Thermodynamics: An Engineering Approach, 5th Edition
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Chapter 10

VAPOR AND COMBINED POWER CYCLES

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Objectives

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

THE CARNOT VAPOR CYCLE

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:

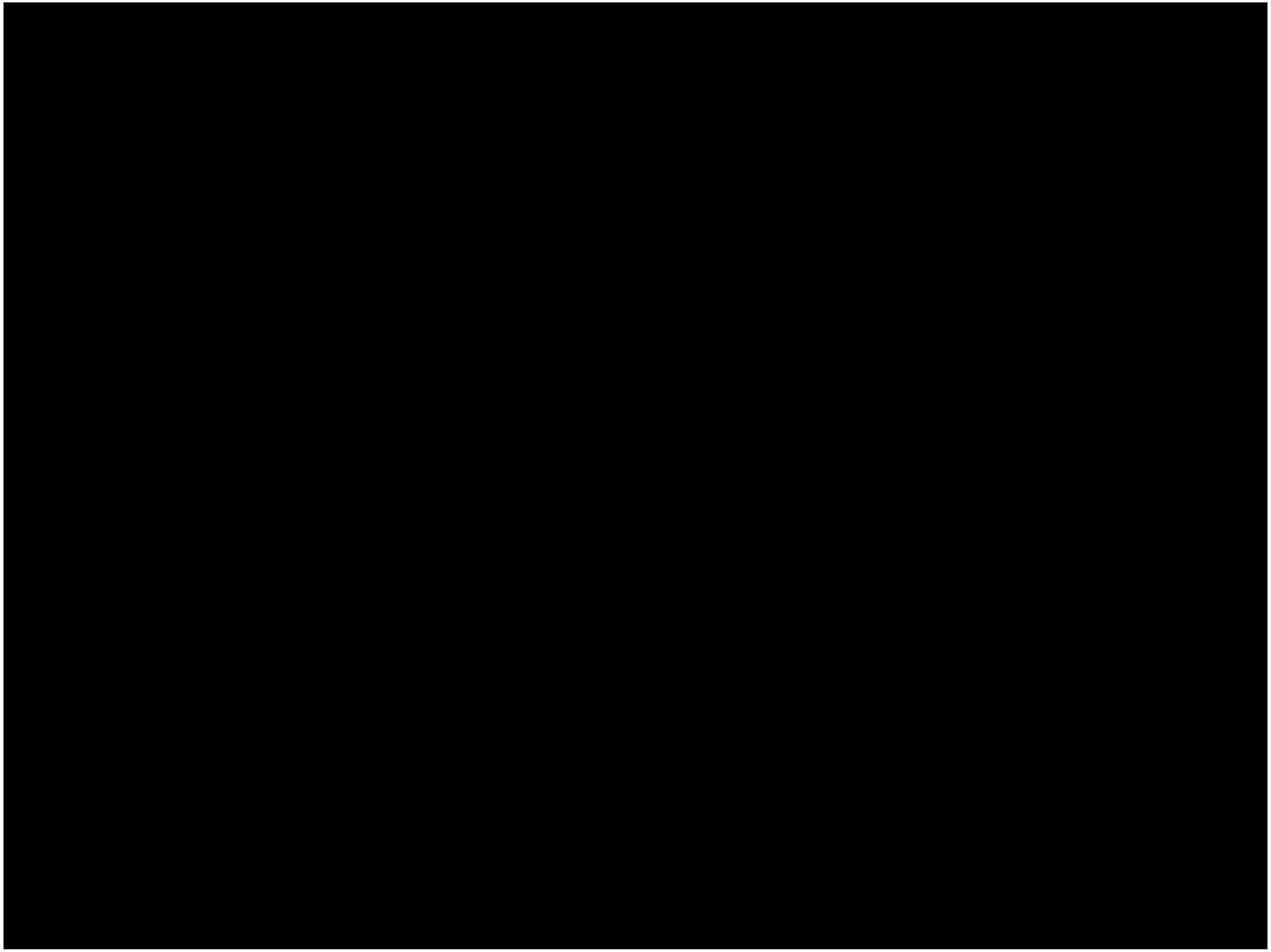
Process 1-2 Limiting the heat transfer processes to two-phase systems severely limits the maximum temperature that can be used in the cycle (374°C for water)

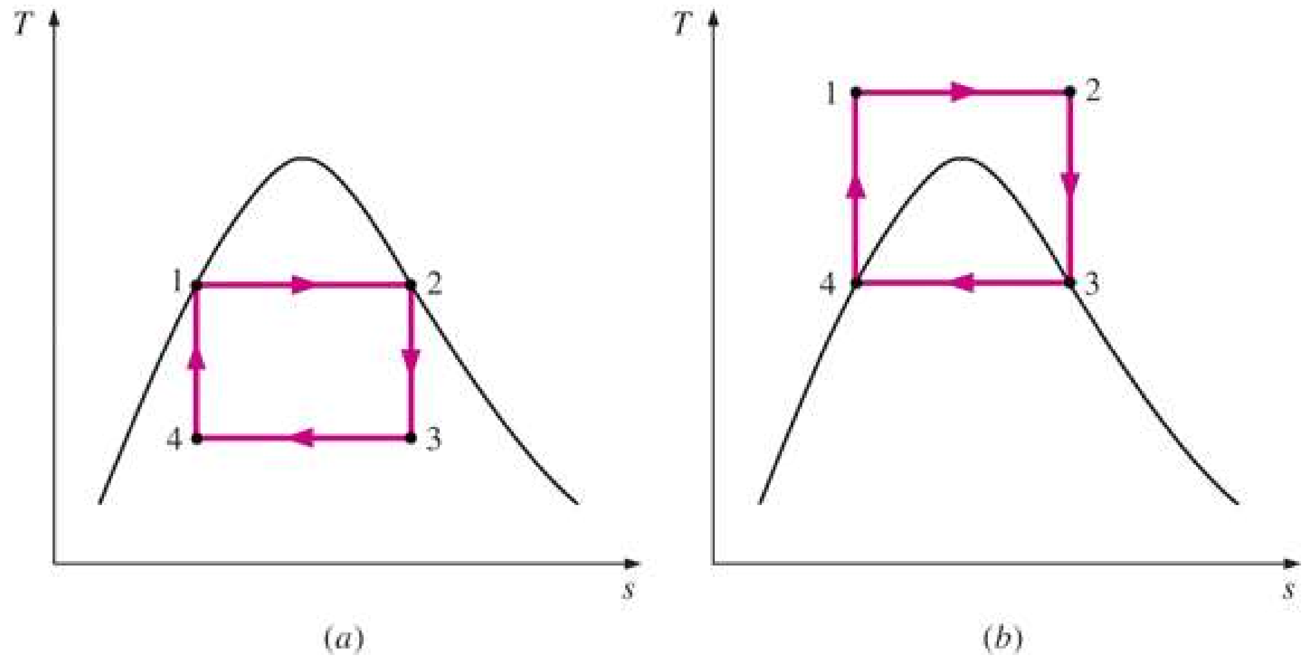
Process 2-3 The turbine cannot handle steam with a high moisture content because of the impingement of liquid droplets on the turbine blades causing erosion and wear.

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.

