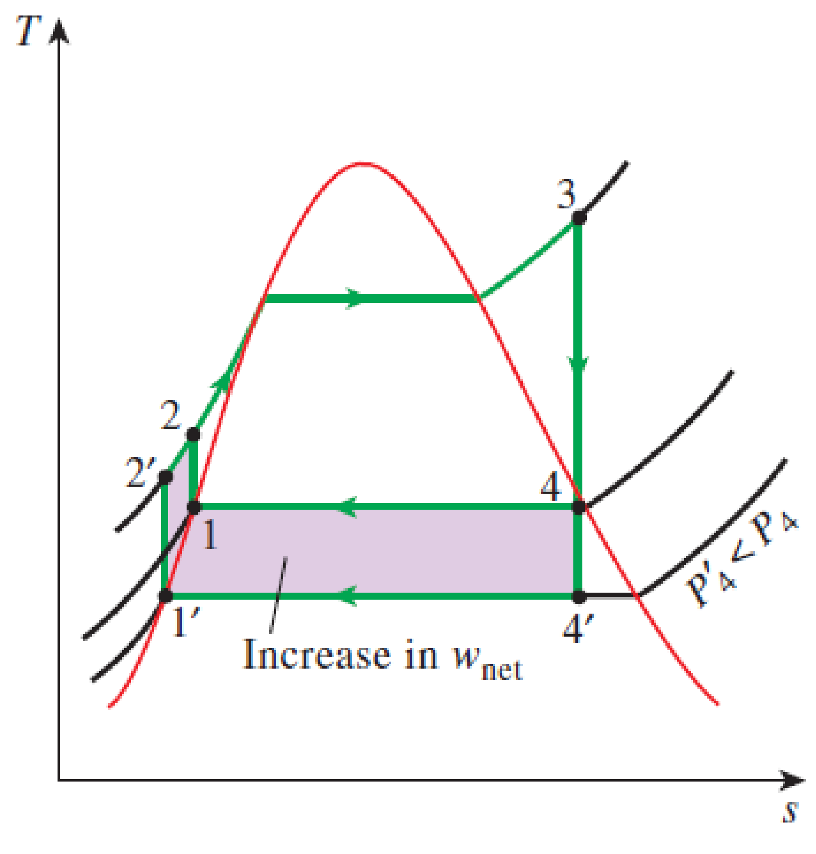


FIGURE 10–6

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

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’

- To take advantage of the increased efficiencies at low condenser temperatures, the condensers of steam power plants usually operate well below the atmospheric pressure
- There is a lower limit to this pressure depending on the temperature of the cooling medium (@P = 3.2 kPa, T_{sat} = 25°C)
- **Side effect:** lowering the condenser pressure increases the moisture content of the steam at the final stages of the turbine

Superheating the Steam to High Temperatures (*Increases* $T_{high,avg}$)

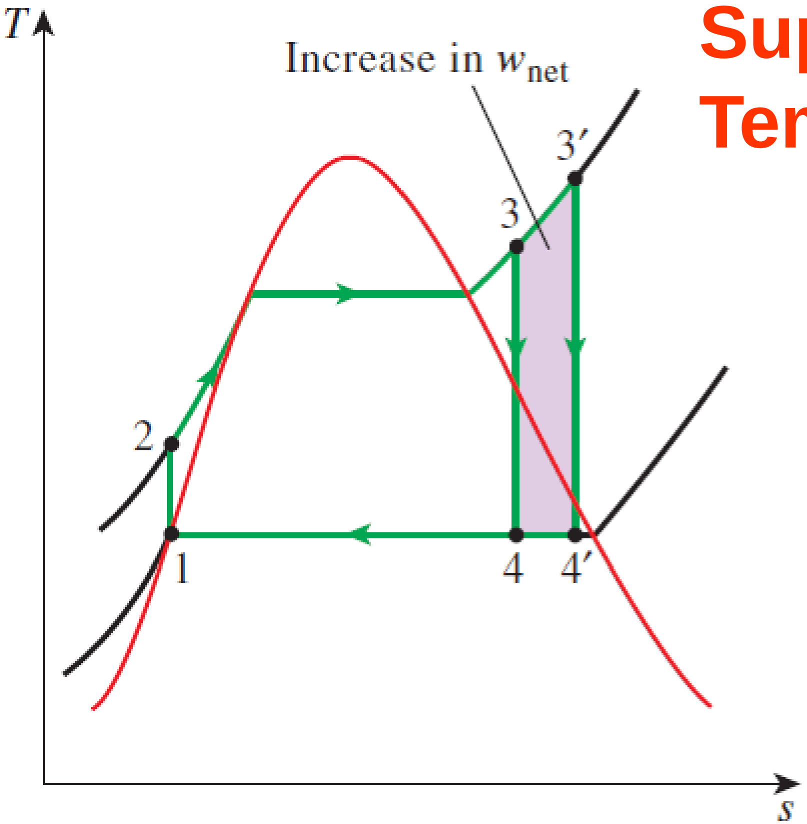


FIGURE 10–7
The effect of superheating the steam to higher temperatures on the ideal Rankine cycle.

The colored area on this diagram represents the increase in net work output as a result of increasing the boiler outlet temperature from T_3 to T_3'

- 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 since the average temperature at which heat is added increases.
- Superheating to higher temperatures decreases the moisture content of the steam at the turbine exit, which is desirable
- The temperature is limited by metallurgical considerations. Presently the highest steam temperature allowed at the turbine inlet is about 620°C

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
- For a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This side effect can be corrected by reheating the steam.

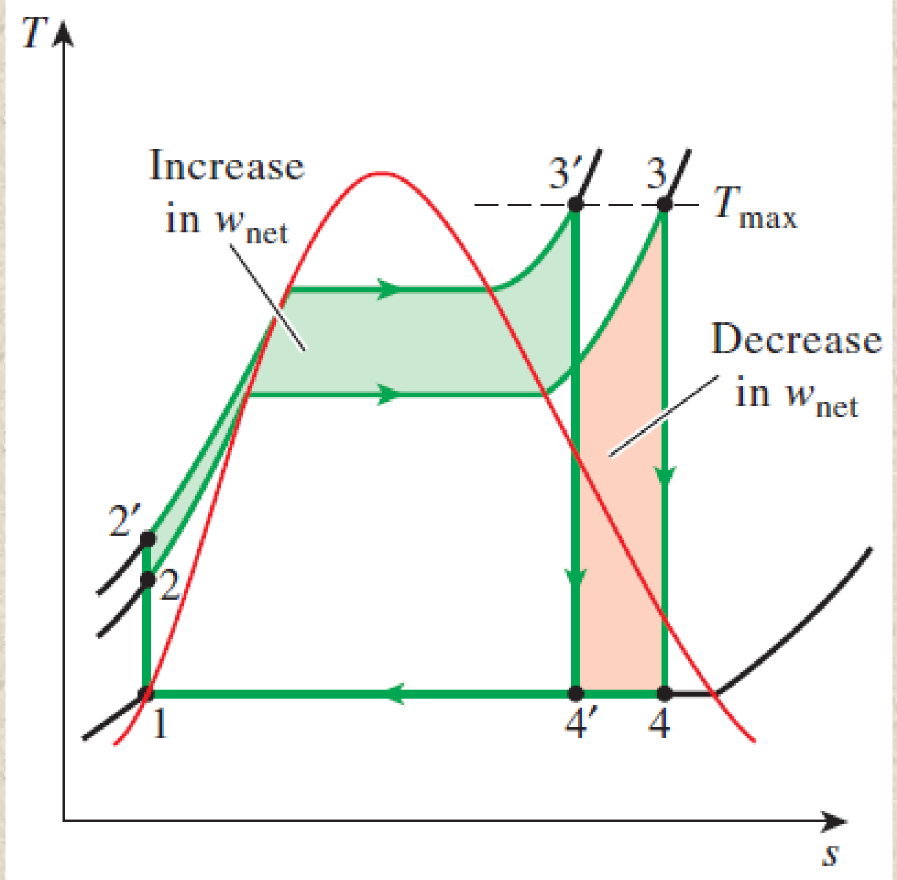


FIGURE 10–8
The effect of increasing the boiler pressure on the ideal Rankine cycle.

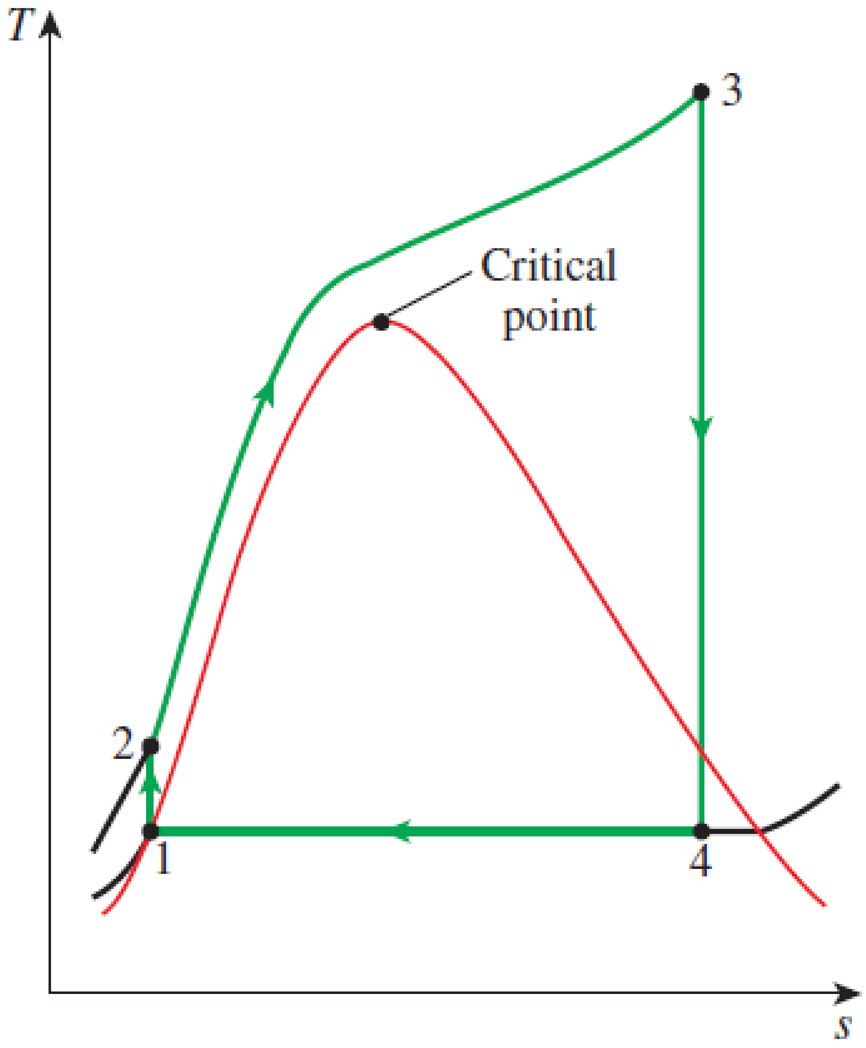
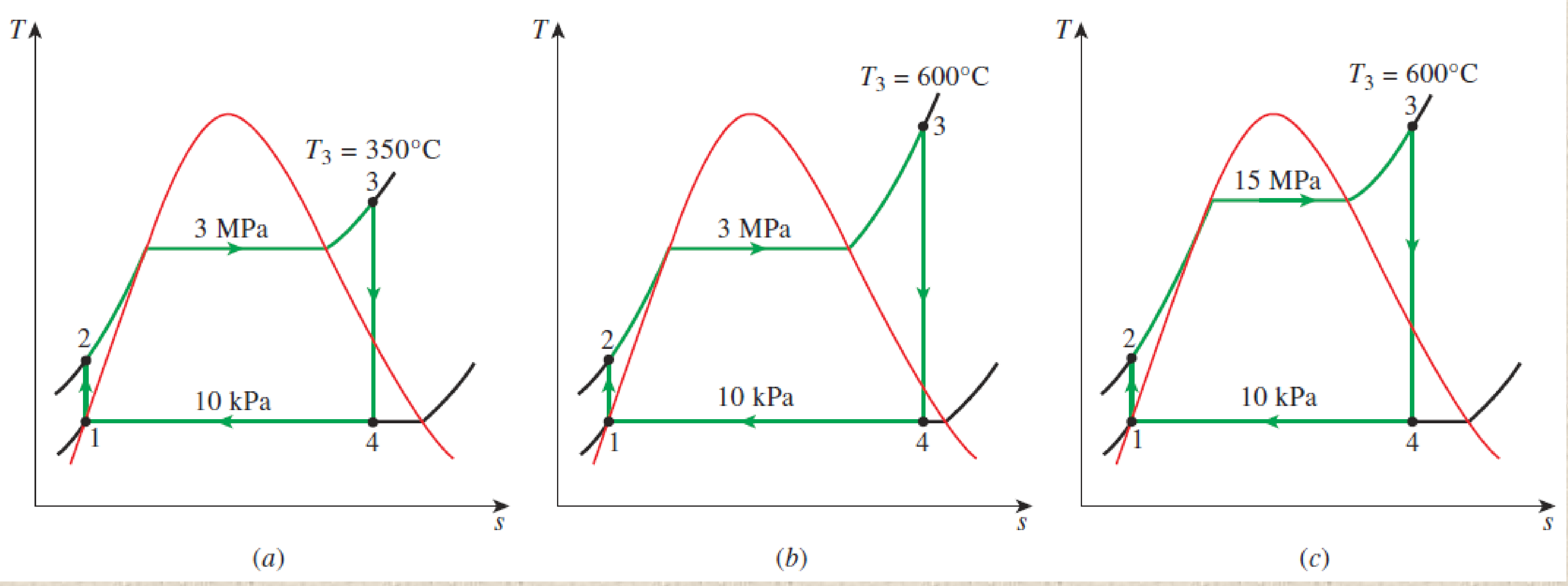


FIGURE 10–9
A supercritical Rankine cycle.

