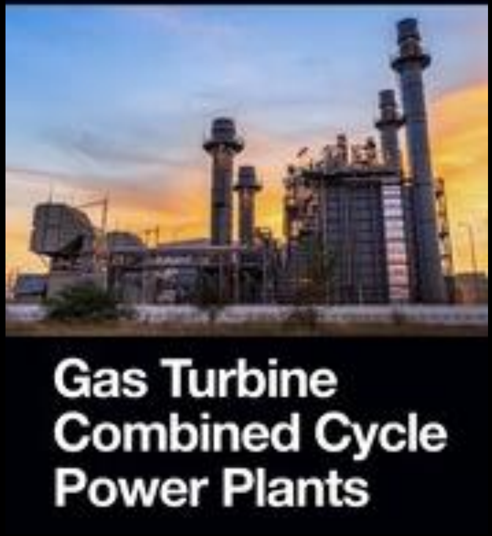
CHAPTER

10

Vapor and Combined Power Cycles

1. The Carnot Vapor Cycle
2. Rankine Cycle : The Ideal Cycle for Vapor Power Cycles
3. Deviation of Actual Vapor Power Cycles
4. How Can We Increase The Efficiency of Cycle
5. The Ideal Reheat Rankine Cycle
6. The Ideal Regenerative Rankine Cycle
7. Second-Law Analysis of Vapor Power Cycles
8. Cogeneration
9. Combined Gas-Vapor Power Cycles



2022 Fall semester

Objectives

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

10-1. THE CARNOT VAPOR CYCLE

1-2 isothermal heat addition in a boiler
2-3 isentropic expansion in a turbine
3-4 isothermal heat rejection in a condenser
4-1 isentropic compression in a compressor

The Carnot cycle is the most efficient cycle operating between two specified temperature limits but it is not a suitable model for power cycles. Because:

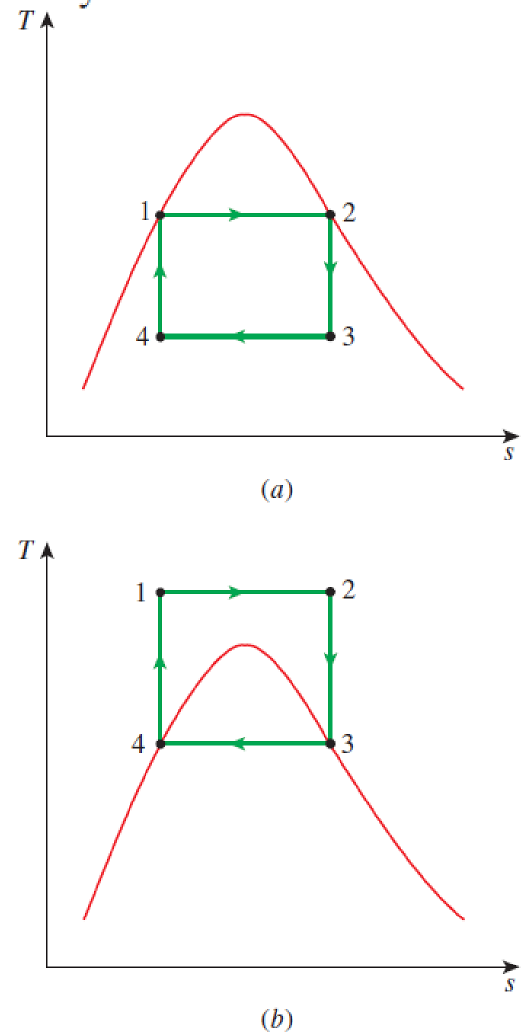
Process 1-2 Limiting the heat transfer processes to two-phase systems severely limits the maximum temperature that can be used in the cycle (374°C for water)

Process 2-3 The turbine cannot handle steam with a high moisture content 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 cycle in (*b*) is not suitable since it requires isentropic compression to extremely high pressures and isothermal heat transfer at variable pressures.

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



10-2. RANKINE CYCLE: THE IDEAL CYCLE FOR VAPOR POWER CYCLES

2ea Isentropic process
2ea Isobaric process

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.

The cycle that results is the **Rankine cycle**, which is the ideal cycle for vapor power plants.

The ideal Rankine cycle does not involve any internal irreversibilities.

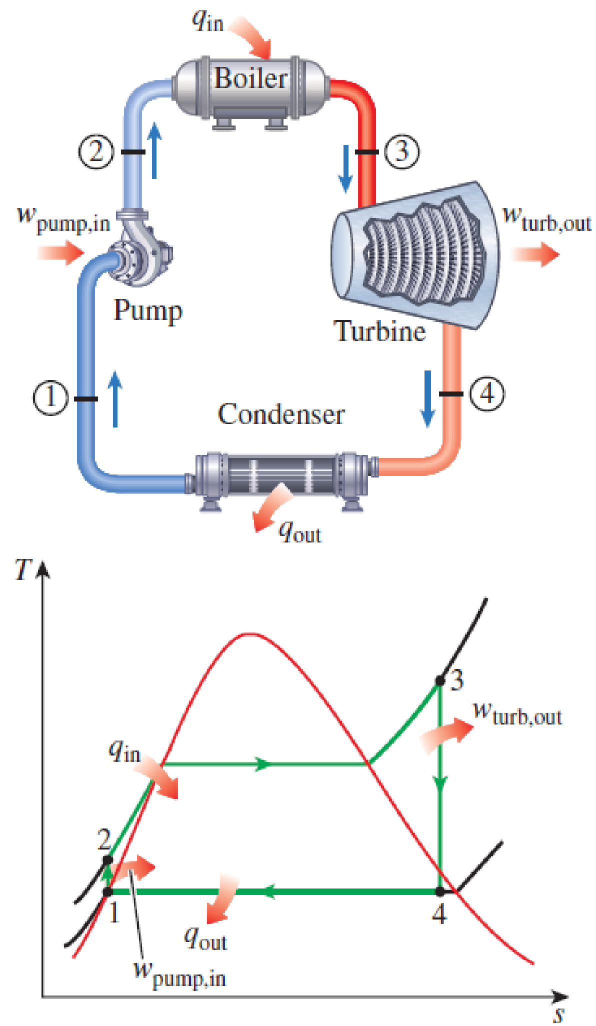
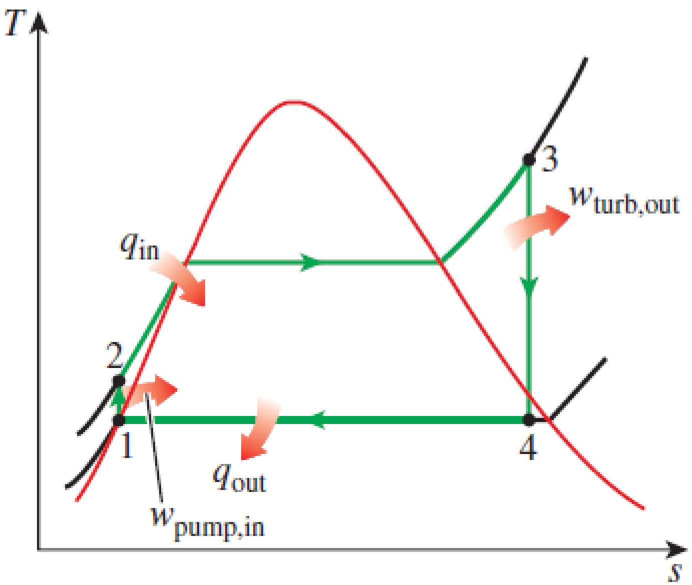


FIGURE 10–2
The simple ideal Rankine cycle.

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

10-2. RANKINE CYCLE: THE IDEAL CYCLE FOR VAPOR POWER CYCLES

Energy Analysis of the Ideal Rankine Cycle



Steady-flow energy equation

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

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

$$w_{pump,in} = \upsilon(P_2 - P_1) \quad (10\text{-}3)$$

$$h_1 = h_{f\,@\,P_1} \quad \text{and} \quad \upsilon \cong \upsilon_1 = \upsilon_{f\,@\,P_1}$$

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

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

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

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

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \quad (10\text{-}8)$$

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.

10-2. RANKINE CYCLE: THE IDEAL CYCLE FOR VAPOR POWER CYCLES

The efficiency of power plants in the U.S. is often expressed in terms of **heat rate**, which is the amount of heat supplied, in Btu's, to generate 1 kWh of electricity.

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

Example 10-1. 단순 이상적 랭킨사이클

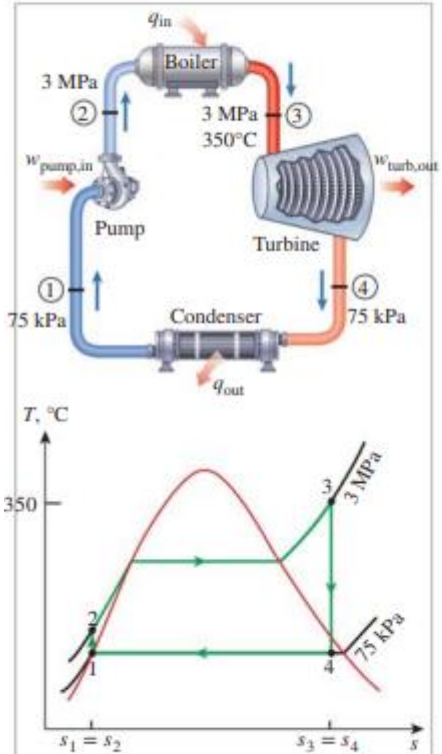


FIGURE 10–3
Schematic and *T-s* diagram for Example 10–1.

EXAMPLE 10–1 The Simple Ideal Rankine Cycle

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

SOLUTION A steam power plant operating on the simple ideal Rankine cycle is considered. The thermal efficiency of the cycle is to be determined.

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

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–3. We note that the power plant operates on the ideal Rankine cycle. Therefore, the pump and the turbine are isentropic, there are no pressure drops in the boiler and condenser, and steam leaves the condenser and enters the pump as saturated liquid at the condenser pressure.