Carnot Cycles





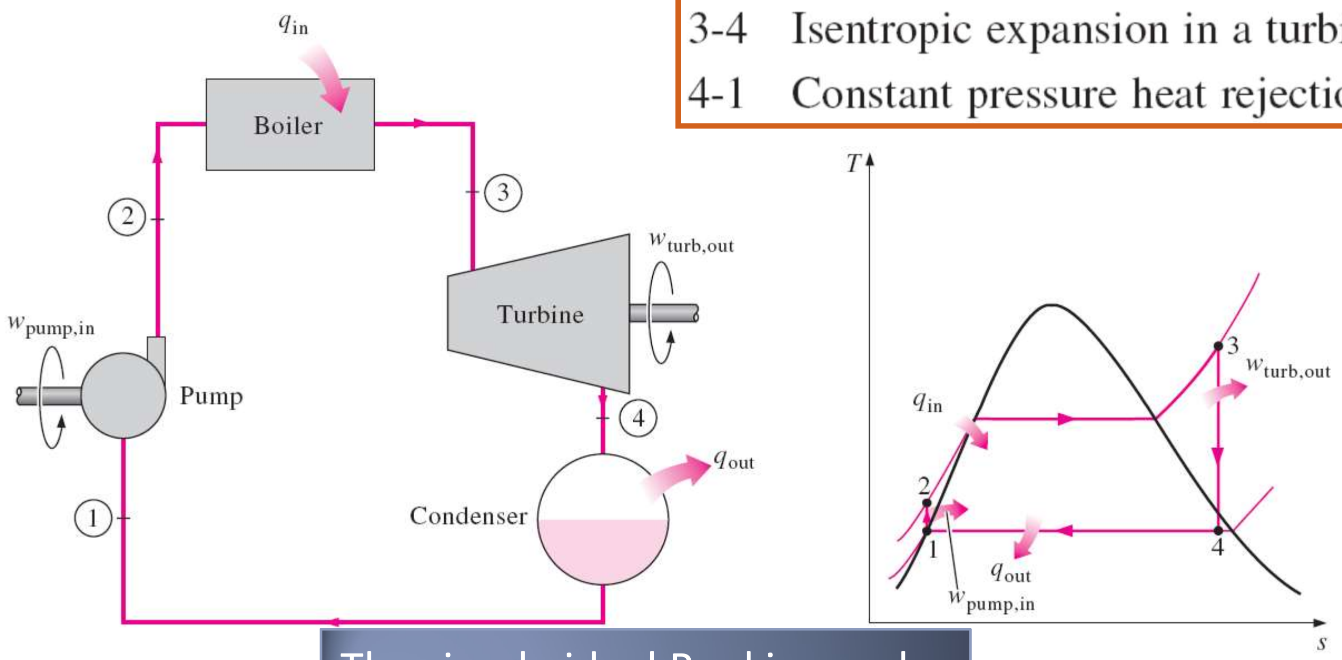
T-s diagram of two Carnot vapor cycles.

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

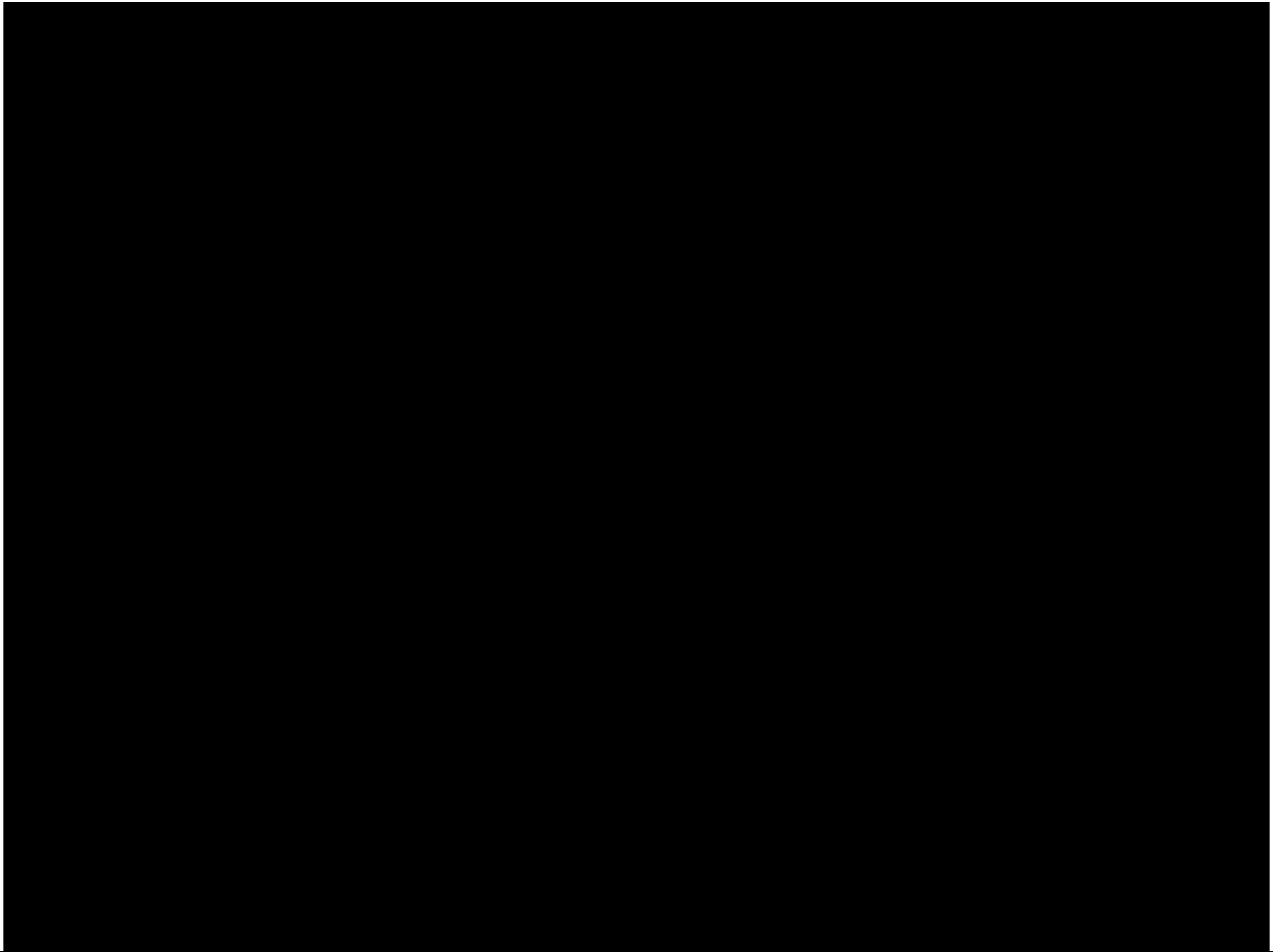
Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser.

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.

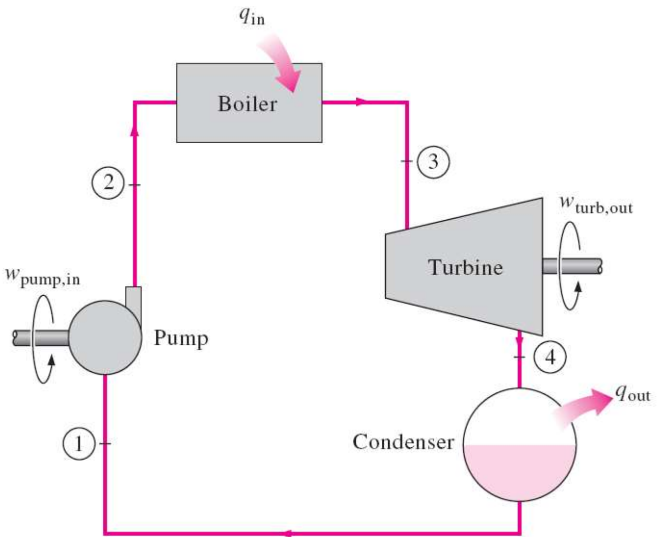
1-2	Isentropic compression in a pump
2-3	Constant pressure heat addition in a boiler
3-4	Isentropic expansion in a turbine
4-1	Constant pressure heat rejection in a condenser

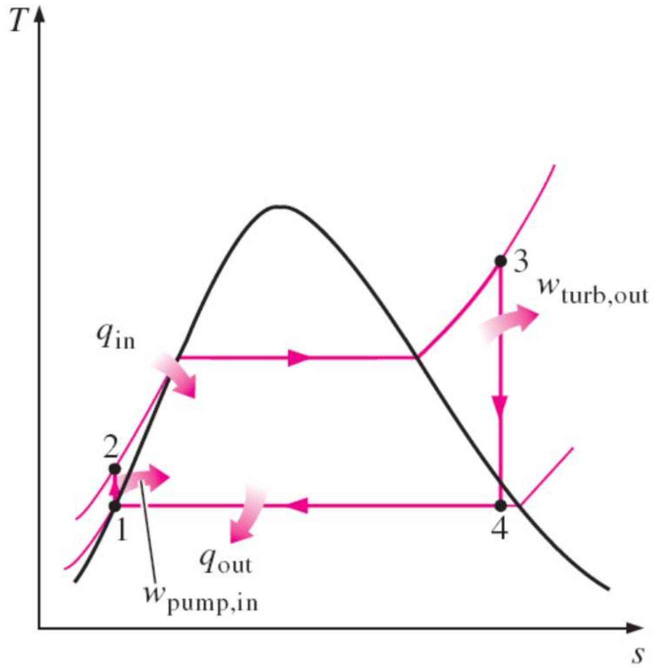


The simple ideal Rankine cycle.