- Operating pressures of boiler have gradually increased over the years from about 2.7 MPa in 1922 to over 30 MPa today
- Today many modern steam power plants operate at supercritical pressures (P > 22.06 MPa) and have thermal efficiencies of about 40% for fossil-fuel plants and 34% for nuclear plants

EXAMPLE: EFFECT OF BOILER PRESSURE AND TEMPERATURE ON THERMAL EFFICIECNY

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.



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

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

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = 3.02\text{ kJ/kg}$$

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

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

$$\text{State 4:}\quad P_4 = 10\text{ kPa}\quad (\text{sat. mixture})$$

$$s_4 = s_3$$

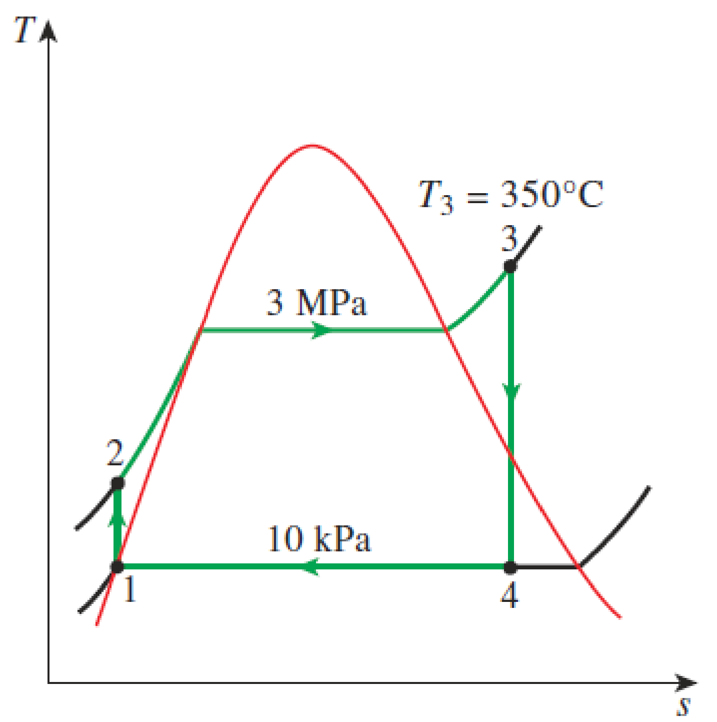
$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 0.6492}{7.4996} = 0.8128$$

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

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

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

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



(a)

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

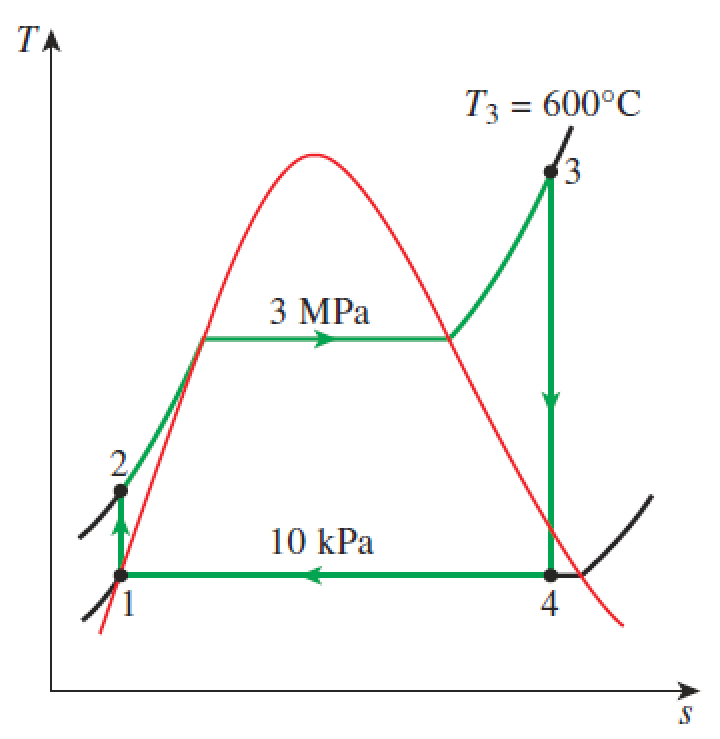
State 2: $P_2 = 3 \text{ MPa}$

$s_2 = s_1$

steam is superheated to 600°C instead of 350°C

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = 3.02 \text{ kJ/kg}$$

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



(b)

States 3 and 4 are different

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

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

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

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

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

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).

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

boiler pressure is raised to 15 MPa from 3 MPa while the turbine inlet temperature is maintained at 600°C

States 2, 3 and 4 are different

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

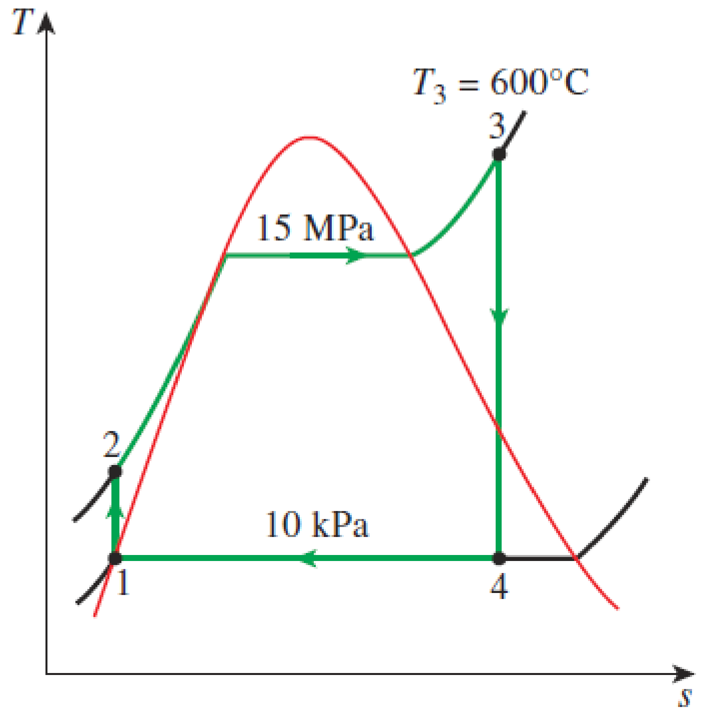
$$h_3 = 3583.1\text{ kJ/kg}$$

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

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

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

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



(c)

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).

THE IDEAL REHEAT RANKINE CYCLE

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?

1. Superheat the steam to very high temperatures before it enters the turbine. However, it is not a viable solution as 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 (**reheat**). Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern power plants.

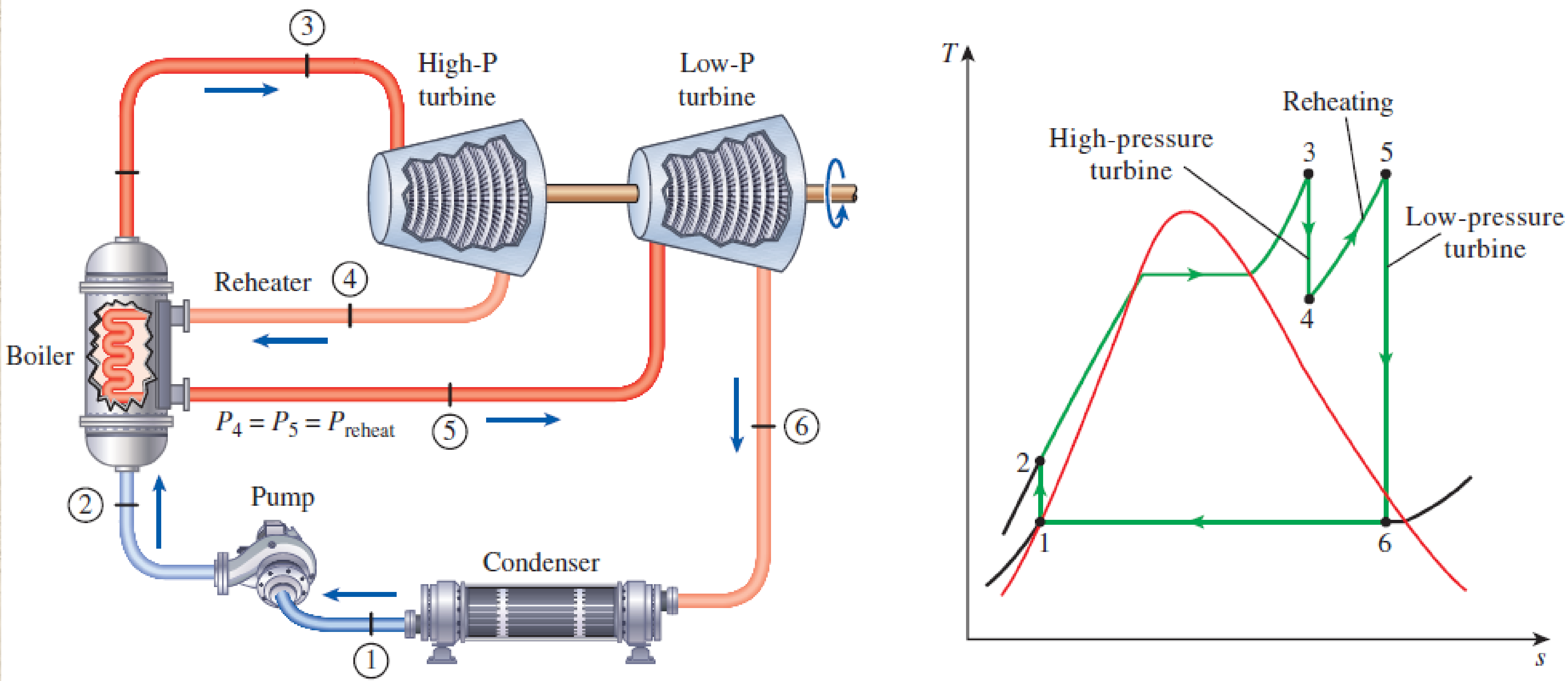


FIGURE 10–11
The ideal reheat Rankine cycle.