First we determine the enthalpies at various points in the cycle, using data from steam tables (Tables A–4, A–5, and A–6):

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

State 2: $P_2 = 3$ MPa
$$s_2 = s_1$$
$$w_{pump,in} = v_1(P_2 - P_1) = (0.001037 \text{ m}^3\text{/kg})[(3000 - 75) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa·m}^3}\right)$$
$$= 3.03 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{pump,in} = (384.44 + 3.03) \text{ kJ/kg} = 387.47 \text{ kJ/kg}$$

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

State 4: $P_4 = 75$ kPa (sat. mixture)
$$s_4 = s_3$$
$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 1.2132}{6.2426} = 0.8861$$
$$h_4 = h_f + x_4 h_{fg} = 384.44 + 0.8861(2278.0) = 2403.0 \text{ kJ/kg}$$

Thus,
$$q_{in} = h_3 - h_2 = (3116.1 - 387.47) \text{ kJ/kg} = 2728.6 \text{ kJ/kg}$$
$$q_{out} = h_4 - h_1 = (2403.0 - 384.44) \text{ kJ/kg} = 2018.6 \text{ kJ/kg}$$

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

The thermal efficiency could also be determined from
$$w_{turb,out} = h_3 - h_4 = (3116.1 - 2403.0) \text{ kJ/kg} = 713.1 \text{ kJ/kg}$$
$$w_{net} = w_{turb,out} - w_{pump,in} = (713.1 - 3.03) \text{ kJ/kg} = 710.1 \text{ kJ/kg}$$

or
$$w_{net} = q_{in} - q_{out} = (2728.6 - 2018.6) \text{ kJ/kg} = 710.0 \text{ kJ/kg}$$

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

That is, this power plant converts 26 percent of the heat it receives in the boiler to net work. An actual power plant operating between the same temperature and pressure limits will have a lower efficiency because of the irreversibilities such as friction.

Discussion Notice that the back work ratio ($r_{bw} = w_{in}/w_{out}$) of this power plant is 0.004, and thus only 0.4 percent of the turbine work output is required to operate the pump. Having such low back work ratios is characteristic of vapor power cycles. This is in contrast to the gas power cycles, which typically involve very high back work ratios (about 40 to 80 percent).

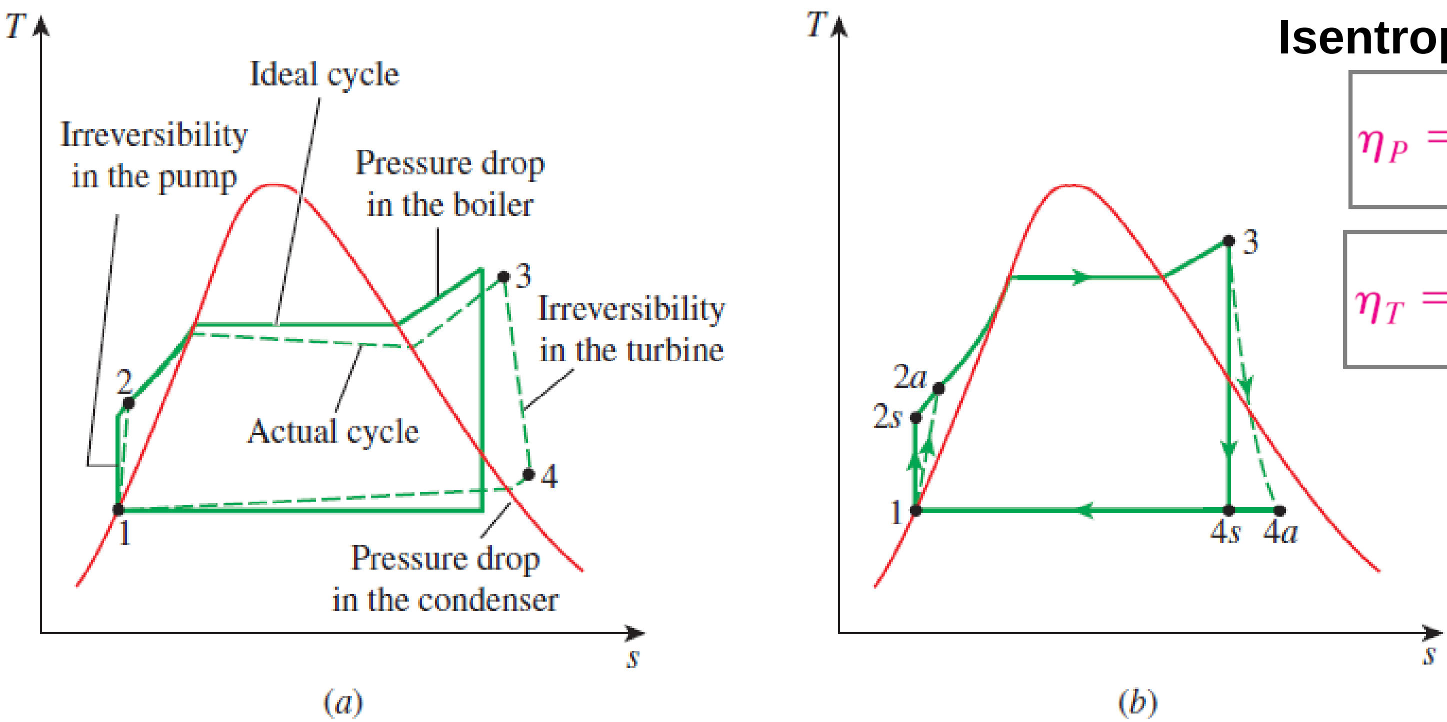
It is also interesting to note the thermal efficiency of a Carnot cycle operating between the same temperature limits
$$\eta_{th,Carnot} = 1 - \frac{T_{min}}{T_{max}} = 1 - \frac{(91.76 + 273) \text{ K}}{(350 + 273) \text{ K}} = 0.415$$

Here T_{min} is taken as the saturation temperature of water at 75 kPa. The difference between the two efficiencies is due to the large external irreversibility in the Rankine cycle caused by the large temperature difference between steam and the heat source.

10-3. 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.



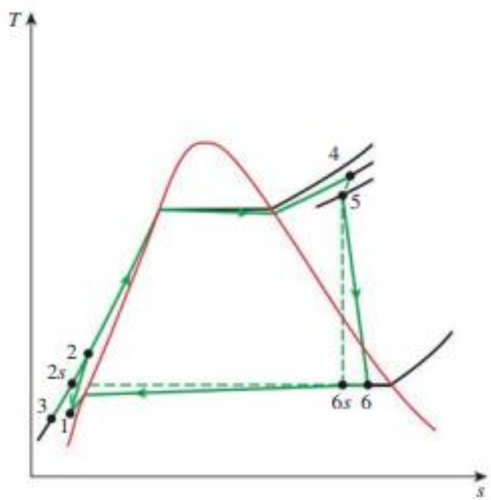
Isentropic efficiencies

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

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

(*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 10-2. 실제 증기동력 사이클



EXAMPLE 10–2 An Actual Steam Power Cycle

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

SOLUTION A steam power cycle with specified turbine and pump efficiencies is considered. The thermal efficiency and the net power output are to be determined.

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

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–5. The temperatures and pressures of steam at various points are also indicated on the figure. We note that the power plant involves steady-flow components and operates on the Rankine cycle, but the imperfections at various components are accounted for.

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

Pump work input:

$$w_{\text{pump,in}} = \frac{w_{s,\text{pump,in}}}{\eta_P} = \frac{\upsilon_1(P_2 - P_1)}{\eta_P}$$

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

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

Turbine work output:

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

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

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

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

Thus,

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

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

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

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

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

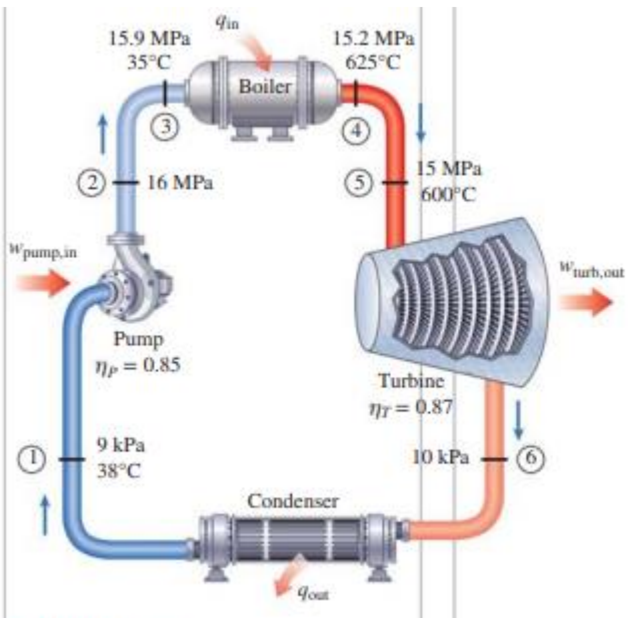


FIGURE 10–5
Schematic and *T-s* diagram for Example 10–2.

10-4. 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

- **Decrease** the average temperature at which heat is rejected from the working fluid in the condenser.

10-4. HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

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

To take advantage of the increased efficiencies at low pressures, 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

Side effect: Lowering the condenser pressure increases the moisture content of the steam at the final stages of the turbine.