Energy Analysis of the Ideal Rankine Cycle





Steady-flow energy equation

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

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

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

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

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

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

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

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

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.

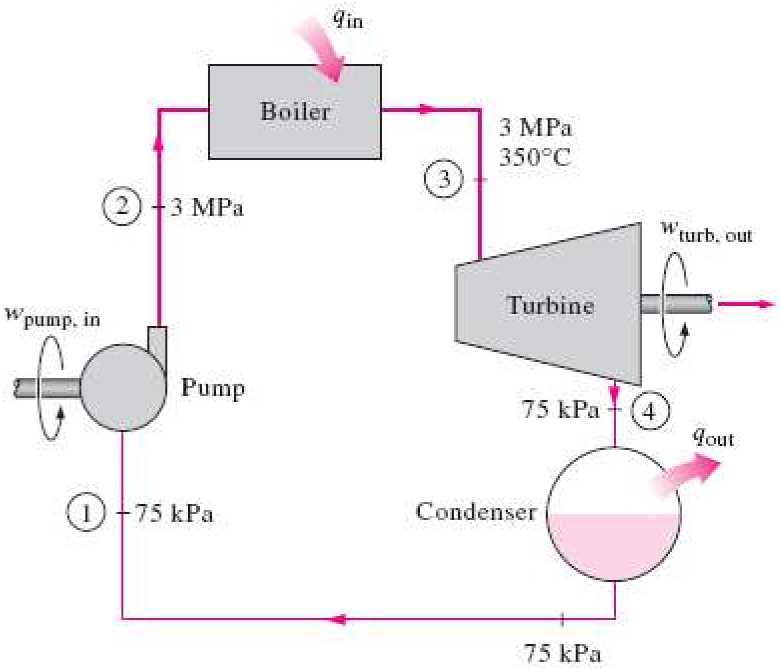
$$\boxed{\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}}}$$

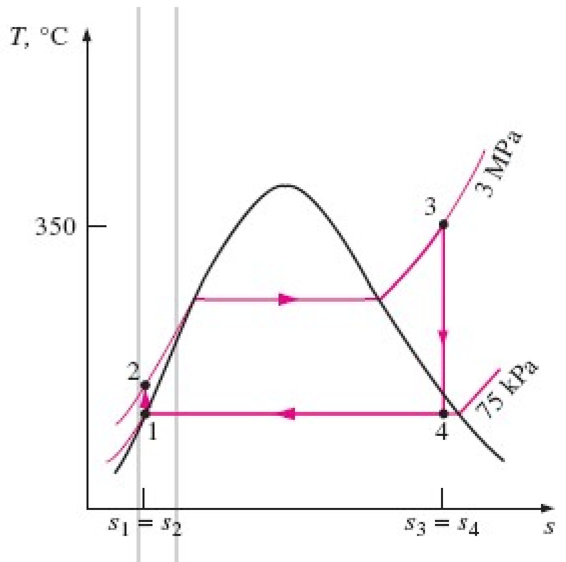
EXAMPLE 10–1

Consider a steam power plant operating on the simple ideal Rankine cycle. The 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.





State 1: $P_1 = 75$ kPa, Sat. liquid $\Big\}$ $h_1 = h_{f\,@\,75\text{ kPa}} = 384.44$ kJ/kg, $v_1 = v_{f\,@\,75\text{ kPa}} = 0.001037$ m^3/kg

State 2: $P_2 = 3$ MPa, $s_2 = s_1$

$$w_{\text{pump, in}} = v_1(P_2 - P_1) = (0.001037\ \text{m}^3/\text{kg})[(3000 - 75)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 3.03\ \text{kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump, in}} = (384.44 + 3.03)\ \text{kJ/kg} = 387.47\ \text{kJ/kg}$$

State 3: $P_3 = 3$ MPa, $T_3 = 350°$C $\Big\}$ $h_3 = 3116.1$ kJ/kg, $s_3 = 6.7450$ kJ/kg·K

State 4: $P_4 = 75$ kPa (sat. mixture) $s_4 = s_3$

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

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

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

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

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

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

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

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

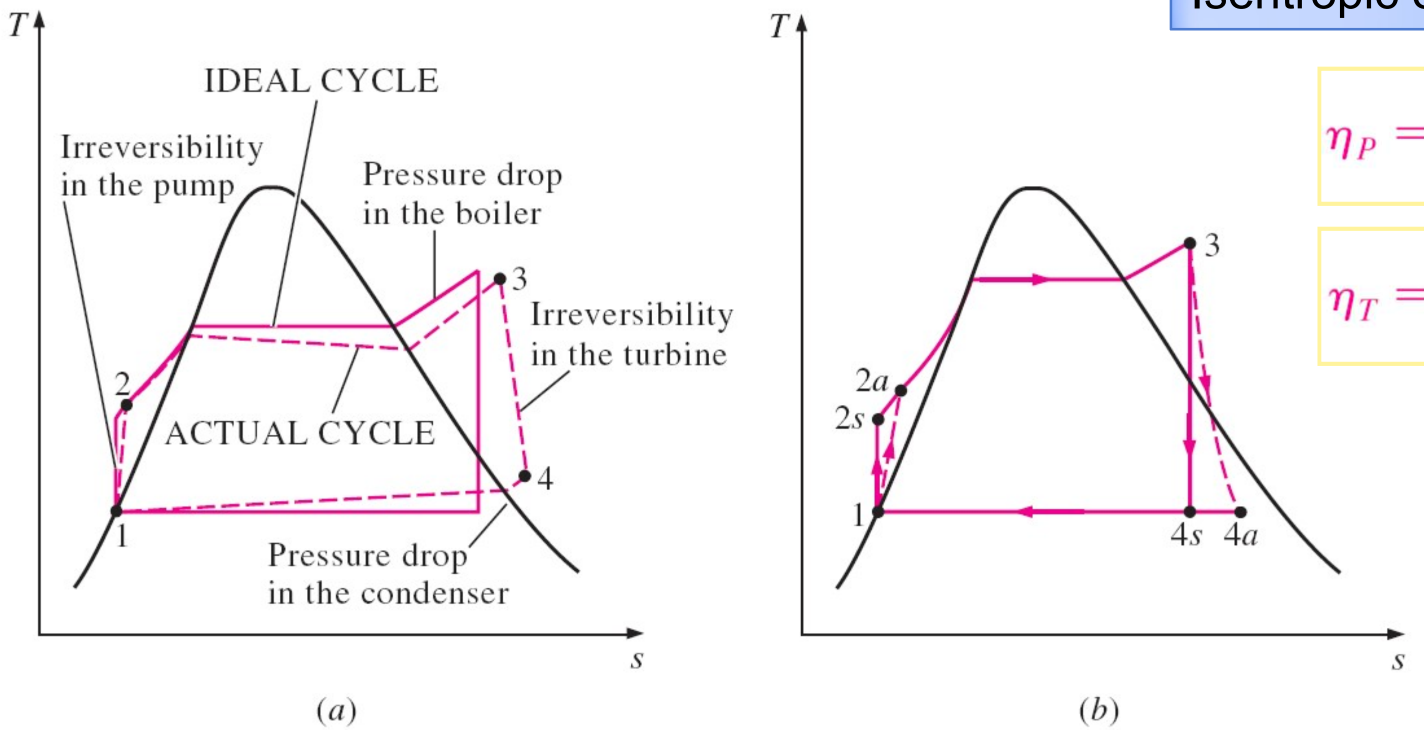
That is, this power plant converts 26 % of the heat it receives in the boiler to net work. An actual power plant operating between the same temperature and pressure limits will have a lower efficiency because of the irreversibilities such as friction.