$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4)$$

$$w_{turb,out} = w_{turb,I} + w_{turb,II} = (h_3 - h_4) + (h_5 - h_6)$$

- The single reheat in a modern power plant improves the cycle efficiency by 4 to 5% by increasing the average temperature at which heat is transferred to the steam
- As the number of stages is increased, the expansion and reheat processes approach an isothermal process at the maximum temperature
- The use of more than two reheat stages, however, is not practical as the theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat

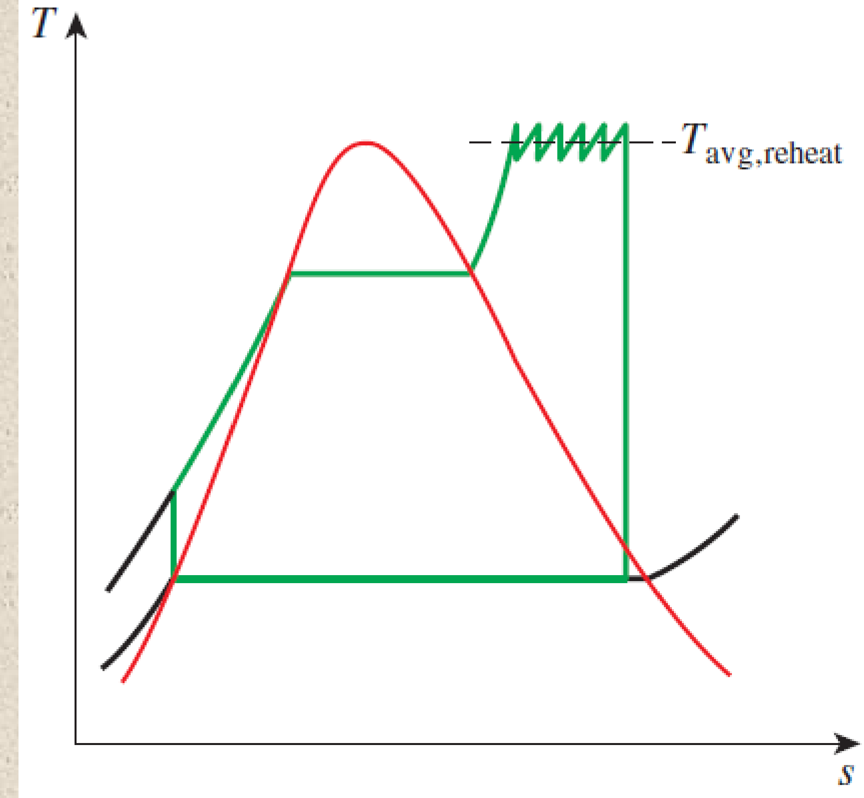
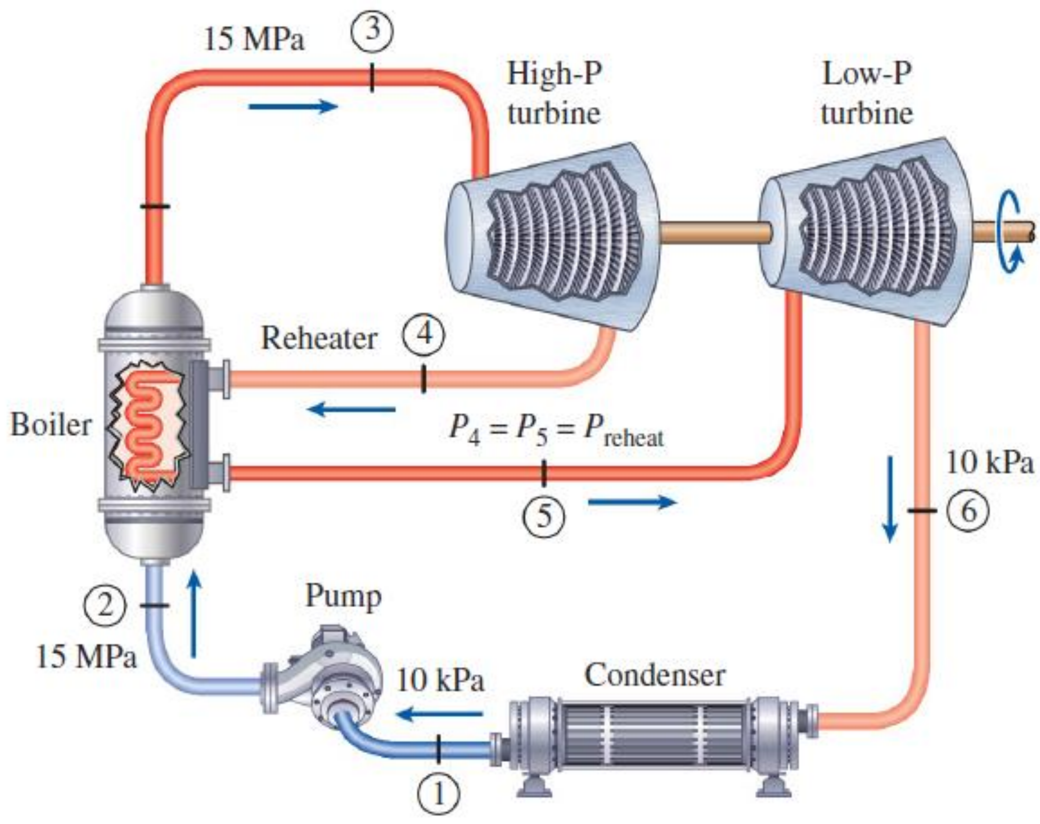


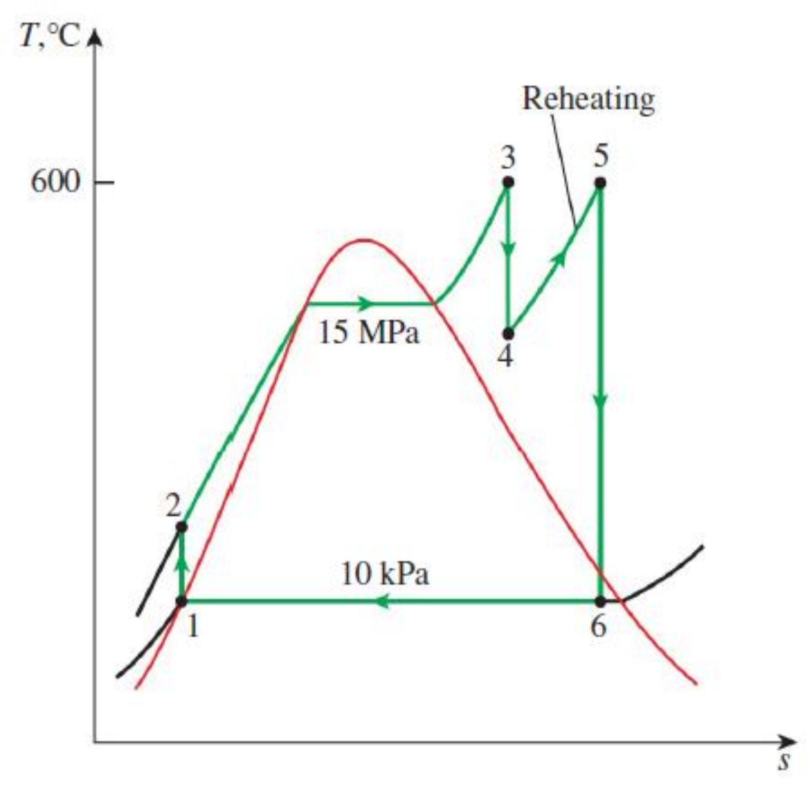
FIGURE 10–12
The average temperature at which heat is transferred during reheating increases as the number of reheat stages is increased.

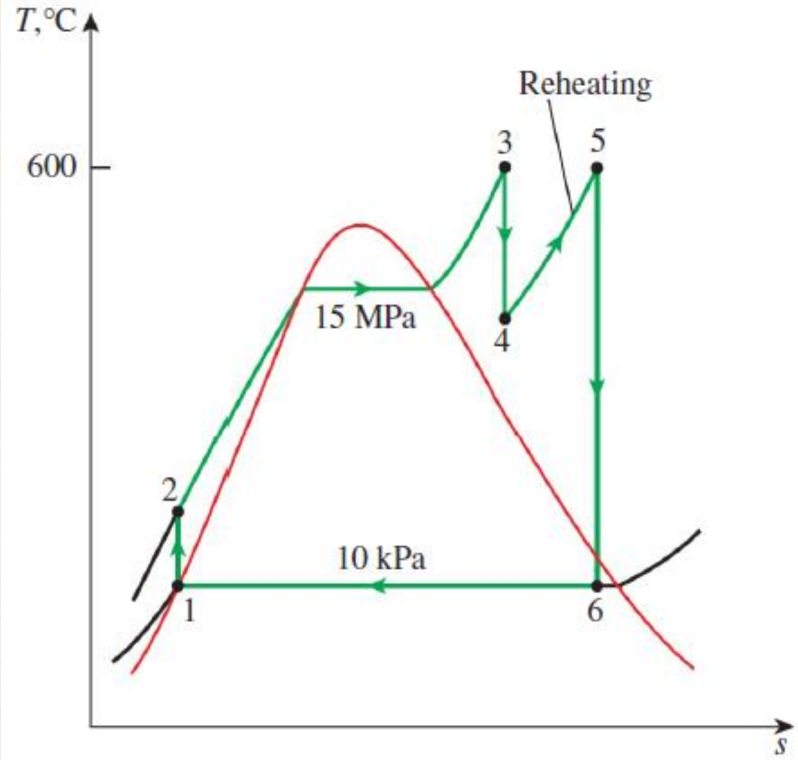
The optimum reheat pressure is about one-fourth of the maximum cycle pressure

EXAMPLE: 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







State 6:

$P_6 = 10$ kPa

$x_6 = 0.896$ (sat. mixture)

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

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

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

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

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

State 3: $\left.\begin{array}{l} P_3 = 15 \text{ MPa} \\ T_3 = 600°\text{C} \end{array}\right\}$ $h_3 = 3583.1$ kJ/kg, $s_3 = 6.6796$ kJ/kg·K

State 2: $P_2 = 15$ MPa

$s_2 = s_1$

$w_{\text{pump,in}} = v_1(P_2 - P_1) = 15.14$ kJ/kg

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

State 4: $\left.\begin{array}{l} P_4 = 4 \text{ MPa} \\ s_4 = s_3 \end{array}\right\}$ $h_4 = 3155.0$ kJ/kg, $(T_4 = 375.5°\text{C})$

Thus

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

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

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

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

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

and

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

Discussion This problem was solved previously 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.

THE IDEAL REGENERATIVE RANKINE CYCLE

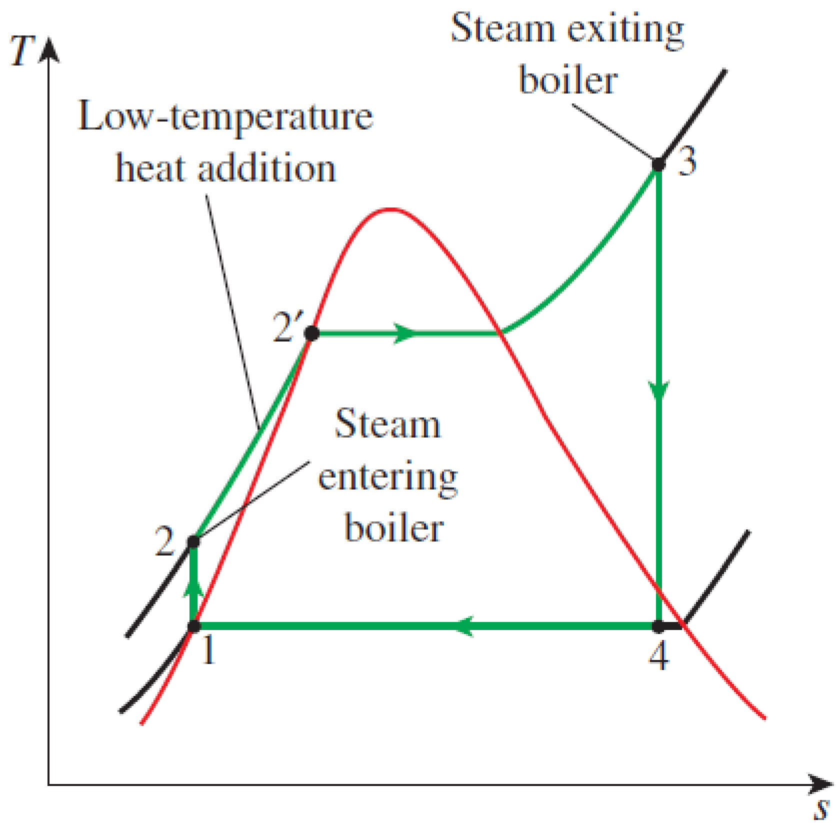


FIGURE 10–14

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

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.