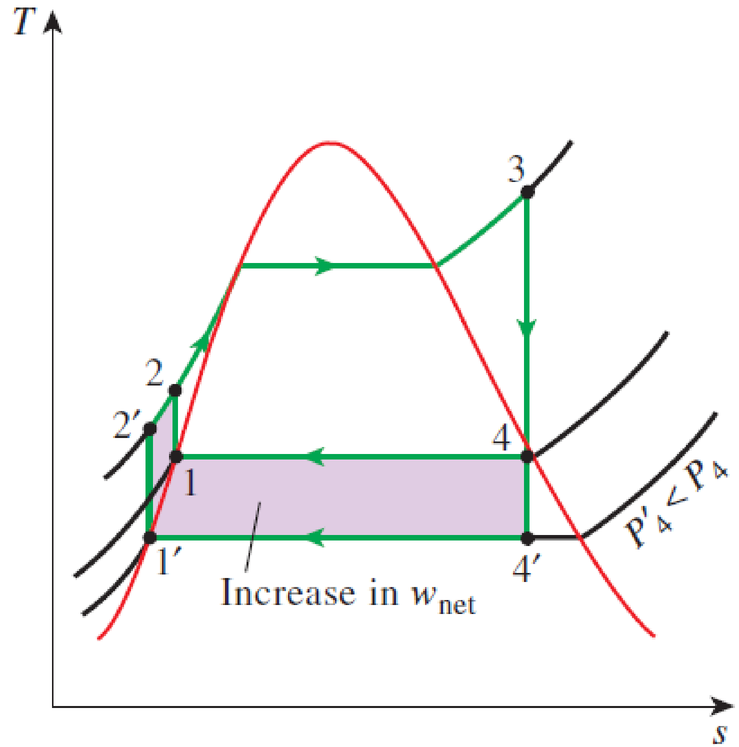


FIGURE 10–6

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

10-4. HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

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

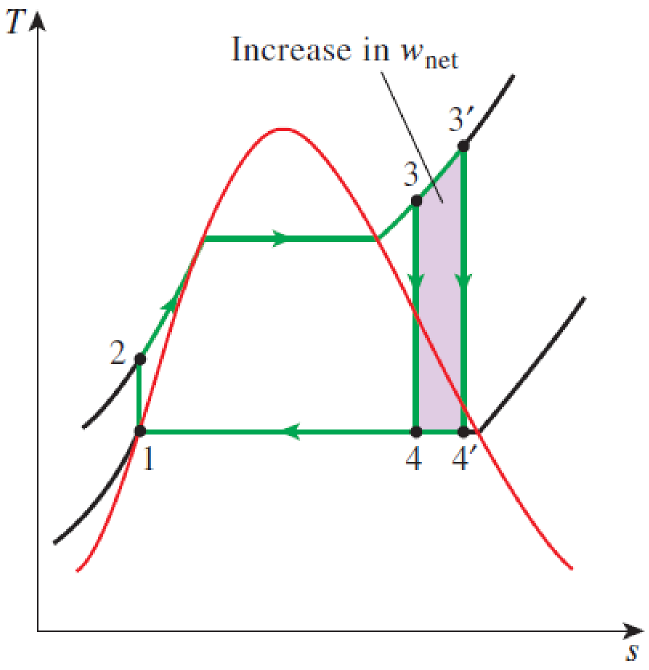


FIGURE 10–7
The effect of superheating the steam to higher temperatures on the ideal Rankine cycle.

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.

10-4. HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

Increasing the Boiler Pressure (*Increases* $T_{high,avg}$)

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.

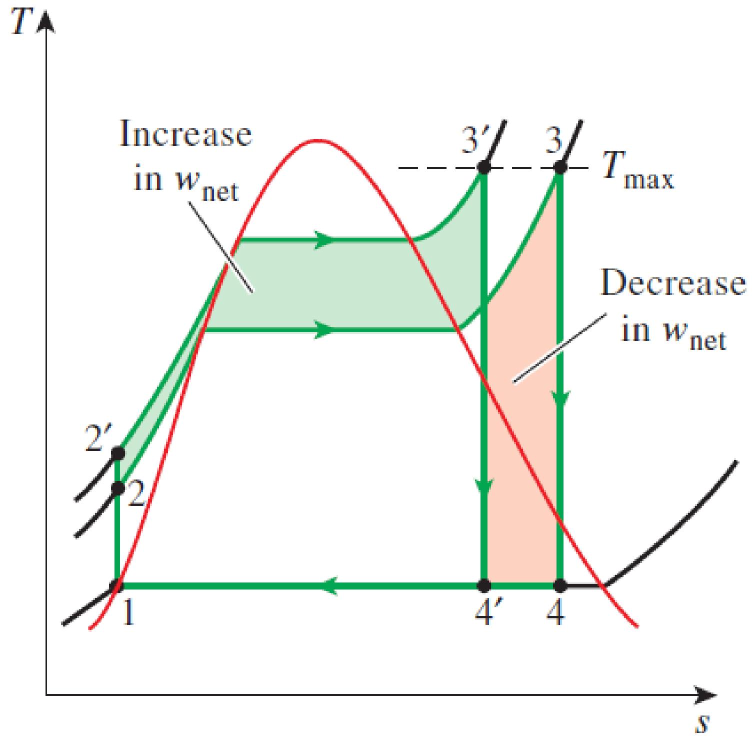


FIGURE 10–8
The effect of increasing the boiler pressure on the ideal Rankine cycle.

10-4. HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

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.

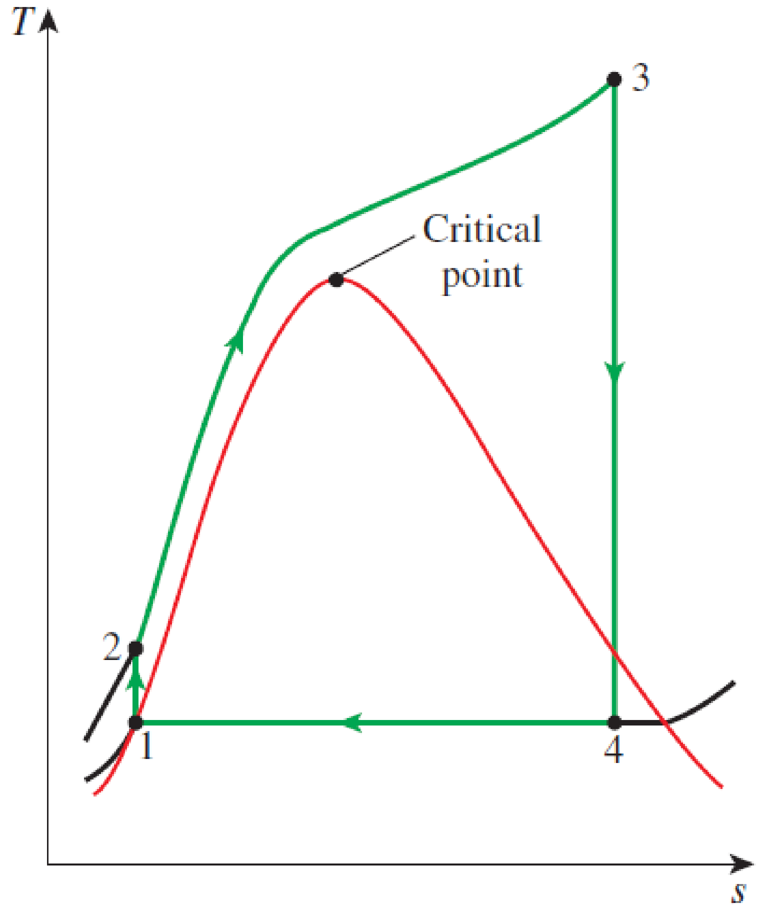


FIGURE 10–9
A supercritical Rankine cycle.

Example 10-3. 보일러의 압력과 온도가 효율에 미치는 영향

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

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

SOLUTION A steam power plant operating on the ideal Rankine cycle is considered. The effects of superheating the steam to a higher temperature and raising the boiler pressure on thermal efficiency are to be investigated.

Analysis The *T-s* diagrams of the cycle for all three cases are given in Fig. 10–10.

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

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

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

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

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

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

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

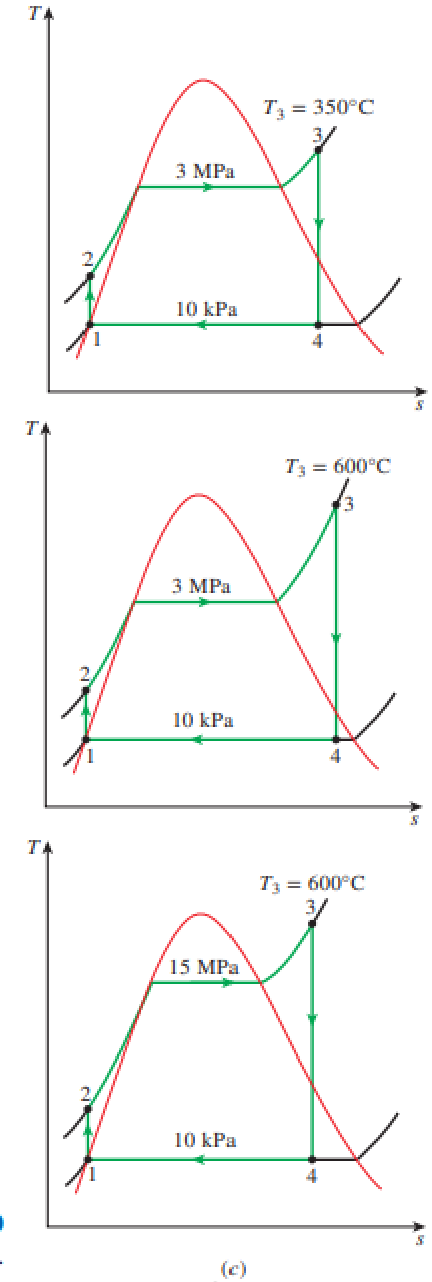


FIGURE 10–10
T-s diagrams of the three cycles discussed in Example 10–3.