Discussion Notice that the back work ratio ($r_{pw} = w_{in}/w_{out}$) of this power plant is 0.004, and thus only 0.4 % 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 have very high back work ratios (about 40 to 80 %). It is also interesting to note the thermal efficiency of a Carnot cycle operating between the same temperature limits

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

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

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

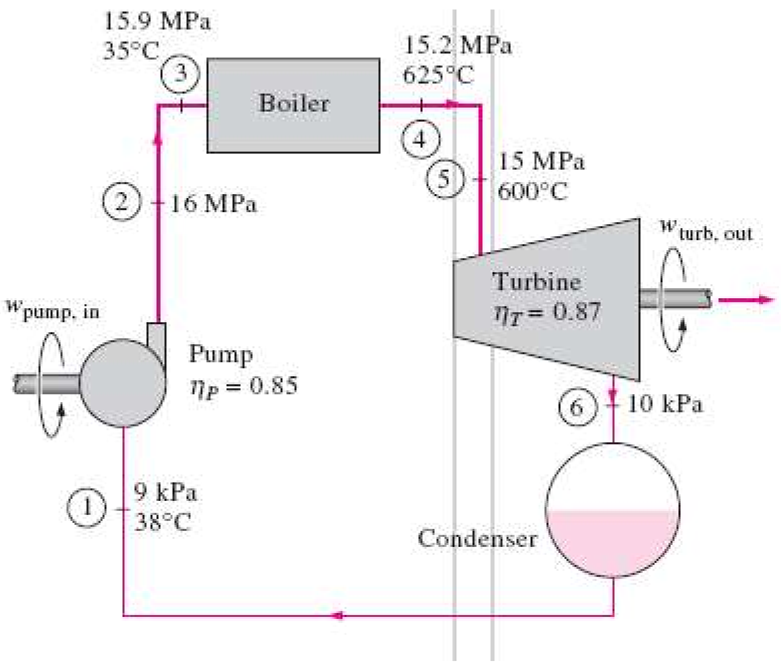
$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

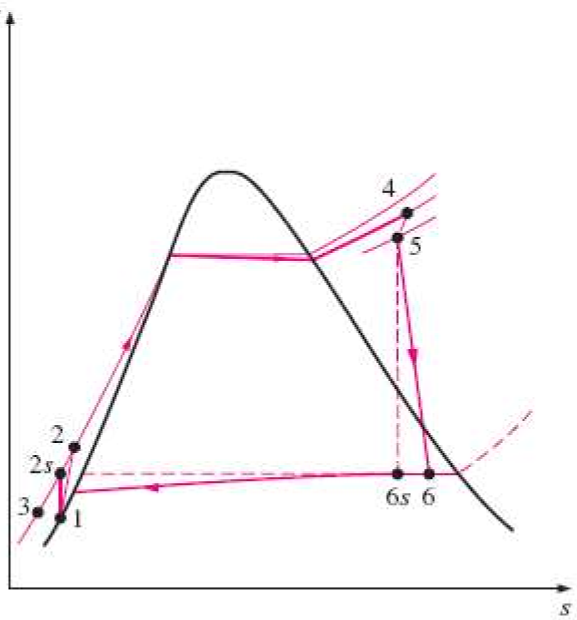
(a) Deviation of actual vapor power cycle from the ideal Rankine cycle.
(b) The effect of pump and turbine irreversibilities on the ideal Rankine cycle.

EXAMPLE 10–2

A steam power plant operates on the cycle shown in Figure. If the isentropic efficiency of the turbine is 87 % and the isentropic efficiency of the pump is 85 %, 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.





(*a*) The thermal efficiency of a cycle;

Pump work input:

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

$$= \frac{(0.001009\ \text{m}^3/\text{kg})[(16{,}000 - 9)\ \text{kPa}]}{0.85}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 19.0\ \text{kJ/kg}$$

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:

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

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

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

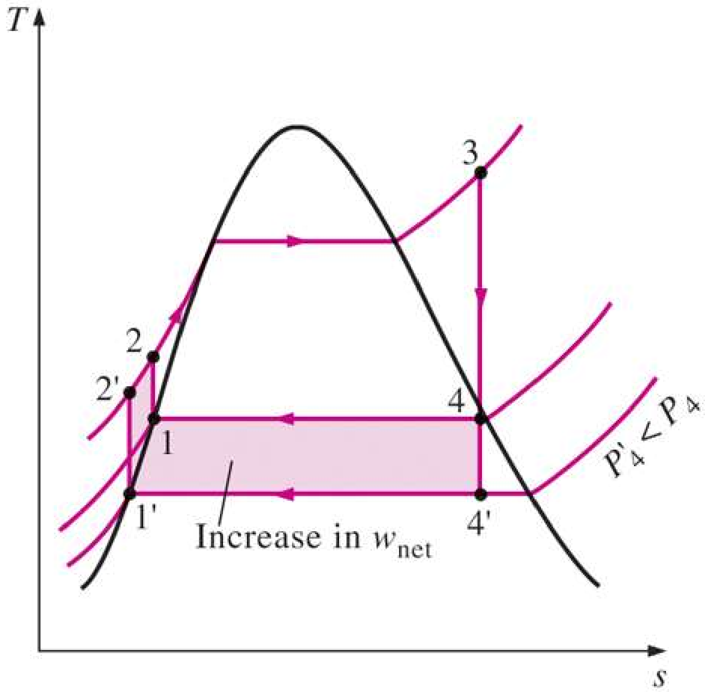
(*b*) The power produced by this power plant is

$$\dot{W}_{\text{net}} = \dot{m}(w_{\text{net}}) = (15\ \text{kg/s})(1258.0\ \text{kJ/kg}) = \mathbf{18{,}870\ kW}$$

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, or decrease the average temperature at which heat is rejected from the working fluid in the condenser.*

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

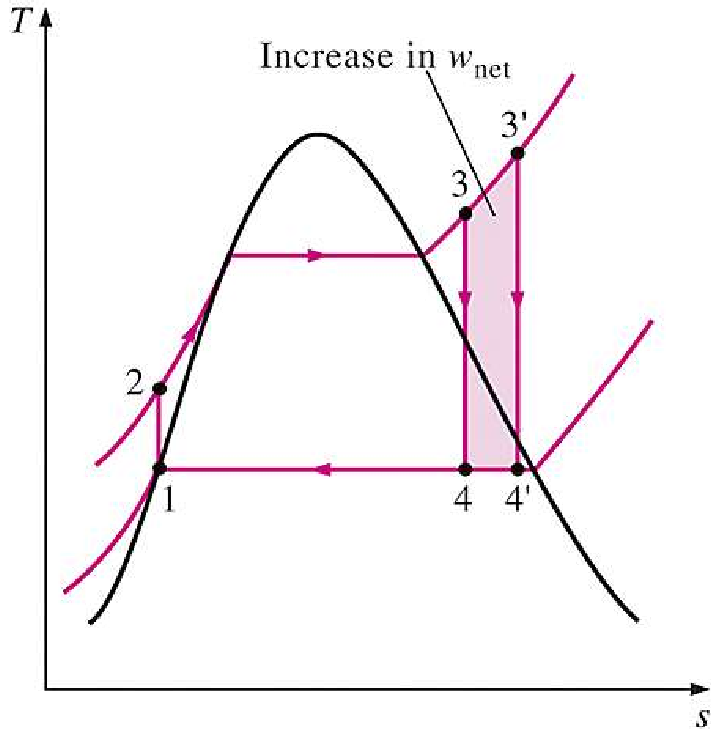


The effect of lowering the condenser pressure on the ideal Rankine cycle.