In steam power plants, steam is extracted 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).**

FEED WATER HEATER

- 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**
- 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.
- 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.

$$q_{in} = h_5 - h_4$$

$$q_{out} = (1 - y)(h_7 - h_1)$$

$$w_{turb,out} = (h_5 - h_6) + (1 - y)(h_6 - h_7)$$

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

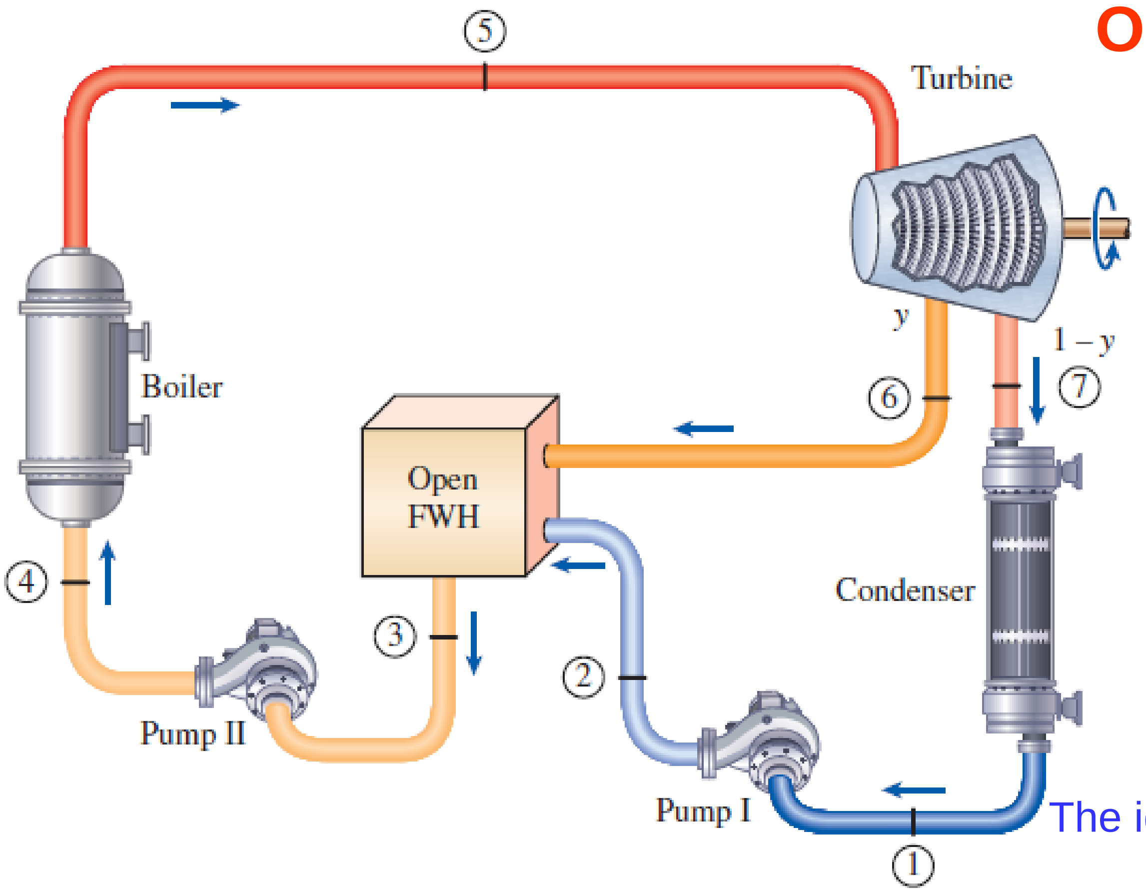
(fraction of steam extracted)

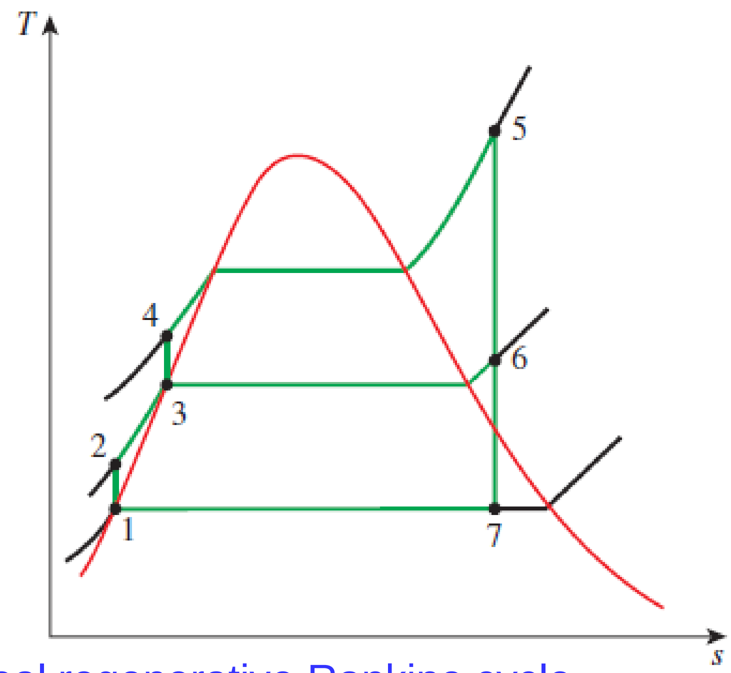
$$y = \dot{m}_6/\dot{m}_5$$

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

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

Open Feedwater Heaters





The ideal regenerative Rankine cycle with an open feedwater heater.

Closed Feedwater Heaters

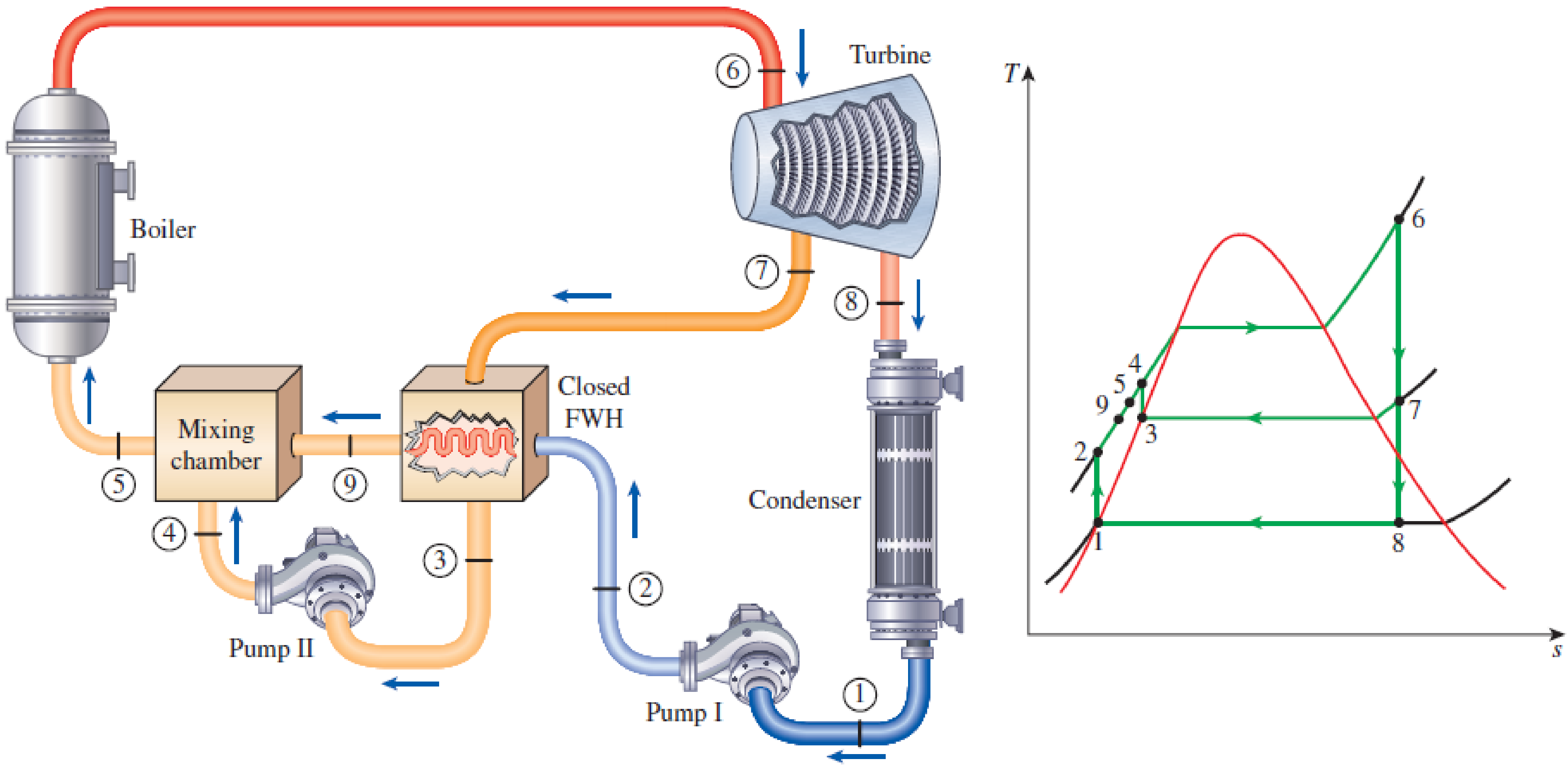


FIGURE 10–16
The ideal regenerative Rankine cycle with a closed feedwater heater.

Closed Feedwater Heaters

- 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 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
- Open feedwater heaters are simple and inexpensive and have good heat transfer characteristics. For each heater, however, a pump is required to handle the feedwater
- Most steam power plants use a combination of open and closed feedwater heaters