State 4: $P_4 = 10$ kPa (sat. mixture)

$$s_4 = s_3$$

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

Thus,

$$h_4 = h_f + x_4 h_{fg} = 191.81 + 0.8128(2392.1) = 2136.1 \text{ kJ/kg}$$
$$q_{in} = h_3 - h_2 = (3116.1 - 194.83) \text{ kJ/kg} = 2921.3 \text{ kJ/kg}$$
$$q_{out} = h_4 - h_1 = (2136.1 - 191.81) \text{ kJ/kg} = 1944.3 \text{ kJ/kg}$$

and

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

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

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

$$h_3 = 3682.8 \text{ kJ/kg}$$
$$h_4 = 2380.3 \text{ kJ/kg} \quad (x_4 = 0.915)$$

Thus,

$$q_{in} = h_3 - h_2 = 3682.8 - 194.83 = 3488.0 \text{ kJ/kg}$$
$$q_{out} = h_4 - h_1 = 2380.3 - 191.81 = 2188.5 \text{ kJ/kg}$$

and

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

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

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

$$h_2 = 206.95 \text{ kJ/kg}$$
$$h_3 = 3583.1 \text{ kJ/kg}$$
$$h_4 = 2115.3 \text{ kJ/kg} \quad (x_4 = 0.804)$$

Thus,

$$q_{in} = h_3 - h_2 = 3583.1 - 206.95 = 3376.2 \text{ kJ/kg}$$
$$q_{out} = h_4 - h_1 = 2115.3 - 191.81 = 1923.5 \text{ kJ/kg}$$

and

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

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

10-5. THE IDEAL REHEAT RANKINE CYCLE

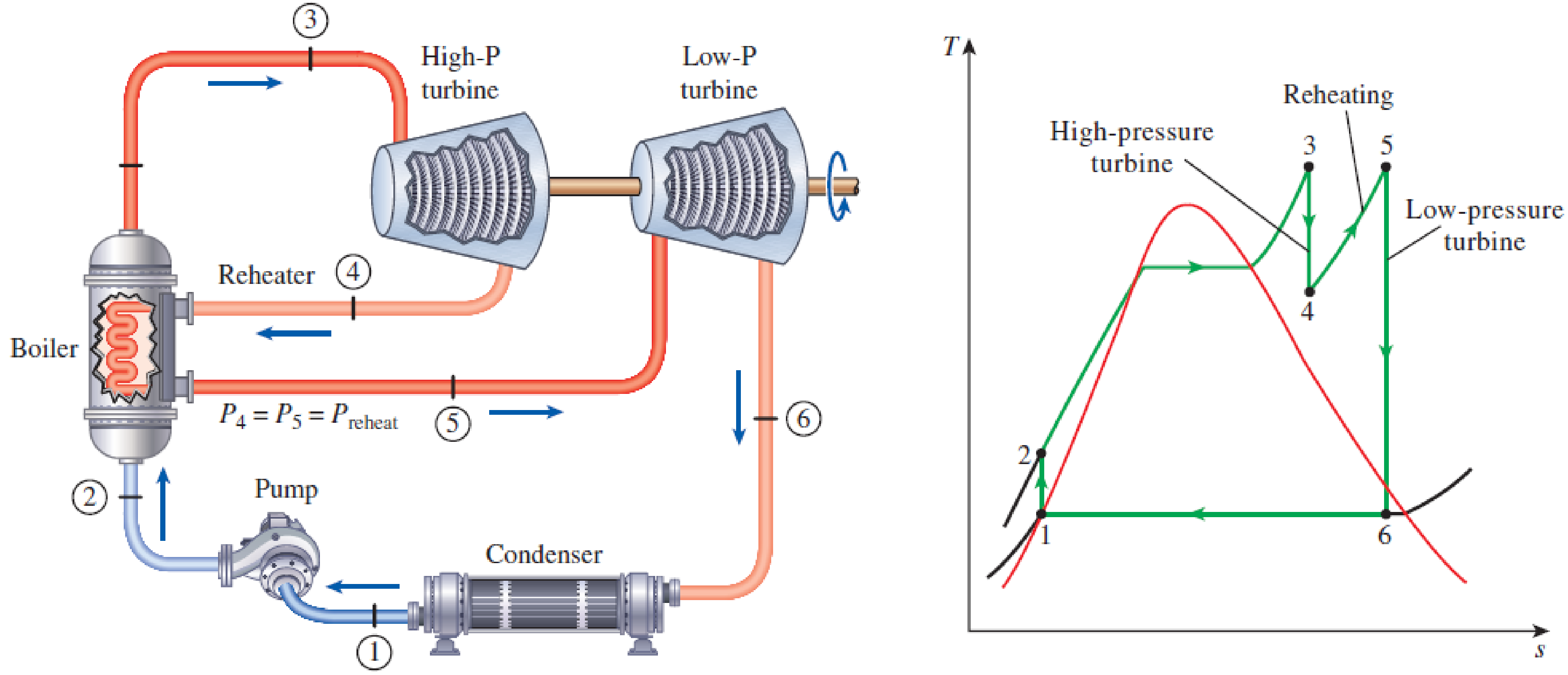
How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?

1. Superheat the steam to very high temperatures. It is limited metallurgically.
2. Expand the steam in the turbine in two stages, and reheat it in between (**reheat**)

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

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

FIGURE 10–11
The ideal reheat Rankine cycle.



10-5. THE IDEAL REHEAT RANKINE CYCLE

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.

The average temperature during the reheat process can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach an isothermal process at the maximum temperature. The use of more than two reheat stages is not practical. The theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat.

The reheat temperatures are very close or equal to the turbine inlet temperature.

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

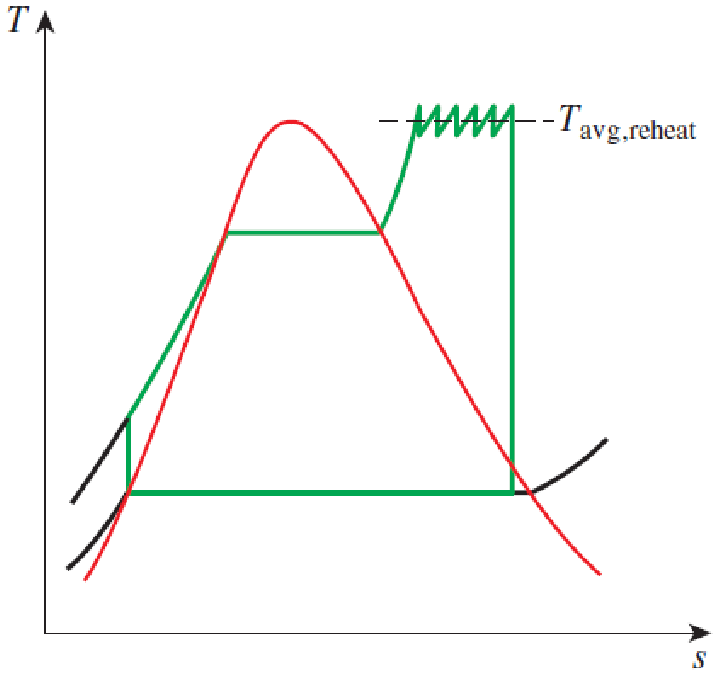


FIGURE 10–12
The average temperature at which heat is transferred during reheating increases as the number of reheat stages is increased.

10-5. THE IDEAL REHEAT RANKINE CYCLE

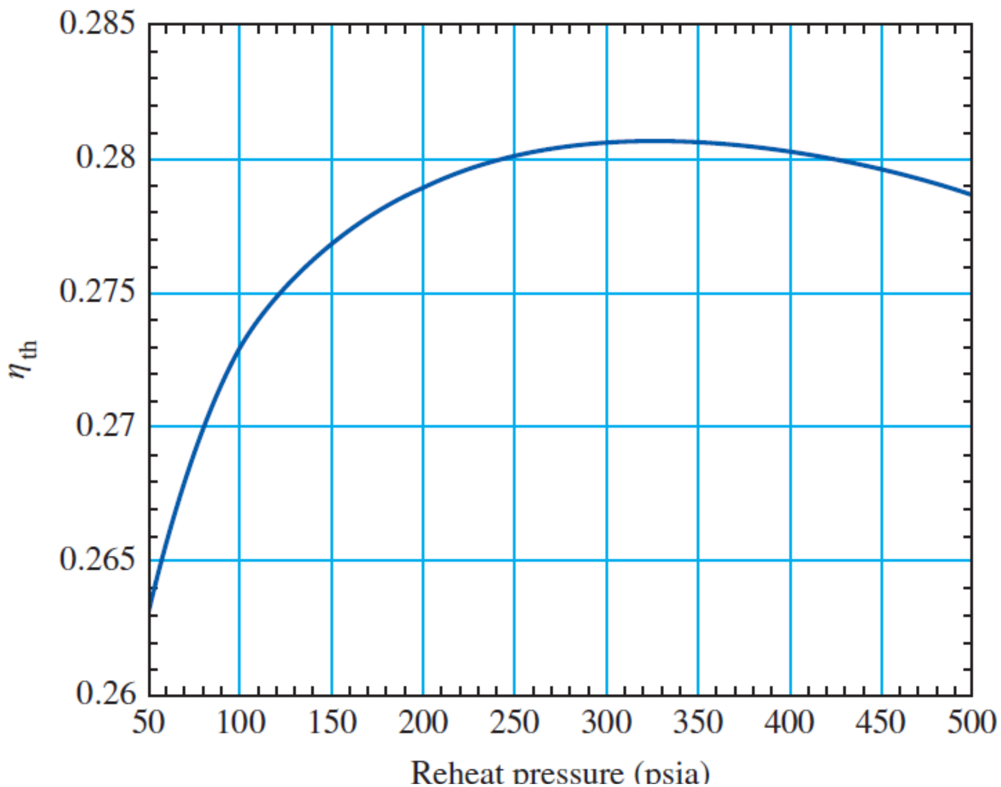


FIGURE 10–14

There is an optimum reheat pressure in the reheat Rankine cycle for which the thermal efficiency is maximum. The values refer to Example 10-4.

Example 10-4. 이상적 재열 랭킨사이클

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

Consider a steam power plant that operates on the ideal reheat Rankine cycle. The plant maintains the inlet of the high-pressure turbine at 600 psia and 600°F, the inlet of the low-pressure turbine at 200 psia and 600°F, and the condenser at 10 psia. The net power produced by this plant is 5000 kW. Determine the rate of heat addition and rejection and the thermal efficiency of the cycle.