To take advantage of the increased efficiencies at low pressures, the condensers of steam power plants usually operate well below the atmospheric pressure. 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.

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



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

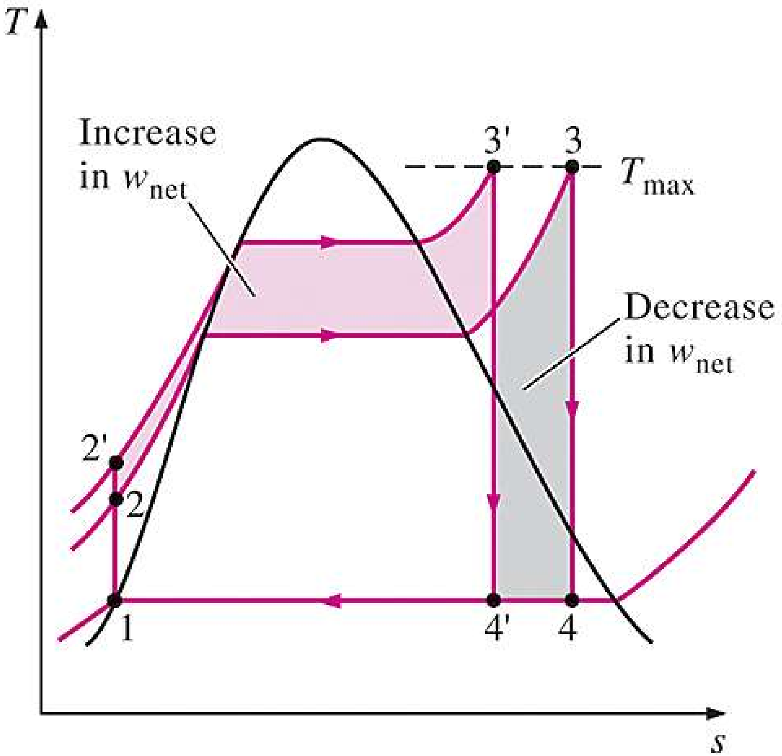
Both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency since the average temperature at which heat is added increases.

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.

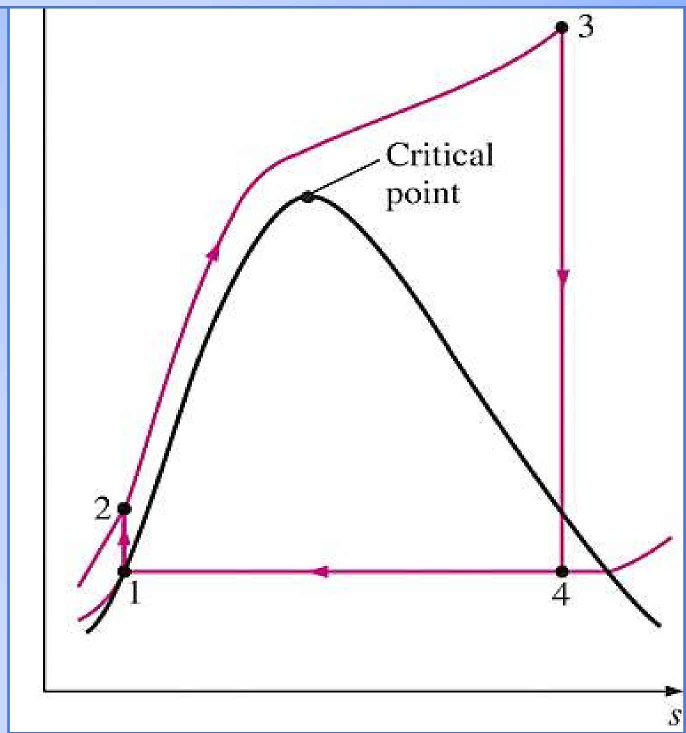
Increasing the Boiler Pressure (Increases $T_{high,avg}$)

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.



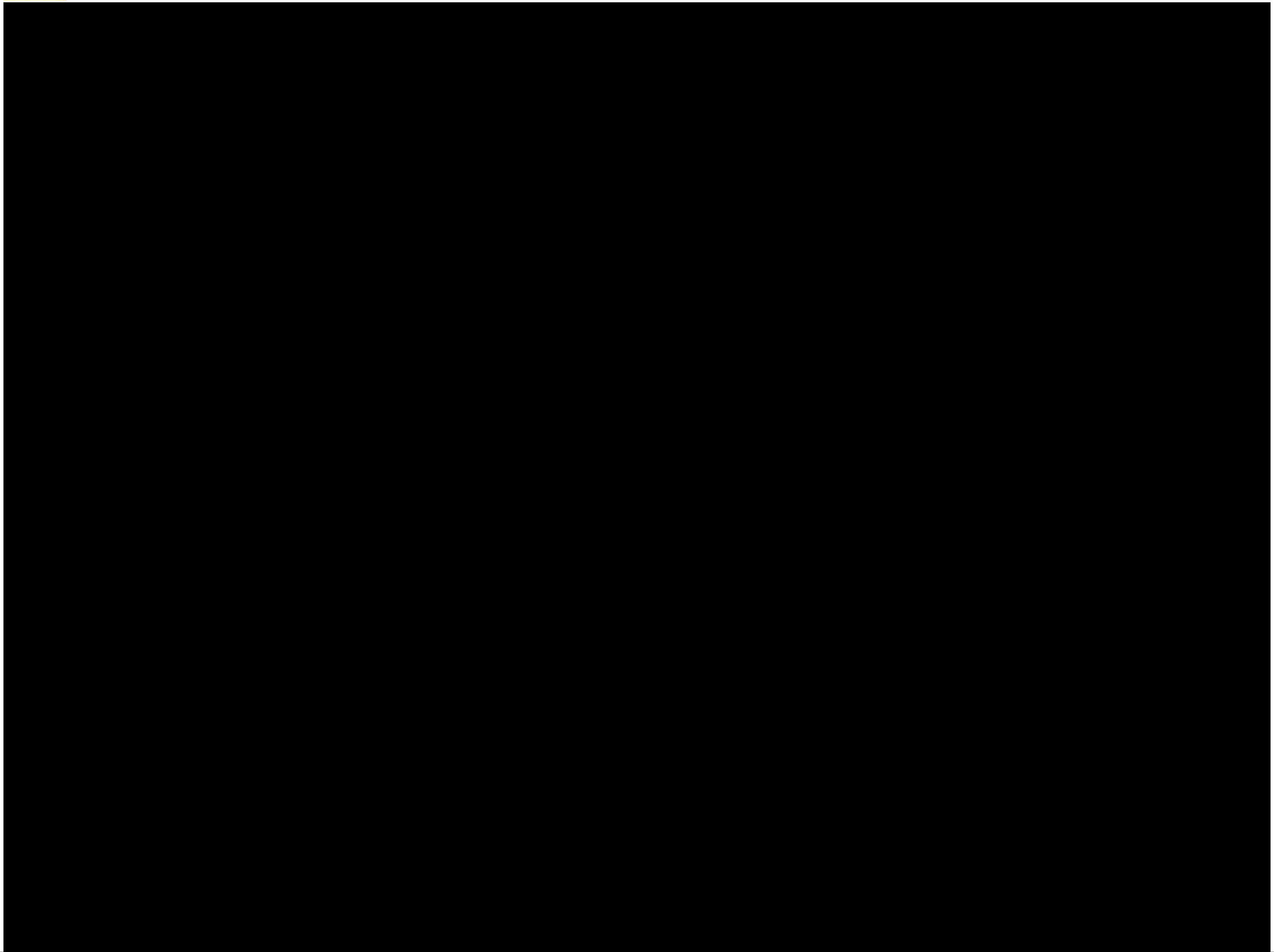
The effect of increasing the boiler pressure on the ideal Rankine cycle.

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.



A supercritical Rankine cycle.