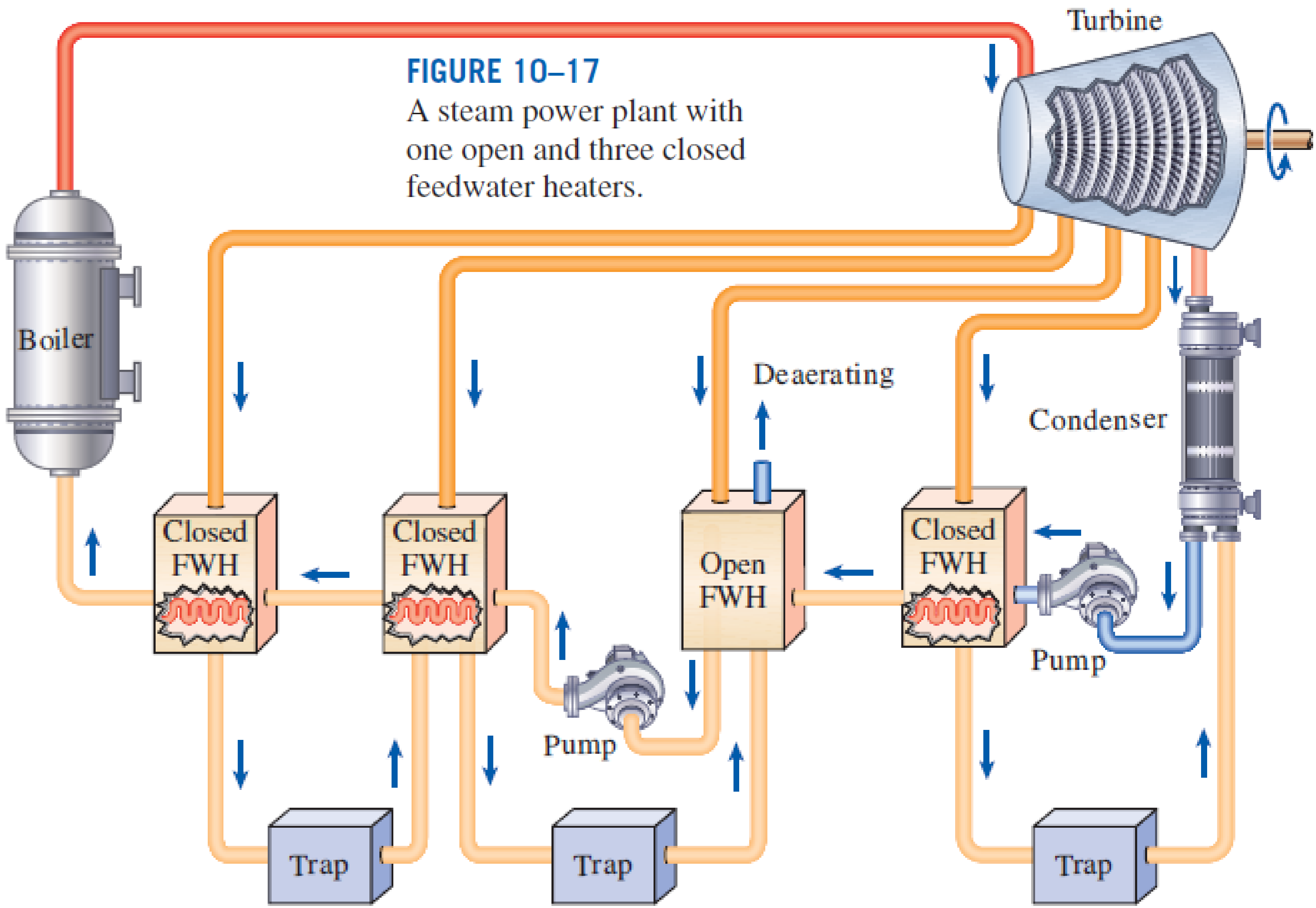
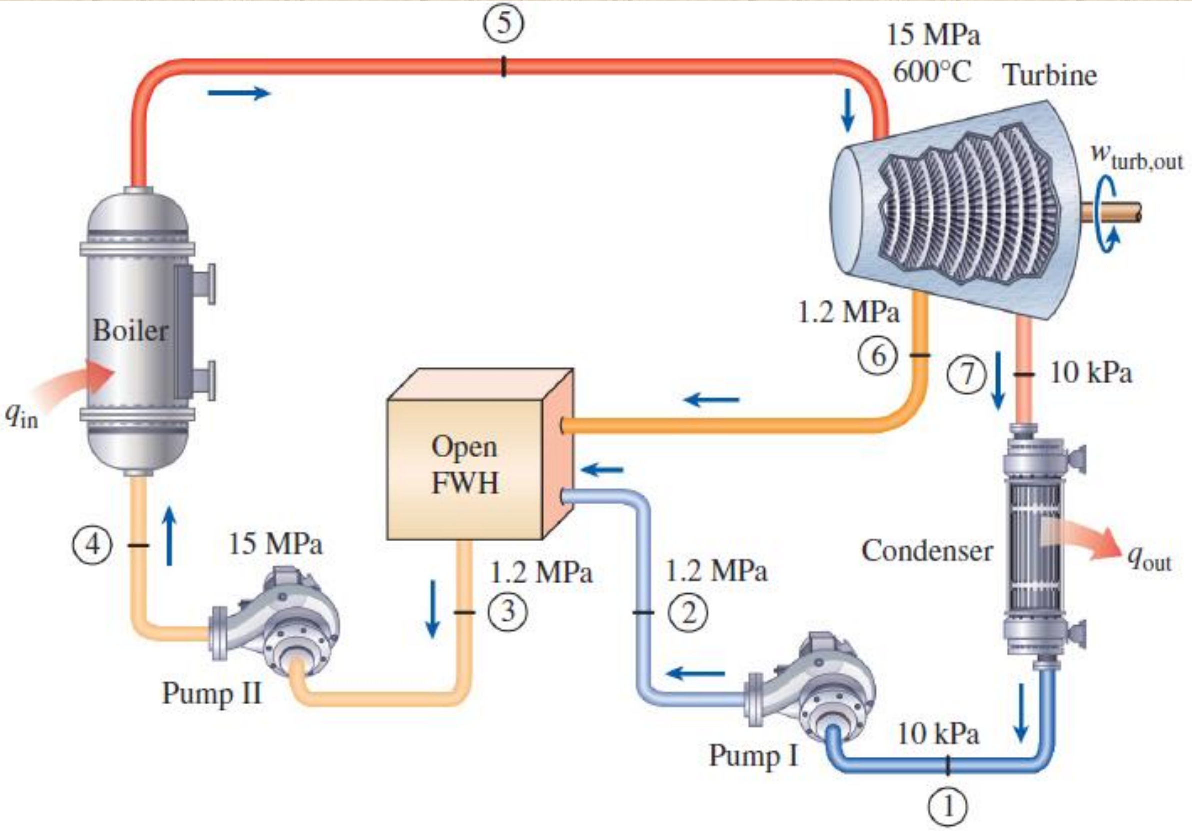
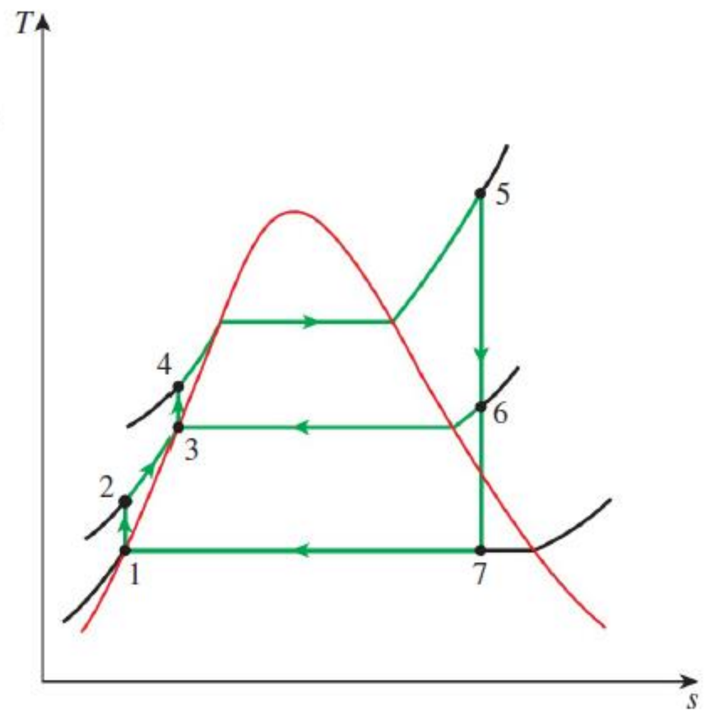


FIGURE 10–17
A steam power plant with one open and three closed feedwater heaters.

EXAMPLE: 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





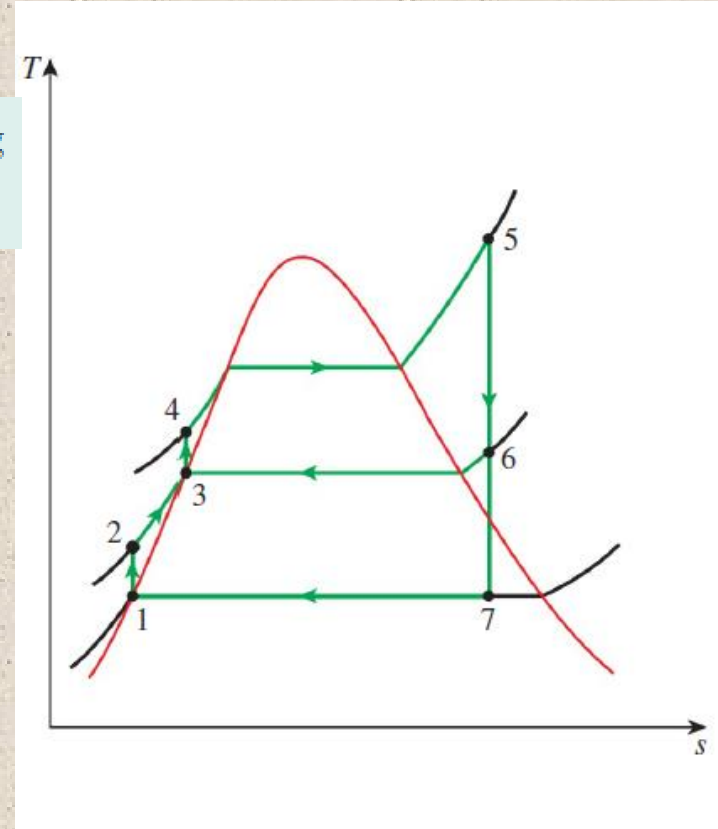
State 1: $\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\}$ $h_1 = h_{f\,@\,10\text{ kPa}} = 191.81 \text{ kJ/kg}$
$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101 \text{ m}^3\text{/kg}$

State 2: $P_2 = 1.2 \text{ MPa}$
$s_2 = s_1$

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

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

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



State 3: $\left.\begin{array}{l} P_3 = 1.2 \text{ MPa} \\ \text{Sat. liquid} \end{array}\right\}$ $\begin{array}{l} v_3 = v_{f\,@\,1.2\text{ MPa}} = 0.001138 \text{ m}^3\text{/kg} \\ h_3 = h_{f\,@\,1.2\text{ MPa}} = 798.33 \text{ kJ/kg} \end{array}$

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

$s_4 = s_3$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3) = 15.70 \text{ kJ/kg}$$

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

State 5: $\left.\begin{array}{l} P_5 = 15 \text{ MPa} \\ T_5 = 600°\text{C} \end{array}\right\}$ $\begin{array}{l} h_5 = 3583.1 \text{ kJ/kg} \\ s_5 = 6.6796 \text{ kJ/kg·K} \end{array}$

State 6: $\left.\begin{array}{l} P_6 = 1.2 \text{ MPa} \\ s_6 = s_5 \end{array}\right\}$ $\begin{array}{l} h_6 = 2860.2 \text{ kJ/kg} \\ (T_6 = 218.4°\text{C}) \end{array}$

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

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

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

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

$$\dot{E}_{in} = \dot{E}_{out} \longrightarrow \sum_{in} \dot{m}h = \sum_{out} \dot{m}h$$

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

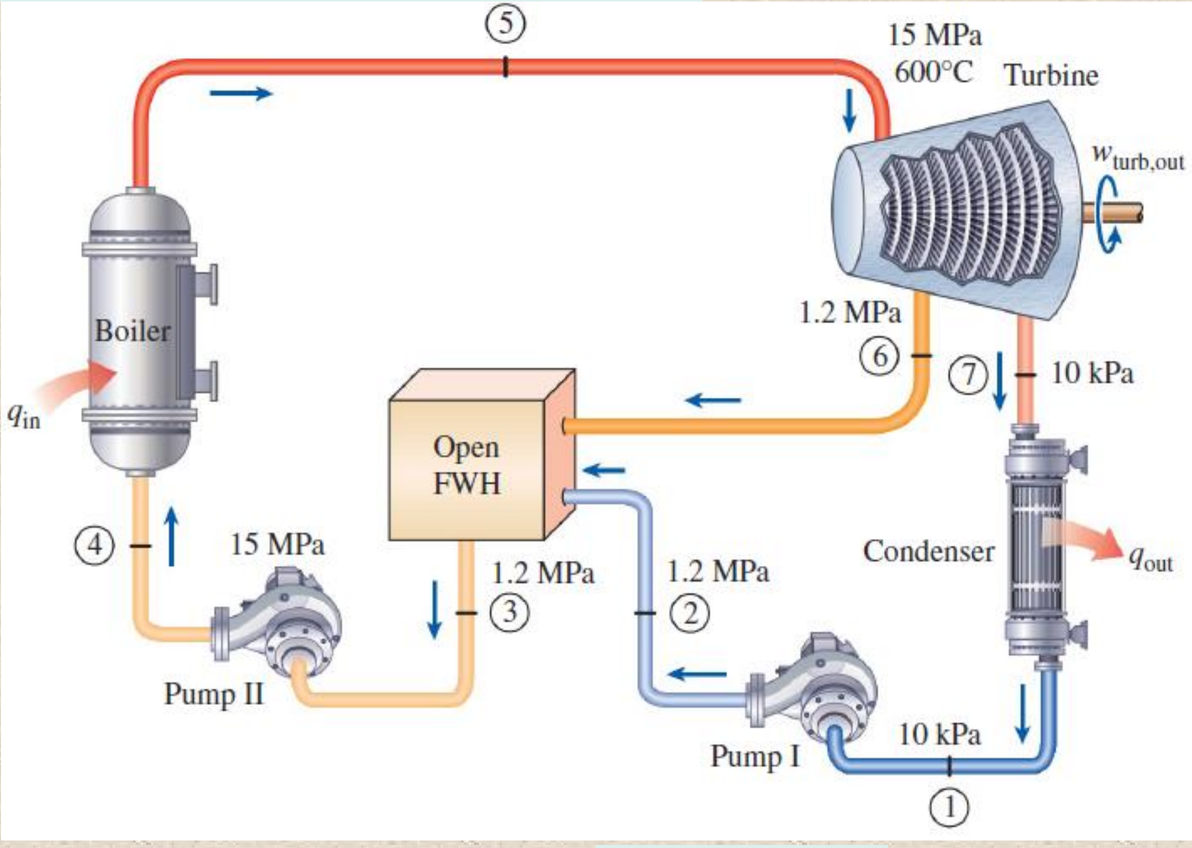
$$y = \frac{h_3 - h_2}{h_6 - h_2} = \mathbf{0.2270}$$

y is the fraction of steam extracted from the turbine $(=\dot{m}_6/\dot{m}_5)$

$$q_{in} = h_5 - h_4 = 2769.1 \text{ kJ/kg}$$

$$q_{out} = (1 - y)(h_7 - h_1) = 1486.9 \text{ kJ/kg}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = \mathbf{0.463 \text{ or } 46.3\%}$$



Discussion This problem was worked out in previously 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.