Is there any advantage to operating the reheat section of the boiler at 100 psia rather than 200 psia while maintaining the same low-pressure turbine inlet temperature?

SOLUTION An ideal reheat steam Rankine cycle produces 5000 kW of power. The rates of heat addition and rejection and the thermal efficiency of the cycle are to be determined. Also, the effect of changing reheat pressure is to be investigated.

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

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–13. The power plant operates on the ideal reheat Rankine cycle. Therefore, the pump and the turbines are isentropic, there are no pressure drops in the boiler and condenser, and steam leaves the condenser and enters the pump as saturated liquid at the condenser pressure. From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f\,@\,10\text{ psia}} = 161.25 \text{ Btu/lbm}$$

$$v_1 = v_{f\,@\,10\text{ psia}} = 0.01659 \text{ ft}^3\text{/lbm}$$

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

$$= (0.01659 \text{ ft}^3\text{/lbm})[(600 - 100) \text{ psia}]\left(\frac{1 \text{ Btu}}{5.404 \text{ psia-ft}^3}\right)$$

$$= 1.81 \text{ Btu/lbm}$$

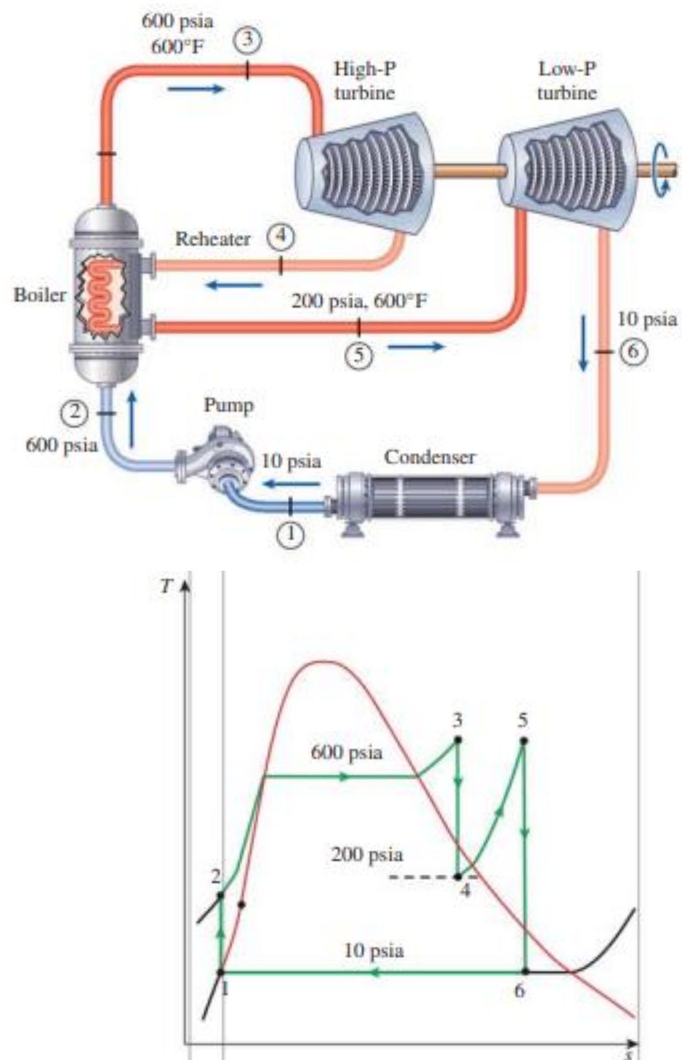


FIGURE 10–13
Schematic and *T-s* diagram for Example 10-4.

$$h_2 = h_1 + w_{pump,in} = 161.25 + 1.81 = 163.06 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 600 \text{ psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1289.9 \text{ Btu/lbm} \\ s_3 = 1.5325 \text{ Btu/lbm·R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 200 \text{ psia} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.5325 - 0.54379}{1.00219} = 0.9865 \\ h_4 = h_f + x_4 h_{fg} = 355.46 + (0.9865)(843.33) = 1187.5 \text{ Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_5 = 200 \text{ psia} \\ T_5 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_5 = 1322.3 \text{ Btu/lbm} \\ s_5 = 1.6771 \text{ Btu/lbm·R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{1.6771 - 0.28362}{1.50391} = 0.9266 \\ h_6 = h_f + x_6 h_{fg} = 161.25 + (0.9266)(981.82) = 1071.0 \text{ Btu/lbm} \end{array}$$

Thus,

$$q_{in} = (h_3 - h_2) + (h_5 - h_4) = 1289.9 - 163.06 + 1322.3 - 1187.5 = 1261.7 \text{ Btu/lbm}$$

$$q_{out} = h_6 - h_1 = 1071.0 - 161.25 = 909.7 \text{ Btu/lbm}$$

$$w_{net} = q_{in} - q_{out} = 1261.7 - 909.8 = 352.0 \text{ Btu/lbm}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{net} = \dot{m} w_{net} \rightarrow \dot{m} = \frac{\dot{W}_{net}}{w_{net}} = \frac{5000 \text{ kJ/s}}{352.0 \text{ Btu/lbm}} \left(\frac{0.94782 \text{ Btu}}{1 \text{ kJ}}\right) = 13.47 \text{ lbm/s}$$

The rates of heat addition and rejection are

$$\dot{Q}_{in} = \dot{m} q_{in} = (13.47 \text{ lbm/s})(1261.7 \text{ Btu/lbm}) = 16{,}995 \text{ Btu/s}$$

$$\dot{Q}_{out} = \dot{m} q_{out} = (13.47 \text{ lbm/s})(909.7 \text{ Btu/lbm}) = 12{,}250 \text{ Btu/s}$$

and the thermal efficiency of the cycle is

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{5000 \text{ kJ/s}}{16{,}995 \text{ Btu/s}} \left(\frac{0.94782 \text{ Btu}}{1 \text{ kJ}}\right) = 0.279 \text{ or } 27.9\%$$

If we repeat the analysis for a reheat pressure of 100 psia at the same reheat temperature, we obtain a thermal efficiency of 27.3 percent. Thus, operating the reheater at 100 psia causes a slight decrease in the thermal efficiency.

Discussion Now we try to address this question: At what reheat pressure will the thermal efficiency be maximum? We repeat the analysis at various reheat pressures using appropriate software. The results are plotted in Fig. 10–14. The thermal efficiency reaches a maximum value of 28.1 percent at an optimum reheat pressure of about 325 psia.

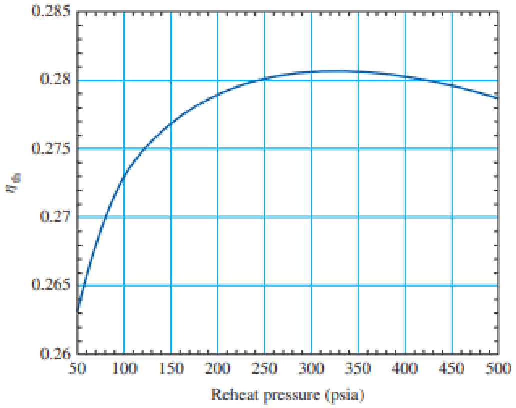


FIGURE 10–14
There is an optimum reheat pressure in the reheat Rankine cycle for which the thermal efficiency is maximum. The values refer to Example 10-4.

10-6. THE IDEAL REGENERATIVE RANKINE CYCLE

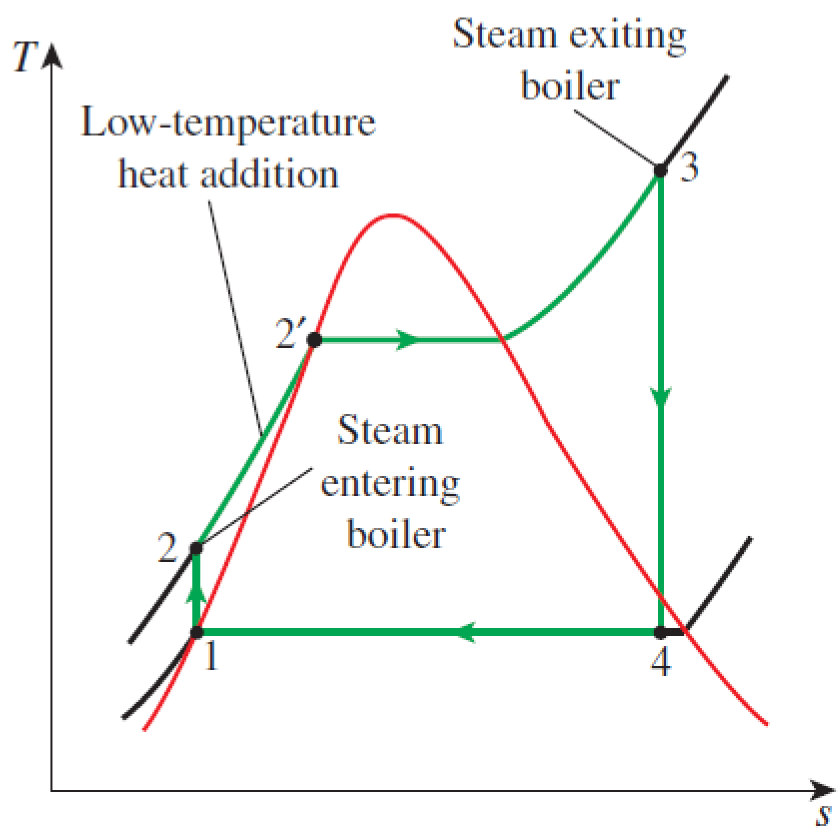


FIGURE 10–14

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).