Rankine Cycles (*Increases* $T_{high,avg}$)



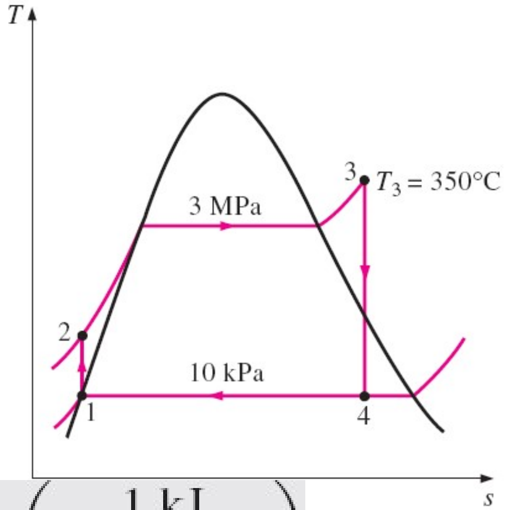
EXAMPLE 10–3

Consider a steam power plant operating on the ideal Rankine cycle. The 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.

(*a*) The thermal efficiency is determined in a similar manner:

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

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



$$w_{\text{pump, in}} = \mathbf{v}_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg})[(3000 - 10) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

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

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

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

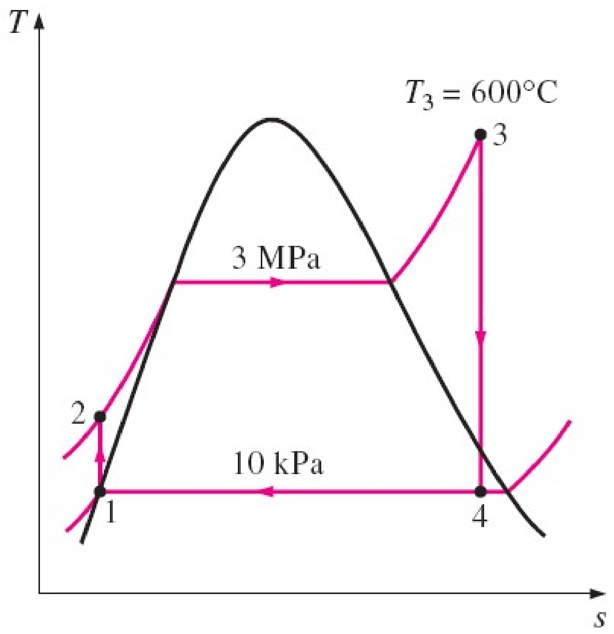
$$\textit{State 4:} \quad P_4 = 10 \text{ kPa} \quad \text{(sat. mixture)}$$
$$s_4 = s_3$$
$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 0.6492}{7.4996} = 0.8128$$

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

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

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



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

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

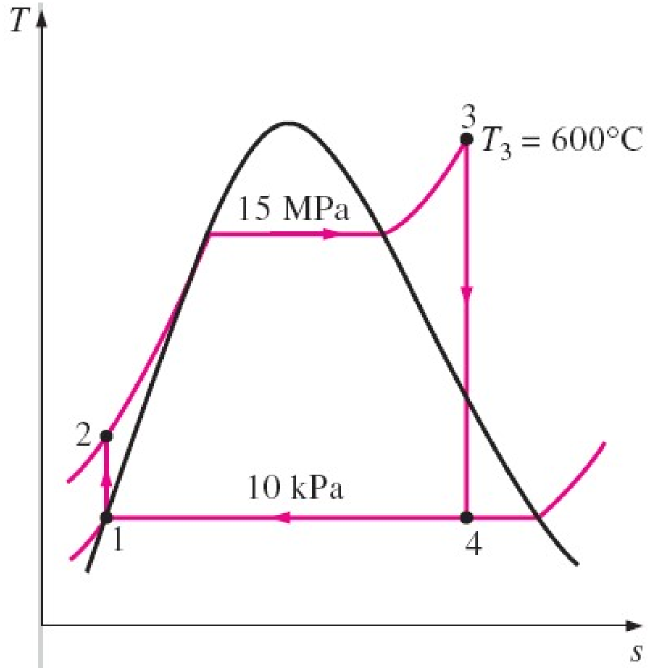
$$q_{in} = h_3 - h_2 = 3682.8 - 194.83 = 3488.0 \text{ kJ/kg}$$

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

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

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



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

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

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

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

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

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

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 % (in other words, the moisture content increases from 8.5 to 19.6 %).

THE IDEAL REHEAT RANKINE CYCLE

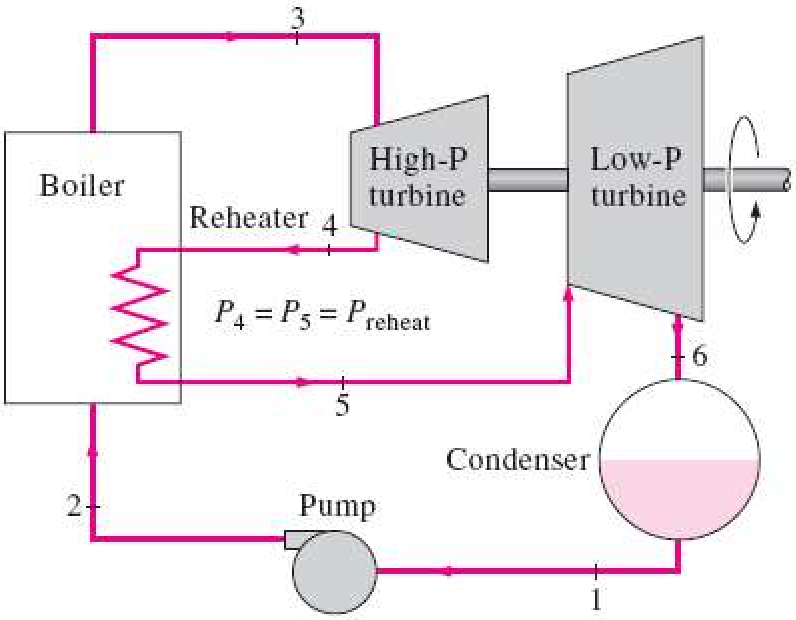
How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?

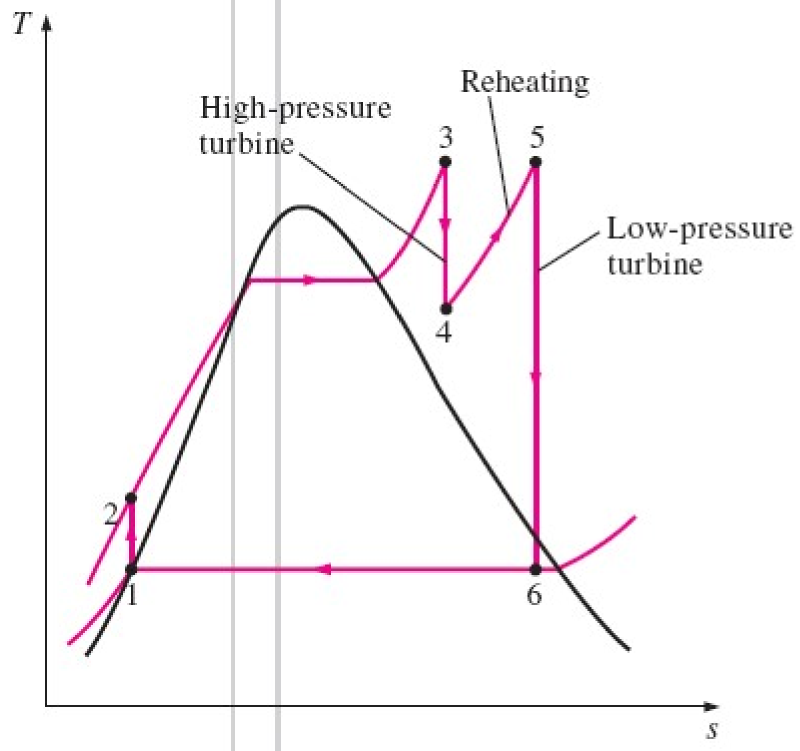
1. Superheat the steam to very high temperatures. It is limited metallurgically.
2. Expand the steam in the turbine in two stages, and reheat it in between (**reheat**)

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

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

The ideal reheat Rankine cycle.