SECOND-LAW ANALYSIS OF VAPOR POWER CYCLES

- The ideal Carnot cycle is ***totally reversible*****;** thus they do 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 where to start improvements

EXERGY DESTRUCTION IN A STEADY FLOW SYSTEM

Rate of exergy destruction for a steady-flow system

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

Steady-flow, one-inlet, one-exit, unit-mass basis

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

Subscripts *i* and *e* denote the inlet and exit states, respectively

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

EXERGY DESTRUCTION OF A CYCLE

Exergy destruction (unit-mass basis) for a **cycle** that involves heat transfer only with a source at T_H and a sink at T_L:

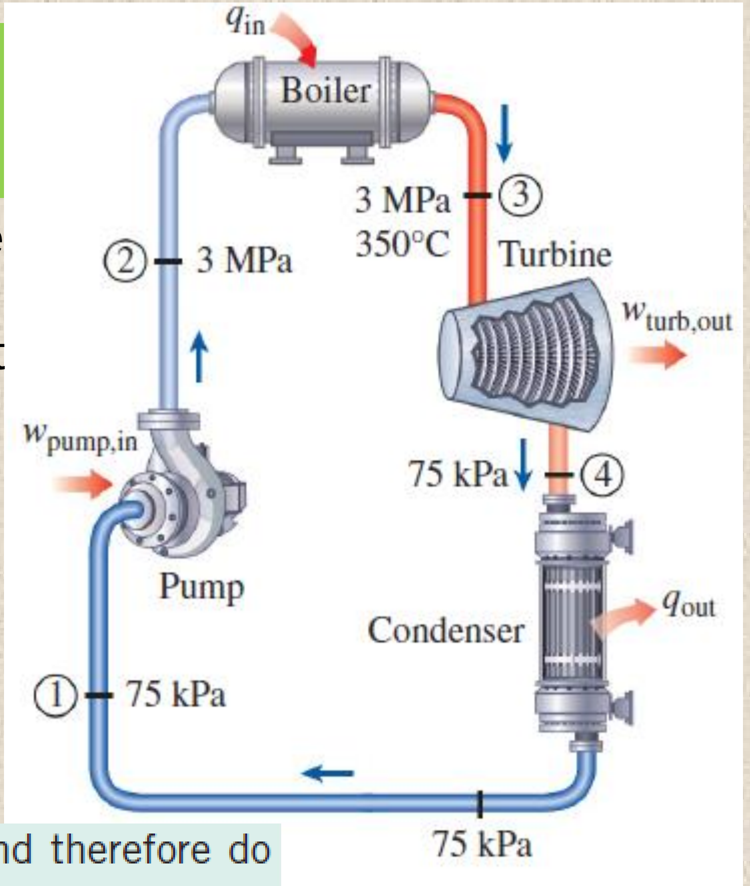
$$x_{dest} = T_0\left(\frac{q_{out}}{T_L} - \frac{q_{in}}{T_H}\right) \qquad \text{(kJ/kg)}$$

Exergy of a fluid stream ψ:

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

EXAMPLE: SECOND-LAW ANALYSIS OF AN 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. Heat is supplied to the steam in a furnace maintained at 800 K, and waste heat is rejected to the surroundings at 300 K. Determine (*a*) the exergy destruction associated with each of the four processes and the whole cycle and (*b*) the second-law efficiency of this cycle.



(*a*) 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,

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

Processes 2-3 and 4-1

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

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

$q_{in,23}$ = 2729 kJ/kg

$q_{out,41}$ = 2019 kJ/kg

$$x_{\text{dest},23} = T_0\left(s_3 - s_2 - \frac{q_{\text{in},23}}{T_{\text{source}}}\right) = \mathbf{636\ kJ/kg}$$

$$x_{\text{dest},41} = T_0\left(s_1 - s_4 + \frac{q_{\text{out},41}}{T_{\text{sink}}}\right) = \mathbf{360\ kJ/kg}$$

$$x_{\text{dest,cycle}} = x_{\text{dest},12} + x_{\text{dest},23} + x_{\text{dest},34} + x_{\text{dest},41} = \mathbf{996\ kJ/kg}$$

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy expended}} = \frac{x_{recovered}}{x_{expended}} = 1 - \frac{x_{destroyed}}{x_{expended}}$$

$$x_{heat,in} = \left(1 - \frac{T_0}{T_H}\right) q_{in} = \left(1 - \frac{300\text{ K}}{800\text{ K}}\right)(2729\text{ kJ/kg}) = 1706\text{ kJ/kg}$$

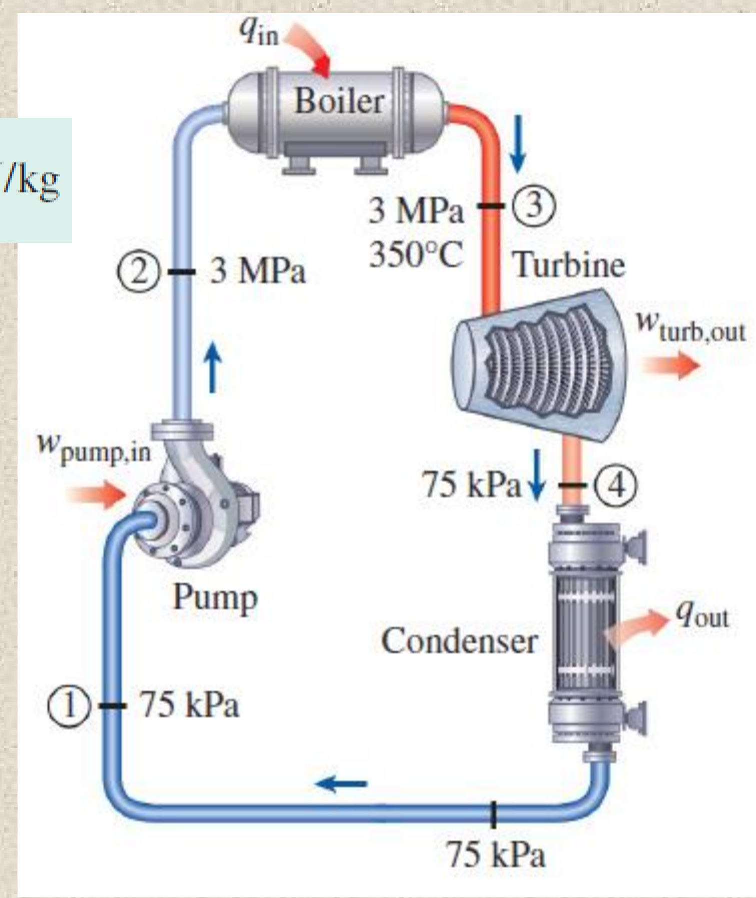
$$x_{expended} = x_{heat,in} = 1706\text{ kJ/kg}$$

$w_{pump,in}$ = 3.0 kJ/kg

$w_{turb,out}$ = 713 kJ/kg

$$w_{net} = w_{turb,out} - w_{pump,in} = 710\text{ kJ/kg}$$

$$x_{recovered} = w_{net} = 710\text{ kJ/kg}$$



$$\eta_{II} = \frac{x_{recovered}}{x_{expended}} = \frac{710\text{ kJ/kg}}{1706\text{ kJ/kg}} = \mathbf{0.416} \text{ or } \mathbf{41.6\%}$$

$$\eta_{II} = 1 - \frac{x_{destroyed}}{x_{expended}} = 1 - \frac{996\text{ kJ/kg}}{1706\text{ kJ/kg}} = \mathbf{0.416} \text{ or } \mathbf{41.6\%}$$

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} \quad \text{(heat engines)} = \frac{w_{net}/q_{in}}{1-T_{sink}/T_{source}} = \frac{710/2729}{1-300/800} = \mathbf{0.416 \text{ or } 41.6\%}$$

COGENERATION

- Many industries require energy input in the form of heat, called ***process heat***. Process heat in these industries is usually supplied by steam at 5 to 7 atm and 150 to 200°C
- Energy is usually transferred to the steam by burning coal, oil, natural gas, or another fuel in a furnace

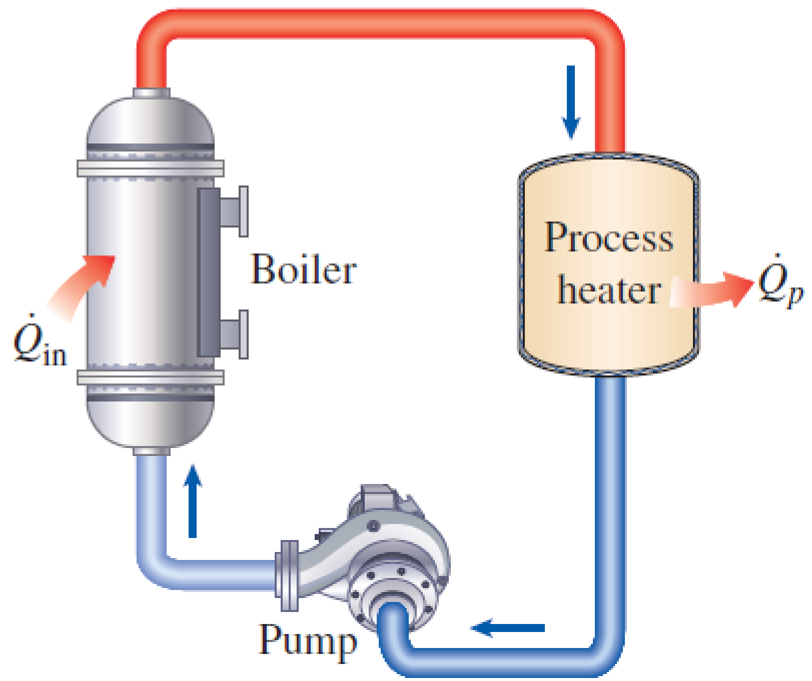


FIGURE 10–21
A simple process-heating plant.