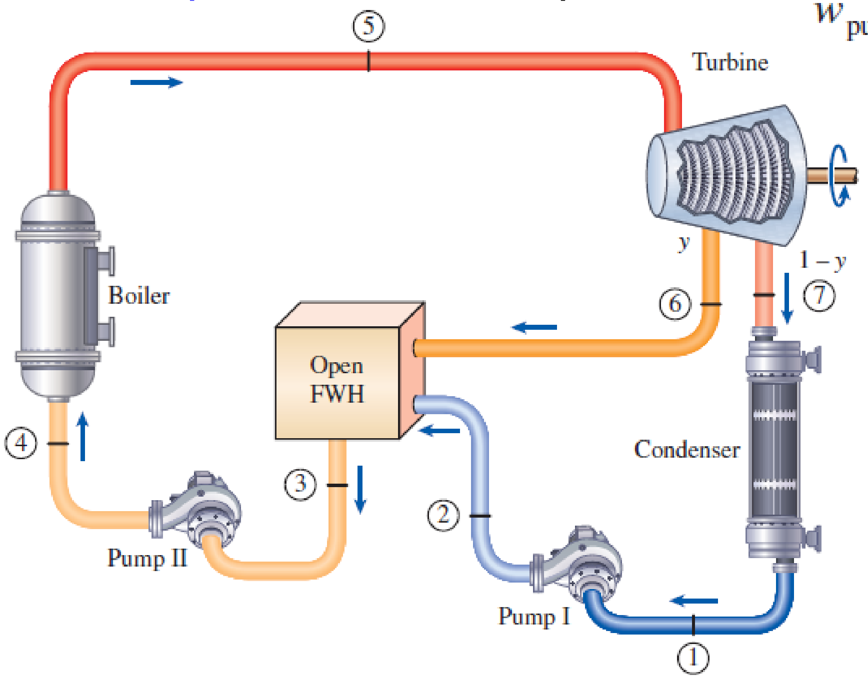
A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (**open feedwater heaters**) or without mixing them (**closed feedwater heaters**).

Open Feedwater Heaters

An **open** (or **direct-contact**) **feedwater heater** is basically a *mixing chamber,* where the steam extracted from the turbine mixes with the feedwater exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure.

$$q_{in} = h_5 - h_4$$
$$q_{out} = (1 - y)(h_7 - h_1)$$
$$w_{turb,out} = (h_5 - h_6) + (1 - y)(h_6 - h_7)$$
$$w_{pump,in} = (1 - y)w_{pump\ I,in} + w_{pump\ II,in}$$

$$y = \dot{m}_6/\dot{m}_5 \qquad \text{(fraction of steam extracted)}$$
$$w_{pump\ I,in} = v_1(P_2 - P_1)$$
$$w_{pump\ II,in} = v_3(P_4 - P_3)$$



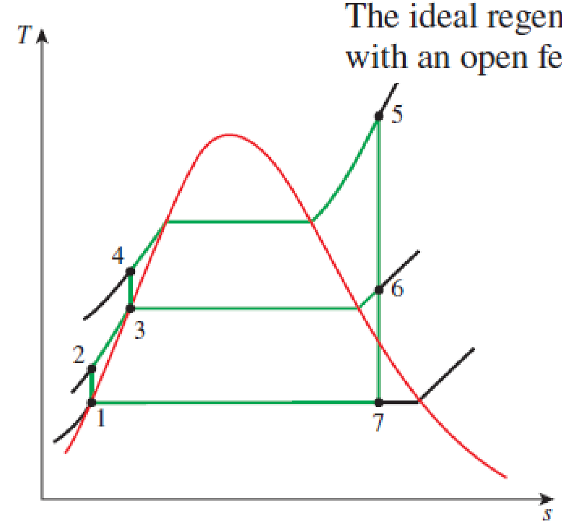


FIGURE 10–16
The ideal regenerative Rankine cycle with an open feedwater heater.

Example 10-5. 이상적 재생 랭킨사이클

EXAMPLE 10–5 The Ideal Regenerative Rankine Cycle

Consider a steam power plant operating on the ideal regenerative Rankine cycle with one open feedwater heater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam leaves the turbine at a pressure of 1.2 MPa and enters the open feedwater heater. Determine the fraction of steam extracted from the turbine and the thermal efficiency of the cycle.

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

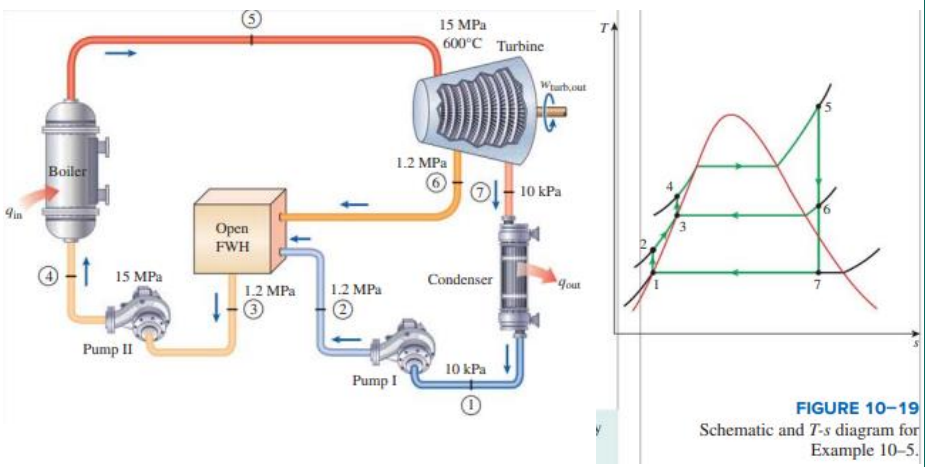


FIGURE 10–19
Schematic and *T-s* diagram for Example 10–5.

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

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–19. We note that the power plant operates on the ideal regenerative Rankine cycle. Therefore, the pumps and the turbines are isentropic; there are no pressure drops in the boiler, condenser, and feedwater heater; and steam leaves the condenser and the feedwater heater as saturated liquid. First, we determine the enthalpies at various states:

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

State 2: $P_2 = 1.2$ MPa, $s_2 = s_1$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101\ \text{m}^3/\text{kg})[(1200 - 10)\text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa·m}^3}\right) = 1.20\ \text{kJ/kg}$$

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

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

State 4: $P_4 = 15$ MPa, $s_4 = s_3$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3) = (0.001138\ \text{m}^3/\text{kg})[(15{,}000 - 1200)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa·m}^3}\right) = 15.70\ \text{kJ/kg}$$

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

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

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: $P_7 = 10$ kPa

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

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

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

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

or

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

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

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{798.33 - 193.01}{2860.2 - 193.01} = \mathbf{0.2270}$$

Thus,

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

$$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.2270)(2115.3 - 191.81) \text{ kJ/kg}$$
$$= 1486.9 \text{ kJ/kg}$$

and

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

Discussion This problem was worked out in Example 10–3*c* 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.

Closed Feedwater Heaters

Another type of feedwater heater frequently used in steam power plants is the **closed feedwater heater**, in which heat is transferred from the extracted steam to the feedwater without any mixing taking place. The two streams now can be at different pressures, since they do not mix.

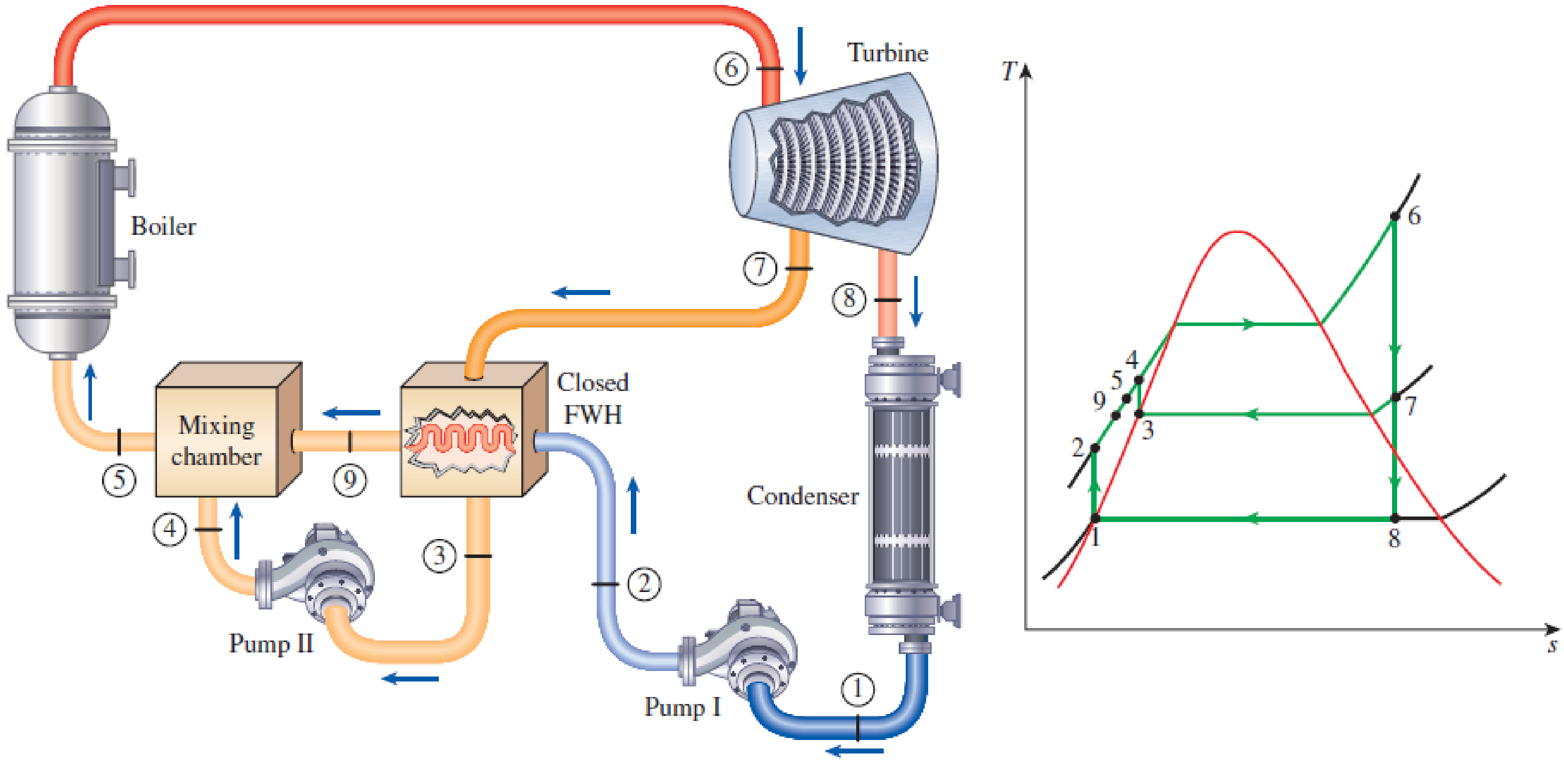


FIGURE 10–16
The ideal regenerative Rankine cycle with a closed feedwater heater.