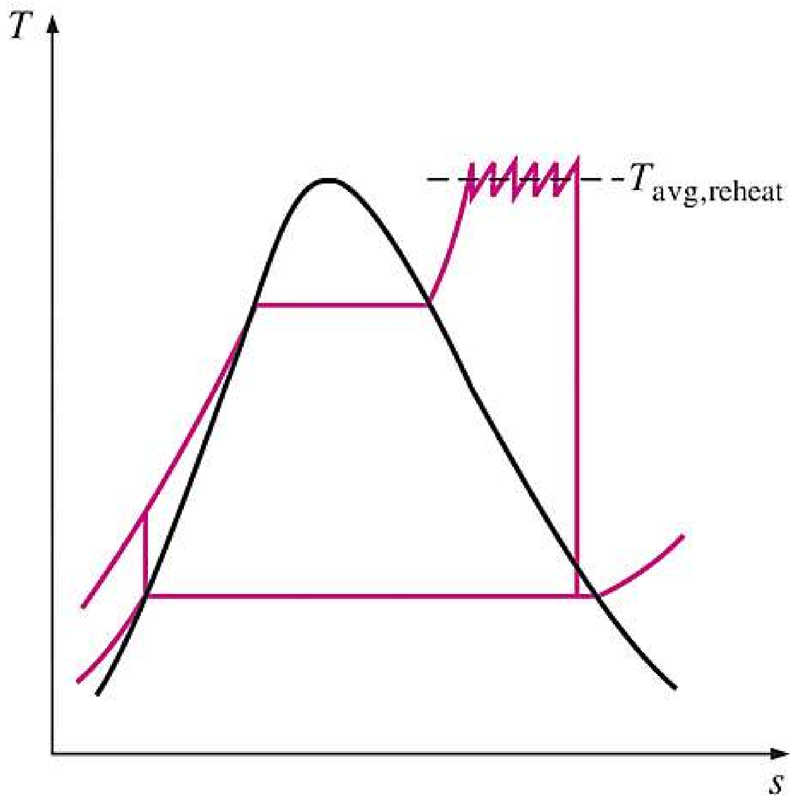


The single reheat in a modern power plant improves the cycle efficiency by 4 to 5% by increasing the average temperature at which heat is transferred to the steam.

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.

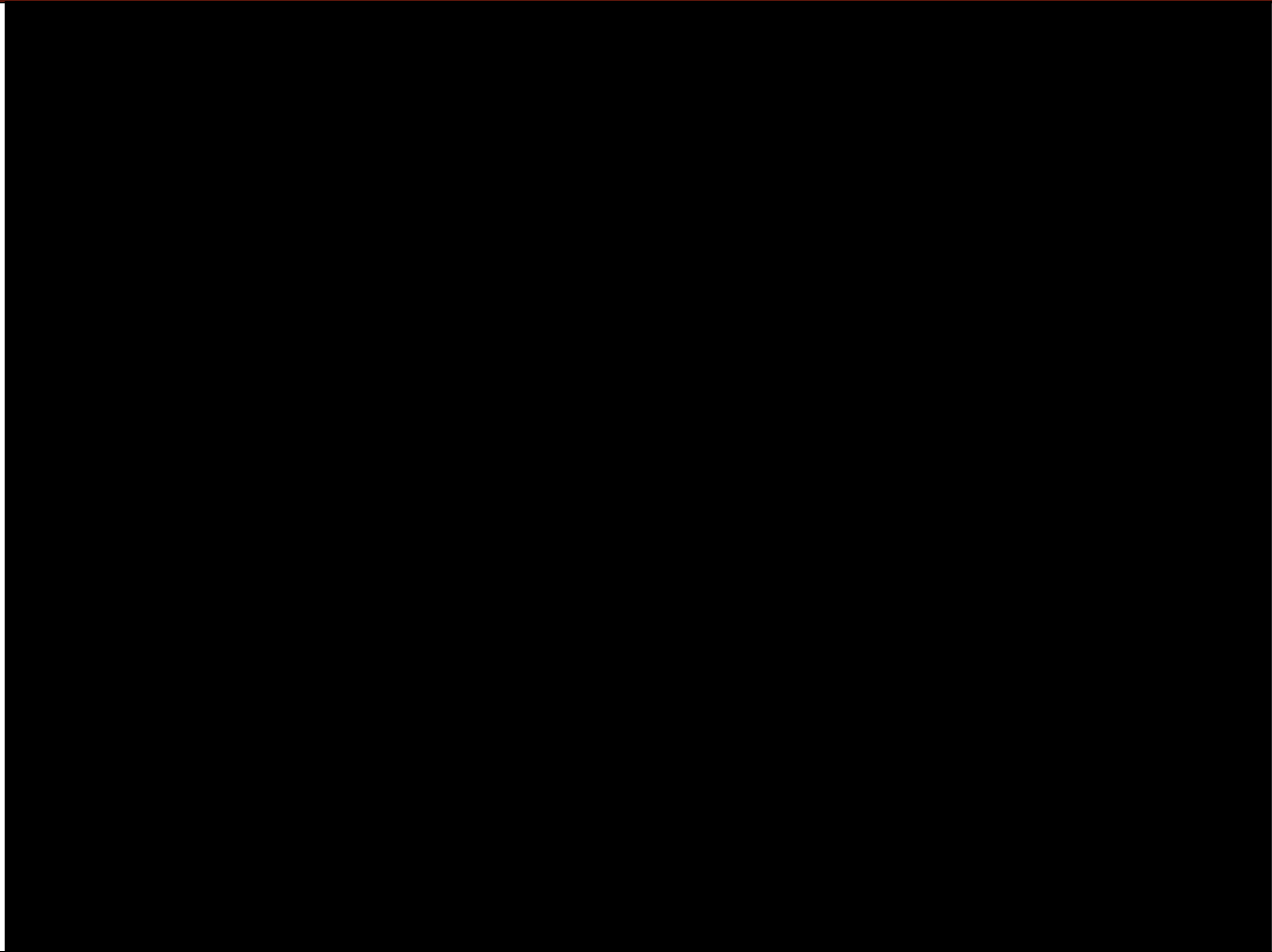
The reheat temperatures are very close or equal to the turbine inlet temperature.

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



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

Rankine Cycles (*the ideal reheat rankine cycle*)