- The temperature in the furnaces is typically very high (around 1400°C), 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, as loss in exergy

COGENERATION

- Industries that use large amounts of process heat also consume a large amount of electric power
- It makes 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 (cogeneration plant)

Cogeneration: The production of more than one useful form of energy (such as process heat and electric power) from the same energy source

Utilization factor

$$\epsilon_u = \frac{\text{Net power output} + \text{Process heat delivered}}{\text{Total heat input}} = \frac{\dot{W}_{\text{net}} + \dot{Q}_p}{\dot{Q}_{\text{in}}}$$

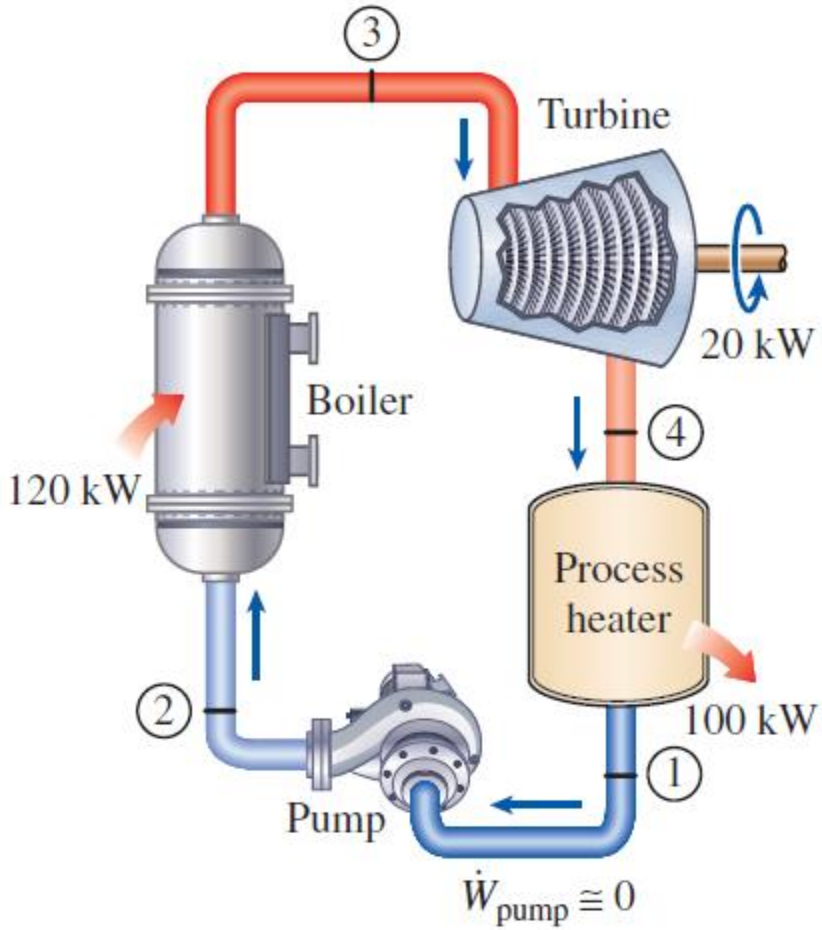


FIGURE 10–22
An ideal cogeneration plant.

$$\epsilon_u = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}}$$

- The utilization factor of the ideal steam-turbine cogeneration plant is 100%
- Actual cogeneration plants have utilization factors as high as 80%
- Ideal steam-turbine cogeneration plant is not practical because it cannot adjust to the variations in power and process-heat loads

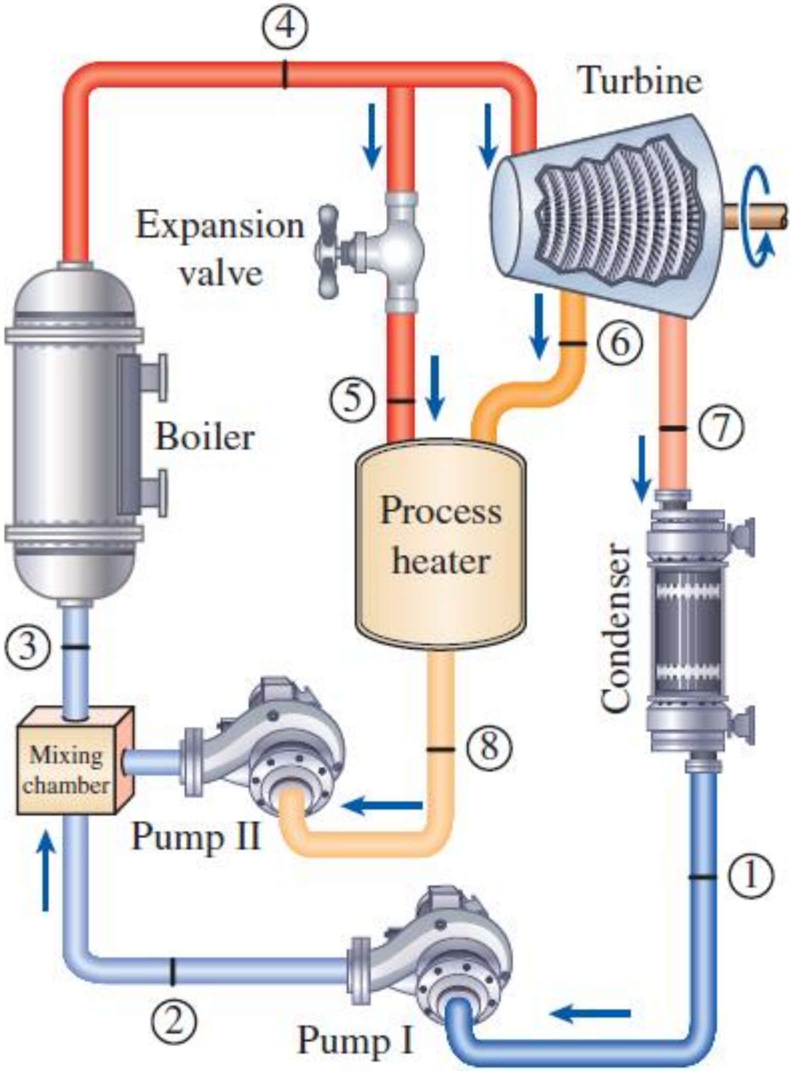


FIGURE 10–23
A cogeneration plant with adjustable loads.

- 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 it is then cooled at constant pressure

$$\dot{Q}_{\text{in}} = \dot{m}_3(h_4 - h_3)$$
$$\dot{Q}_{\text{out}} = \dot{m}_7(h_7 - h_1)$$
$$\dot{Q}_p = \dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_8 h_8$$
$$\dot{W}_{\text{turb}} = (\dot{m}_4 - \dot{m}_5)(h_4 - h_6) + \dot{m}_7(h_6 - h_7)$$

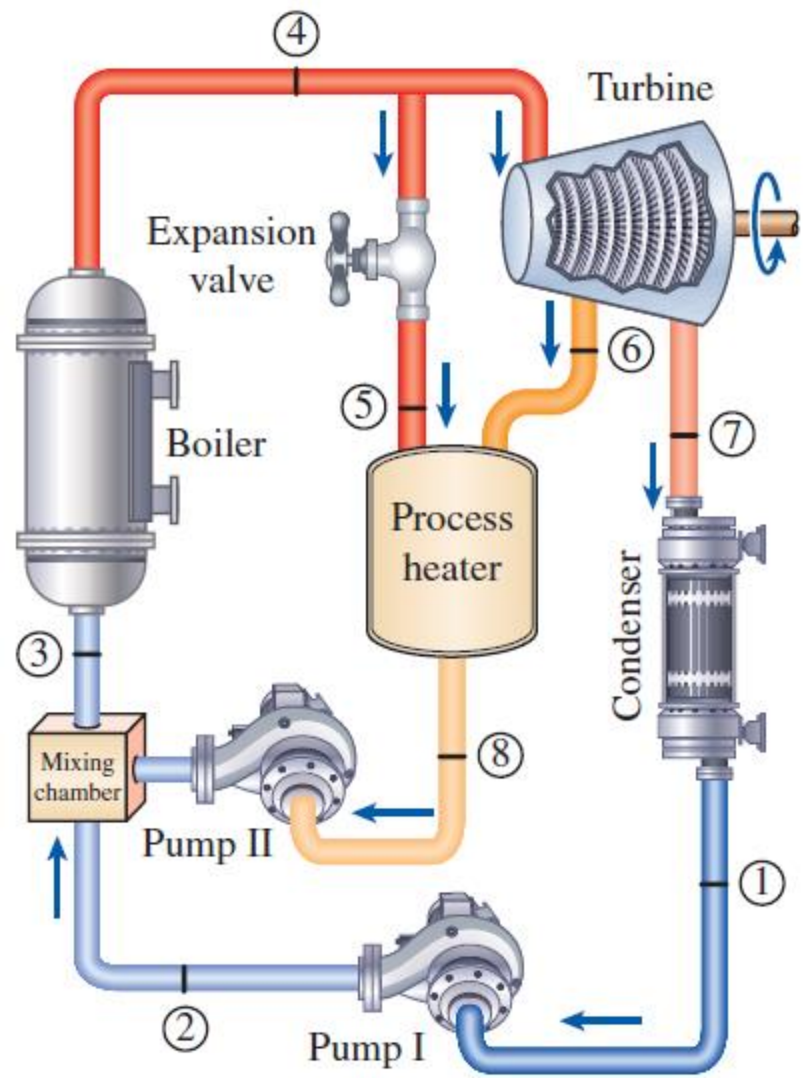


FIGURE 10–23
A cogeneration plant with adjustable loads.

- At times of high demand for process heat, all the steam is routed to the process-heating units and none to the condenser ($m_7 = 0$). 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 ($m_5 = m_4$). No power is produced in this mode.

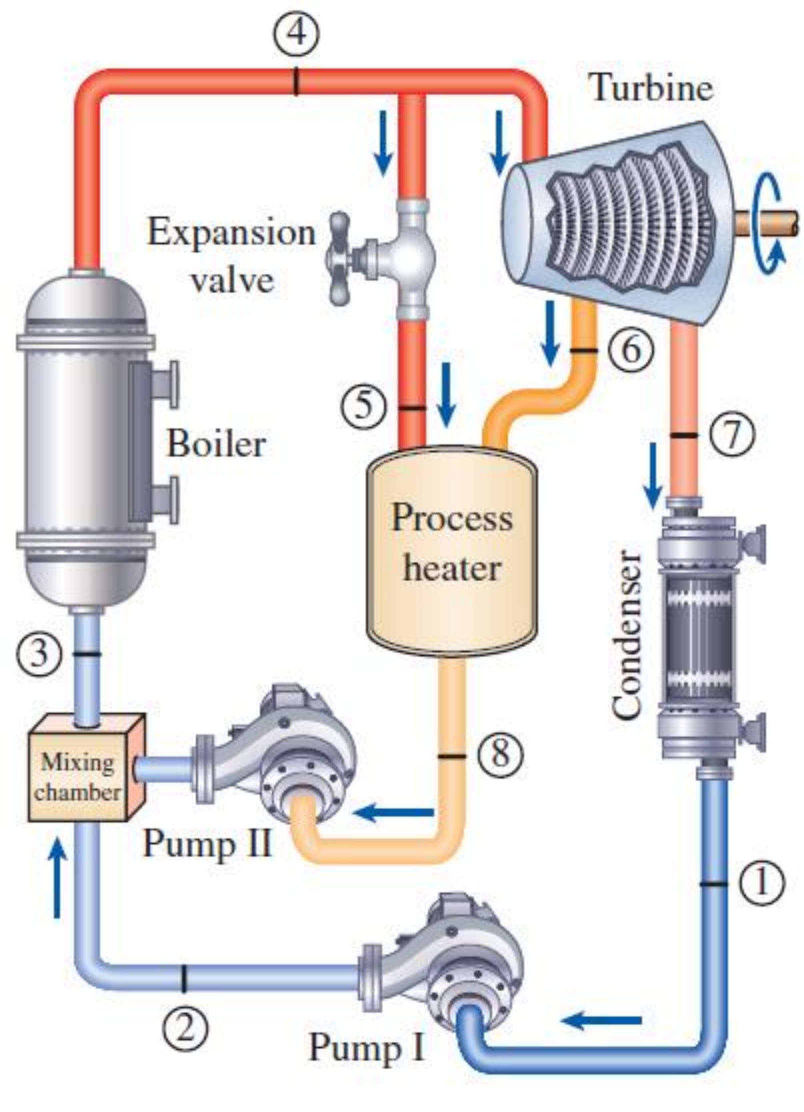


FIGURE 10–23

A cogeneration plant with adjustable loads.

- When there is no demand for process heat, all the steam passes through the turbine and the condenser ($m_5 = m_6 = 0$), and the cogeneration plant operates as an ordinary steam power plant.