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.

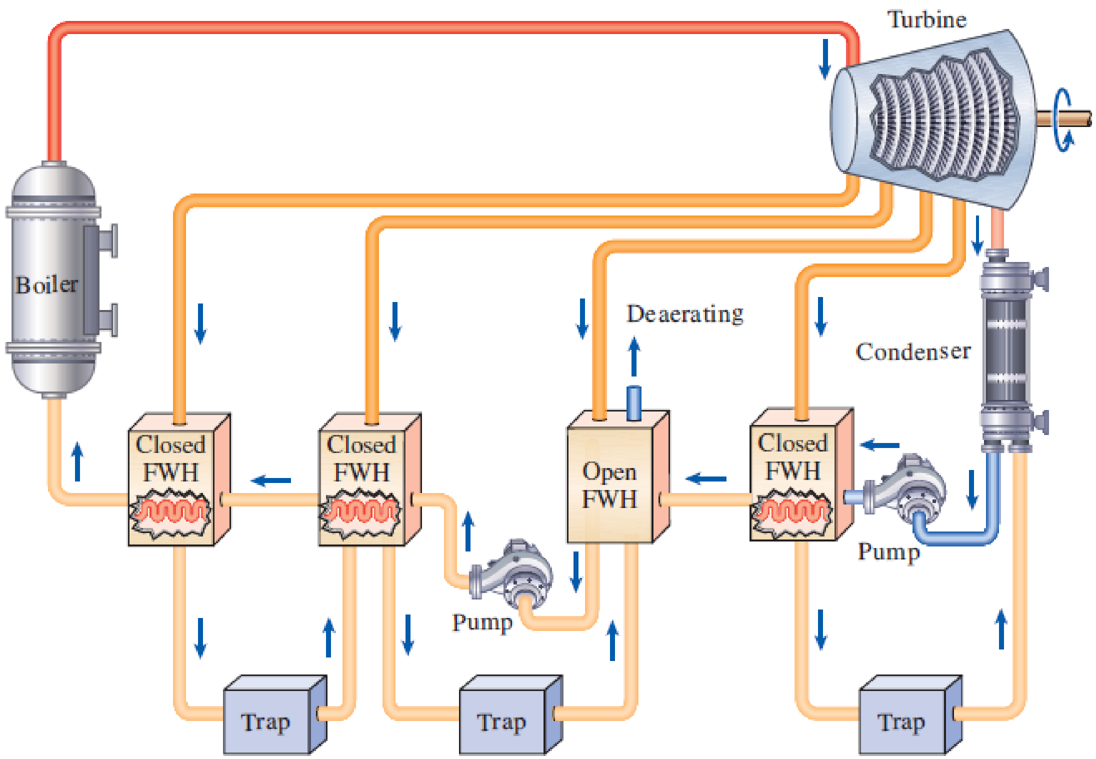


FIGURE 10–17
A steam power plant with one open and three closed feedwater heaters.

10-7. SECOND-LAW ANALYSIS OF VAPOR POWER CYCLES

Exergy destruction for a steady-flow system

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

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

Steady-flow, one-inlet, one-exit

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

Exergy destruction of a cycle

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

For a cycle with heat transfer only with a source and a sink

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

Stream exergy

A second-law analysis of vaporpower cycles reveals where the largest irreversibilities occur and where to start improvements.

10-8. 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.

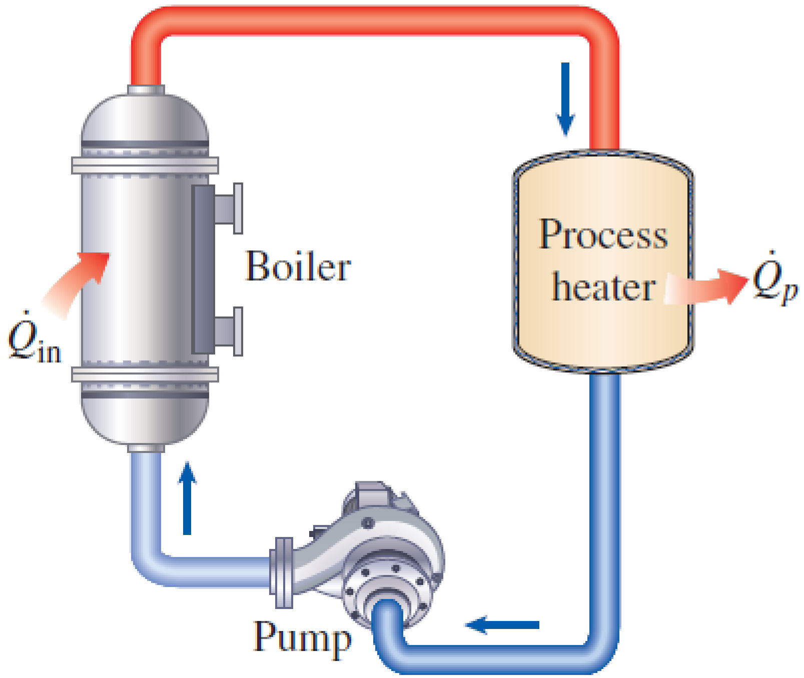


FIGURE 10–21
A simple process-heating plant.

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

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

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

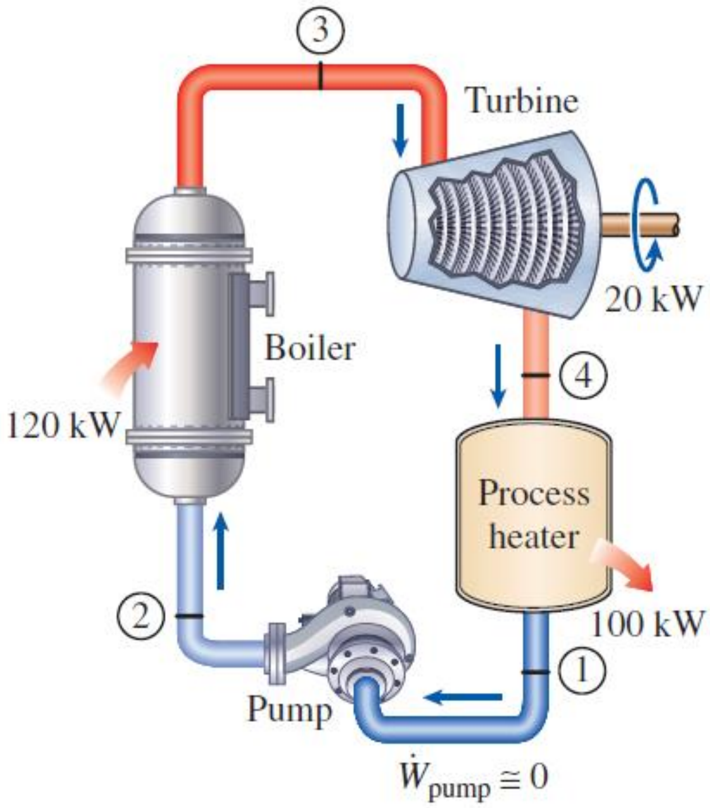


FIGURE 10–22
An ideal cogeneration plant.

The utilization factor of the ideal steam-turbine cogeneration plant is 100%.

Actual cogeneration plants have utilization factors as high as 80%.

Some recent cogeneration plants have even higher utilization factors.

10-8. COGENERATION

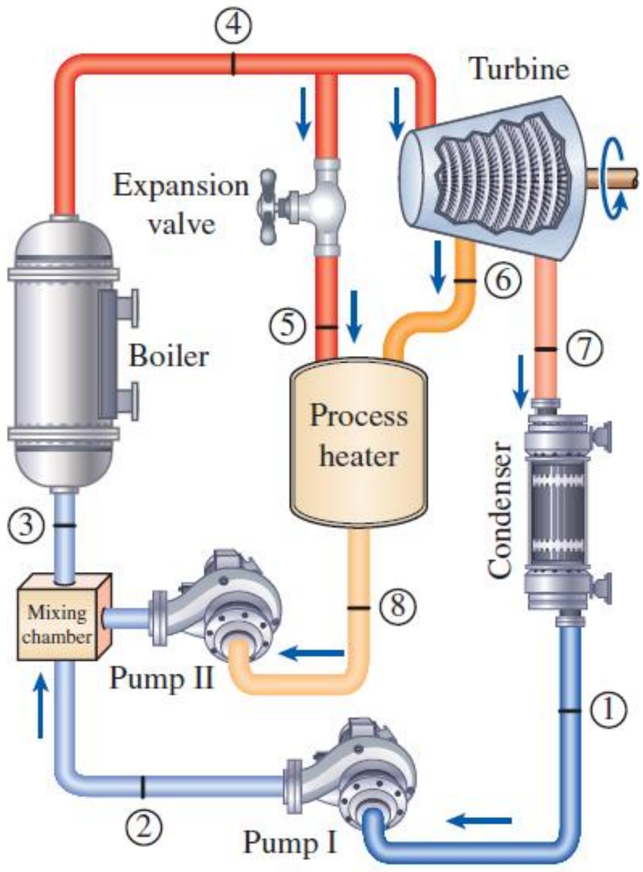


FIGURE 10–23
A cogeneration plant with adjustable loads.

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 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 valve ($m_5 = m_4$). No power is produced in this mode.

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

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

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

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

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

10-9. COMBINED GAS–VAPOR POWER CYCLES

The continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants.

A popular modification involves a gas power cycle topping a vapor power cycle, which is called the **combined gas–vapor cycle**, or just the **combined cycle**.

The combined cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam-turbine (Rankine) cycle, which has a higher thermal efficiency than either of the cycles executed individually.

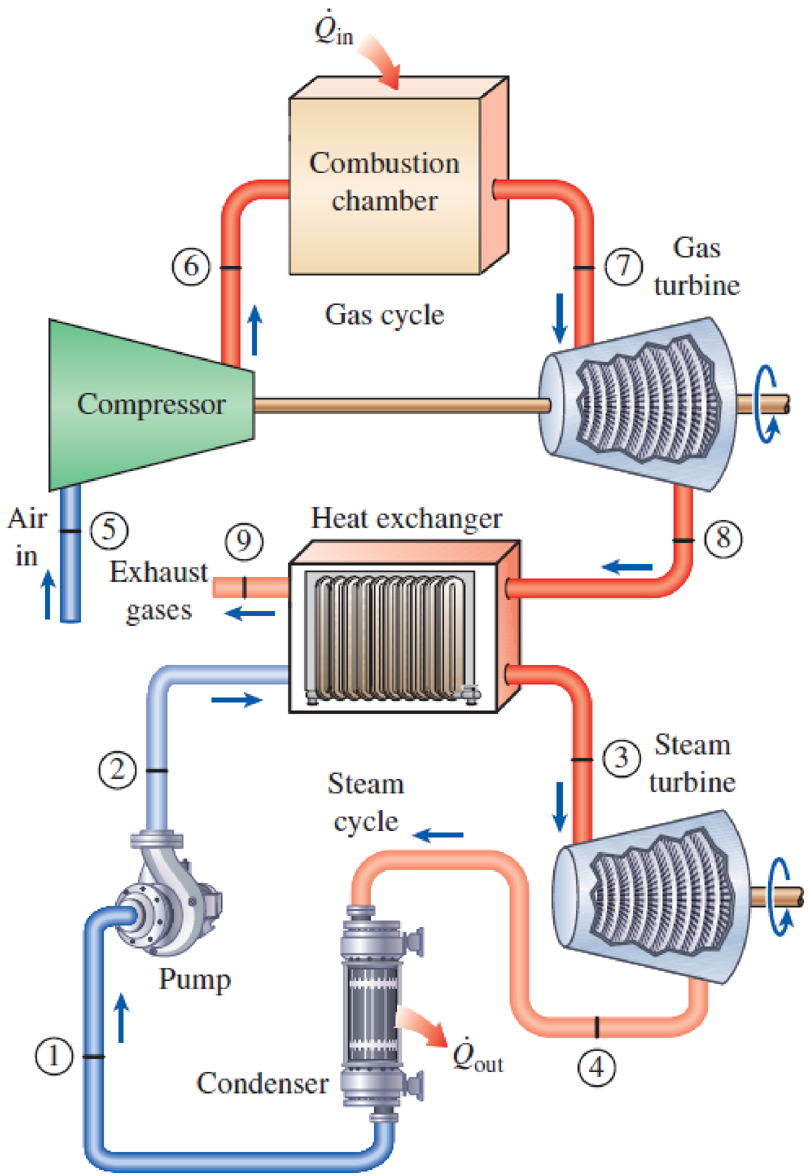
It makes engineering sense to take advantage of the very desirable characteristics of the gas-turbine cycle at high temperatures *and* to use the high-temperature exhaust gases as the energy source for the bottoming cycle such as a steam power cycle. The result is a combined gas–steam cycle.

Recent developments in gas-turbine technology have made the combined gas–steam cycle economically very attractive.

The combined cycle increases the efficiency without increasing the initial cost greatly. Consequently, many new power plants operate on combined cycles, and many more existing steam- or gas-turbine plants are being converted to combined-cycle power plants.

Thermal efficiencies over 50% are reported.

10-9. COMBINED GAS–VAPOR POWER CYCLES



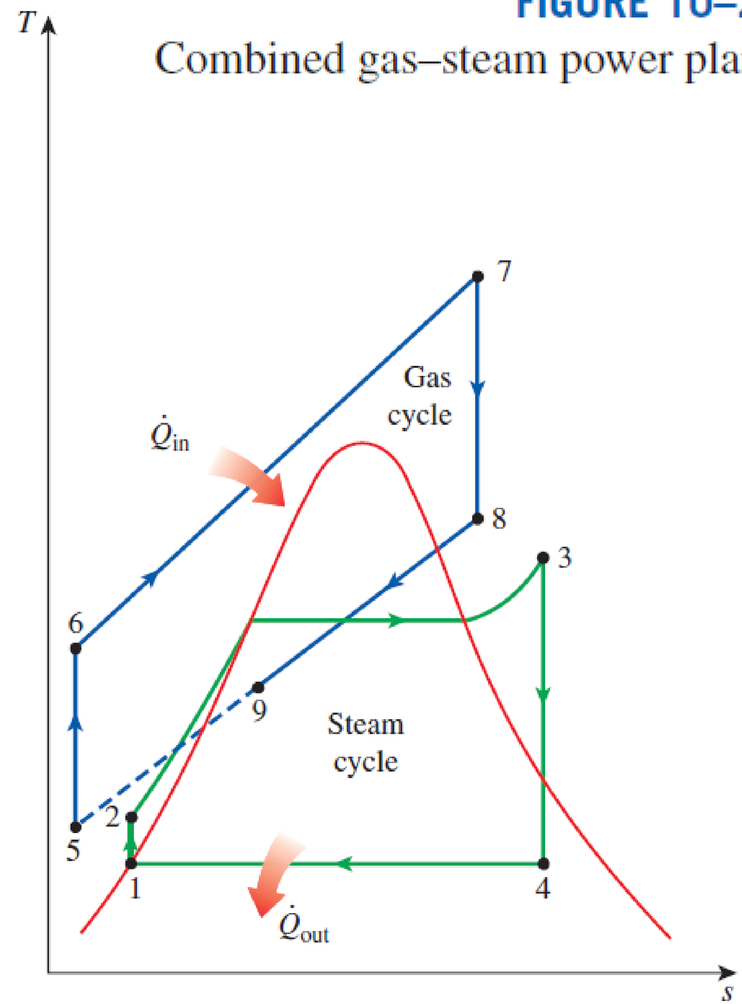


FIGURE 10–25
Combined gas–steam power plant.

10-9. COMBINED GAS–VAPOR POWER CYCLES

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

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

Another plant built by Siemens in Irsching, Germany, in 2011 reached a thermal efficiency of 60.8 percent with an electrical output of 578 MW.

In 2016, General Electric reported 62.2 percent efficiency for its combined cycle plant in Bouchain, France with an output of 594 MW.

The new target for thermal efficiency is 65 percent, approaching the Carnot limit.

Example 10-9. 기체-증기 복합동력사이클

EXAMPLE 10–9 A Combined Gas–Steam Power Cycle

Consider the combined gas–steam power cycle shown in Fig. 10–27. The topping cycle is a gas-turbine cycle that has a pressure ratio of 8. Air enters the compressor at 300 K and the turbine at 1300 K. The isentropic efficiency of the compressor is 80 percent, and that of the gas turbine is 85 percent. The bottoming cycle is a simple ideal Rankine cycle operating between the pressure limits of 7 MPa and 5 kPa. Steam is heated in a heat exchanger by the exhaust gases to a temperature of 500°C. The exhaust gases leave the heat exchanger at 450 K. Determine (*a*) the ratio of the mass flow rates of the steam and the combustion gases and (*b*) the thermal efficiency of the combined cycle.

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

Analysis The *T-s* diagrams of both cycles are given in Fig. 10–27. The gas-turbine cycle alone was analyzed in Example 9–6, and the steam cycle in Example 10–8*b*, with the following results:

Gas cycle: $h_{4'} = 880.36$ kJ/kg $(T_{4'} = 853\text{ K})$
$q_{in} = 790.58$ kJ/kg $w_{net} = 210.41$ kJ/kg $\eta_{th} = 26.6\%$
$h_{5'} = h_{@\,450\text{ K}} = 451.80$ kJ/kg

Steam cycle: $h_2 = 144.78$ kJ/kg $(T_2 = 33°\text{C})$
$h_3 = 3411.4$ kJ/kg $(T_3 = 500°\text{C})$
$w_{net} = 1331.4$ kJ/kg $\eta_{th} = 40.8\%$

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

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}_g h_{5'} + \dot{m}_s h_3 = \dot{m}_g h_{4'} + \dot{m}_s h_2$$
$$\dot{m}_s (h_3 - h_2) = \dot{m}_g (h_{4'} - h_{5'})$$
$$\dot{m}_s (3411.4 - 144.78) = \dot{m}_g (880.36 - 451.80)$$

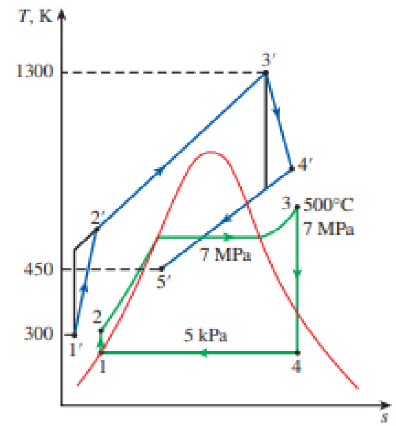


FIGURE 10–27
T-s diagram of the gas–steam combined cycle described in Example 10–9.

Thus,

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

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

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

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

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

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
- Deviation of actual vapor power cycles from idealized ones
- How can we increase the efficiency of the Rankine cycle?
- The ideal reheat Rankine cycle
- The ideal regenerative Rankine cycle
- Second-law analysis of vapor power cycles
- Cogeneration
- Combined gas–vapor power cycles