COMBINED GAS–VAPOR POWER CYCLES

- A popular modification to increase the thermal efficiency involves a gas power cycle topping a vapor power cycle, which is called the **combined gas–vapor cycle**, or just the **combined cycle**
- The maximum fluid temperature at the turbine inlet is about 620°C for modern steam power plants, and over 1425°C for gas turbine power plants. The gases leaves the gas turbine at very high temperatures (usually above 500°C)
- 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
- Some recent combined-cycle power plants have achieved efficiencies above 60%

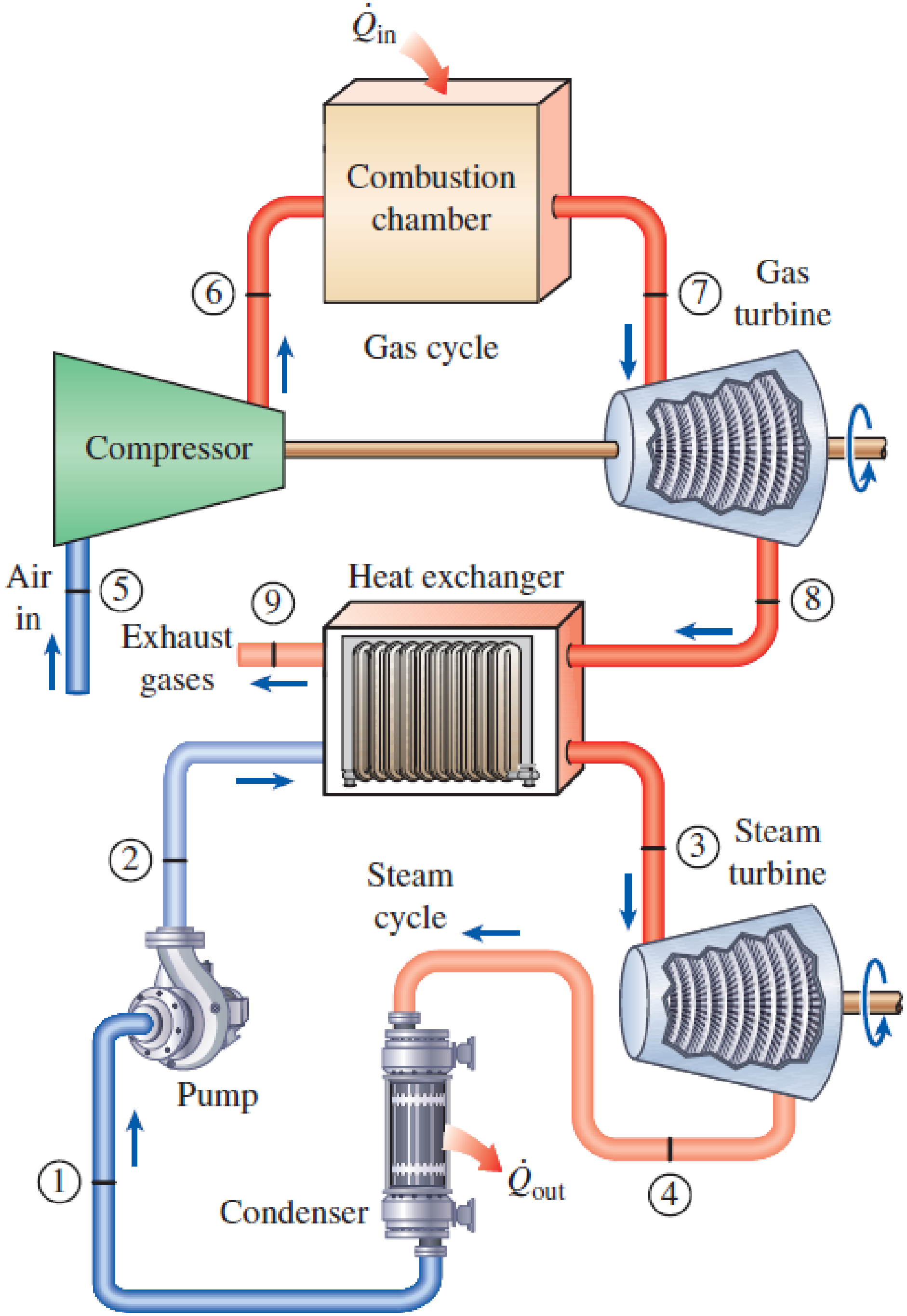
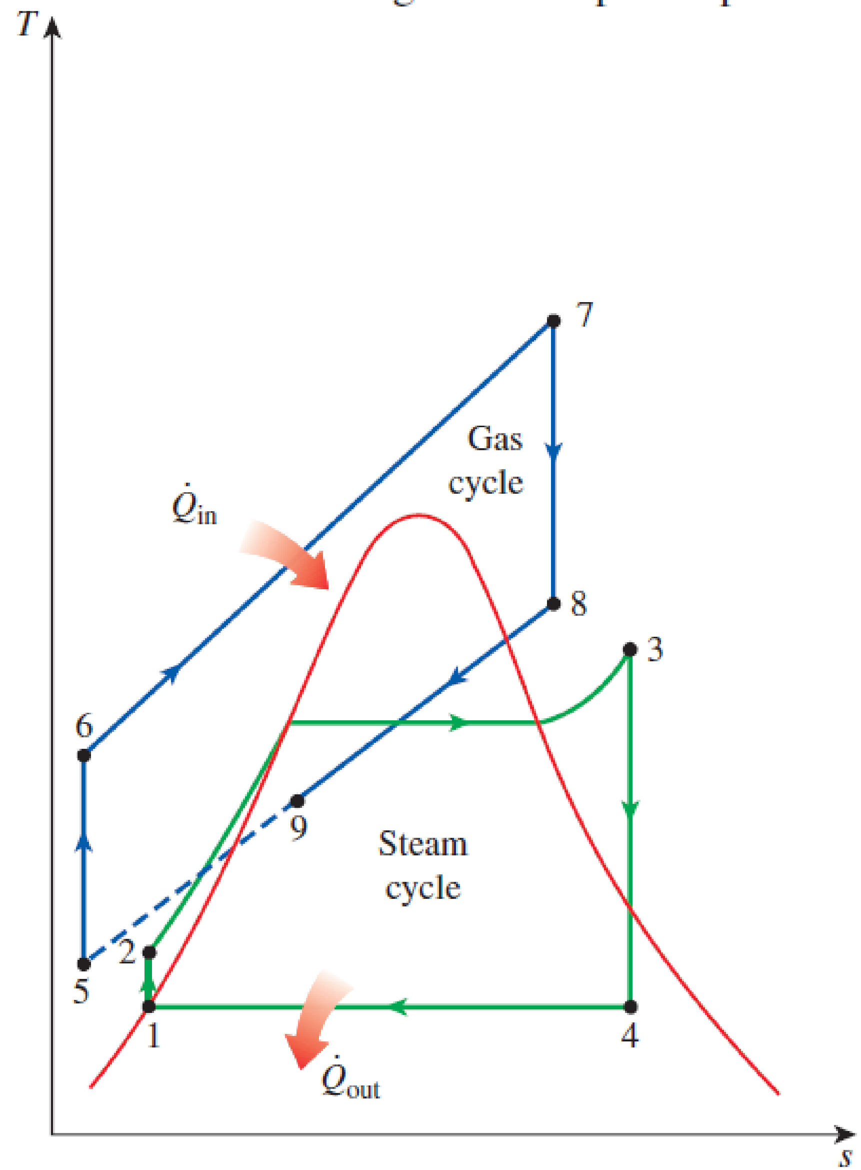


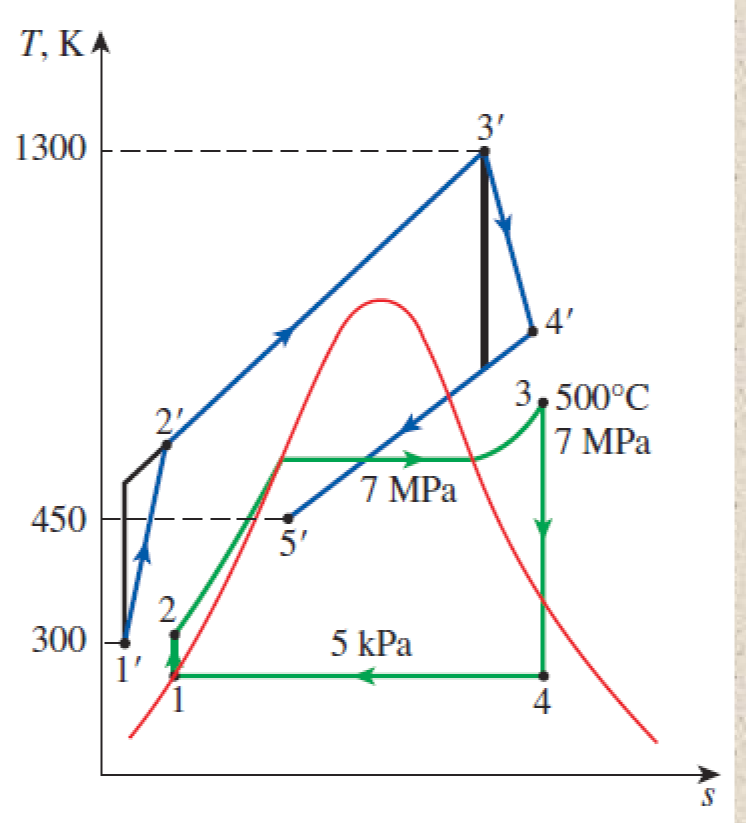
FIGURE 10–25

Combined gas–steam power plant.



EXAMPLE: A COMBINED GAS–STEAM POWER CYCLE

Consider the combined gas–steam power cycle shown. 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.



Gas cycle:

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

$$q_{\text{in}} = 790.58 \text{ kJ/kg} \quad w_{\text{net}} = 210.41 \text{ kJ/kg} \quad \eta_{\text{th}} = 26.6\%$$

$$h_5' = h_{@\,450\text{ K}} = 451.80 \text{ kJ/kg}$$

Steam cycle:

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

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

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

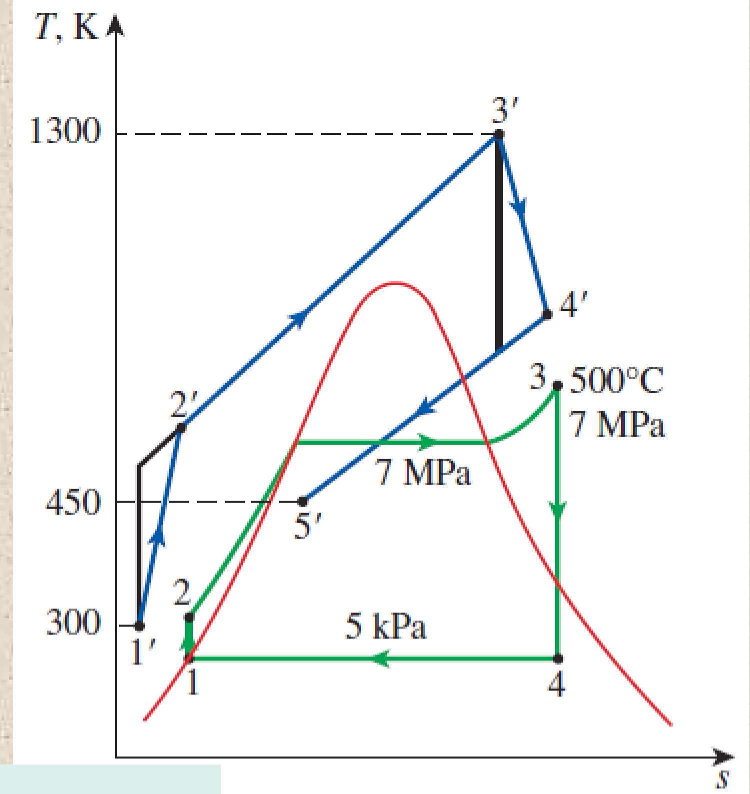
$$\dot{E}_{in} = \dot{E}_{out}$$

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

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

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

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



$$w_{net} = w_{net,gas} + y w_{net,steam}$$
$$= (210.41 \text{ kJ/kg gas}) + (0.131 \text{ kg steam/kg gas})(1331.4 \text{ kJ/kg steam})$$
$$= 384.8 \text{ kJ/kg gas}$$

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

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.

Summary

- The Carnot vapor cycle
- Rankine cycle: The ideal cycle for vapor power cycles
  - ✓ Energy analysis of the ideal Rankine cycle
- Deviation of actual vapor power cycles from idealized ones
- How can we increase the efficiency of the Rankine cycle?
  - ✓ Lowering the condenser pressure (*Lowers* $T_{low,avg}$)
  - ✓ Superheating the steam to high temperatures (*Increases* $T_{high,avg}$)
  - ✓ Increasing the boiler pressure (*Increases* $T_{high,avg}$)
- The ideal reheat Rankine cycle
- The ideal regenerative Rankine cycle
  - ✓ Open feedwater heaters
  - ✓ Closed feedwater heaters
- Second-law analysis of vapor power cycles
- Cogeneration
- Combined gas–vapor power cycles