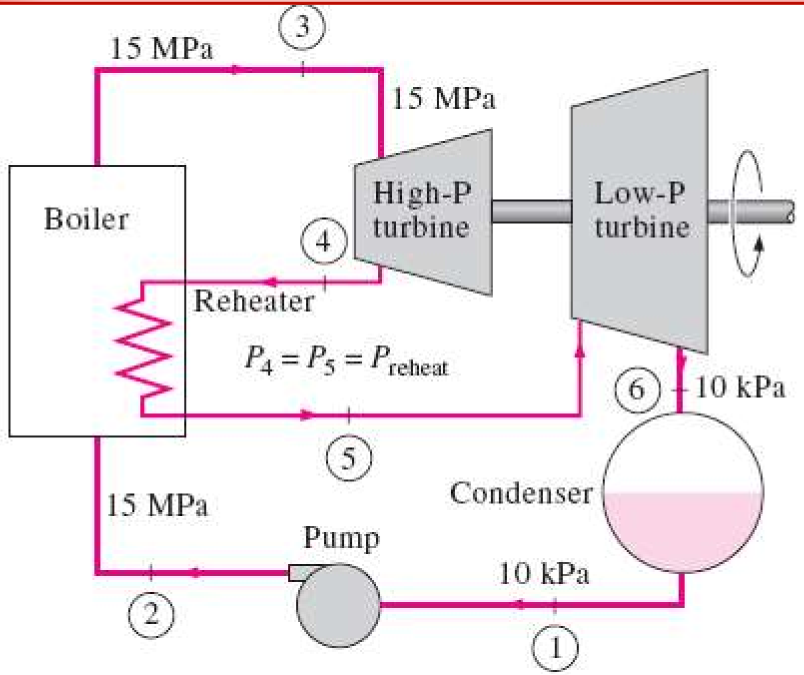


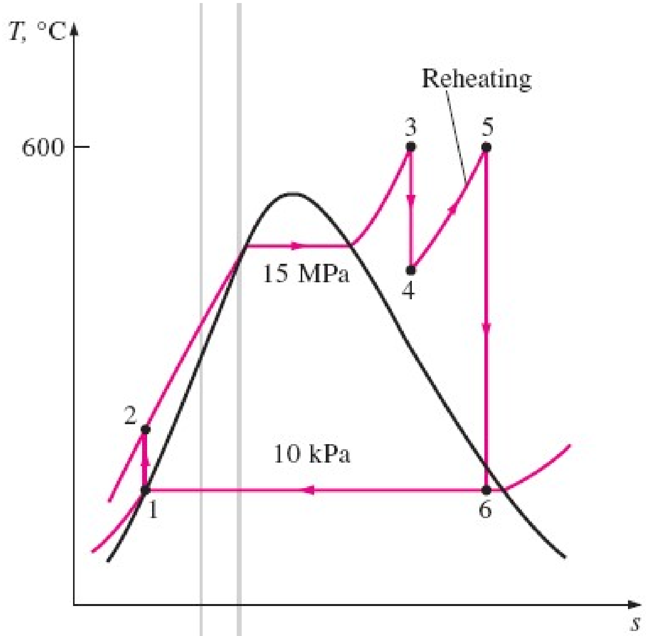
EXAMPLE 10–4

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

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

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





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

State 6:

$$P_6 = 10 \text{ kPa}$$

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

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

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

State 5:

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

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

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

State 1:

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

State 2:

$$P_2 = 15 \text{ MPa}$$
$$s_2 = s_1$$

$$w_{\text{pump, in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg}) \times [(15{,}000 - 10)\text{kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 15.14 \text{ kJ/kg}$$

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

State 3:

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

State 4:

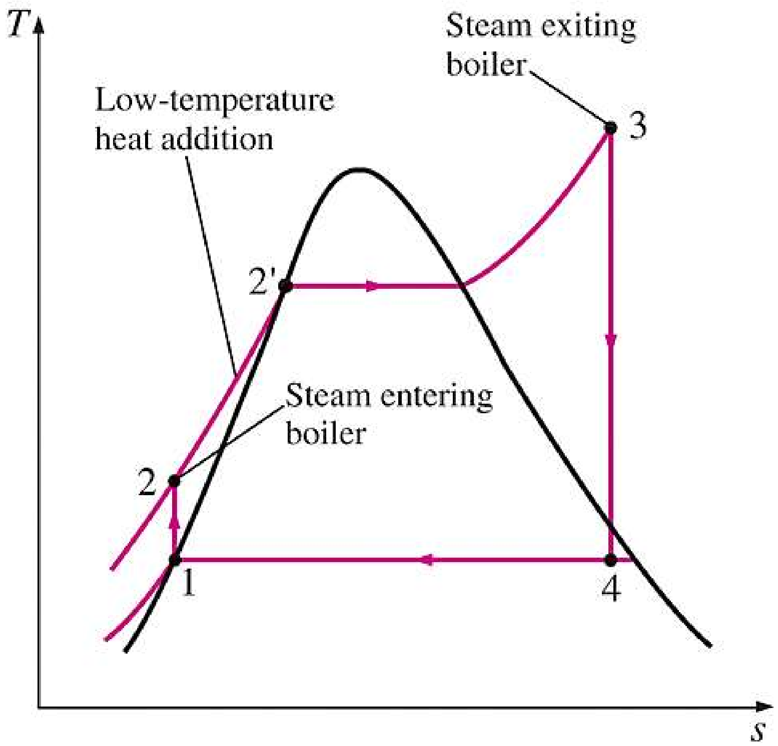
$$\left.\begin{array}{l} P_4 = 4 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} \quad \begin{array}{l} h_4 = 3155.0 \text{ kJ/kg} \\ (T_4 = 375.5°\text{C}) \end{array}$$

$$q_{in} = (h_3 - h_2) + (h_5 - h_4)$$
$$= (3583.1 - 206.95)\ \text{kJ/kg} + (3674.9 - 3155.0)\ \text{kJ/kg}$$
$$= 3896.1\ \text{kJ/kg}$$
$$q_{out} = h_6 - h_1 = (2335.1 - 191.81)\ \text{kJ/kg}$$
$$= 2143.3\ \text{kJ/kg}$$

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

Discussion This problem was solved in Example 10–3*c* 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



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)**.

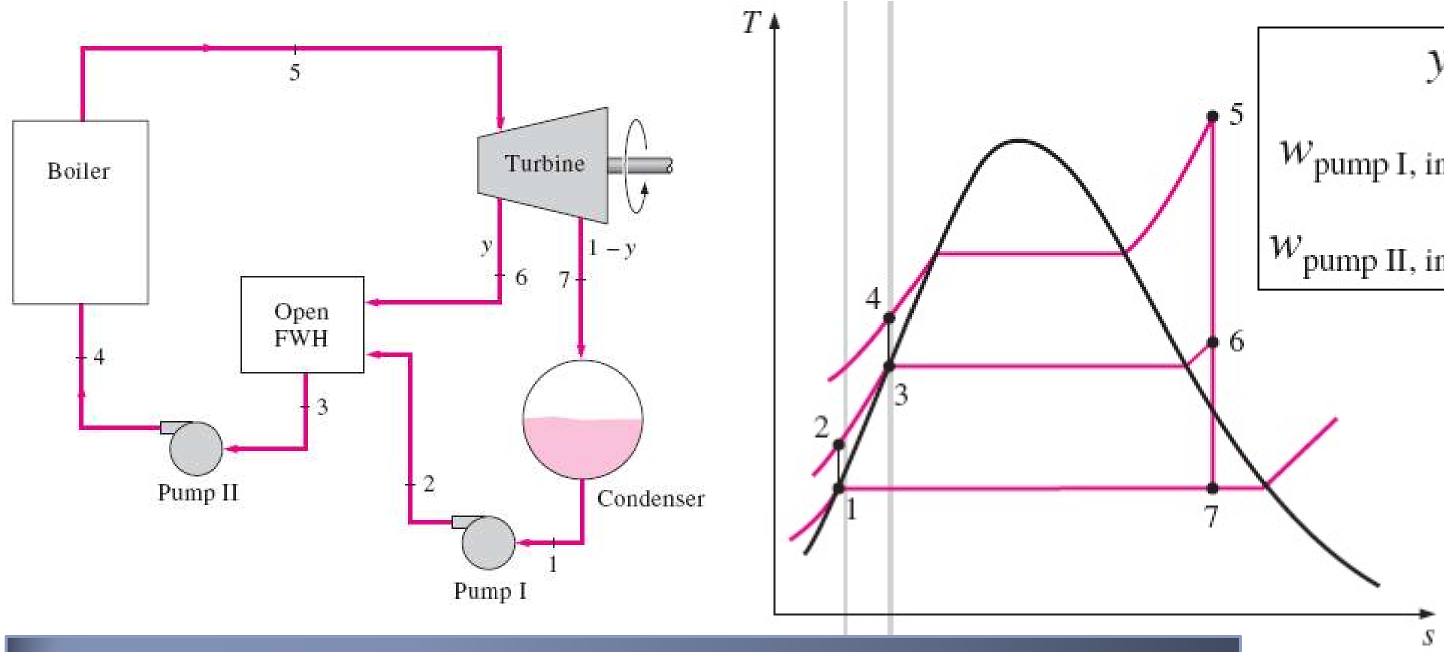
A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (open feedwater heaters) or without mixing them (closed feedwater heaters).

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.

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

fraction of steam extracted

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

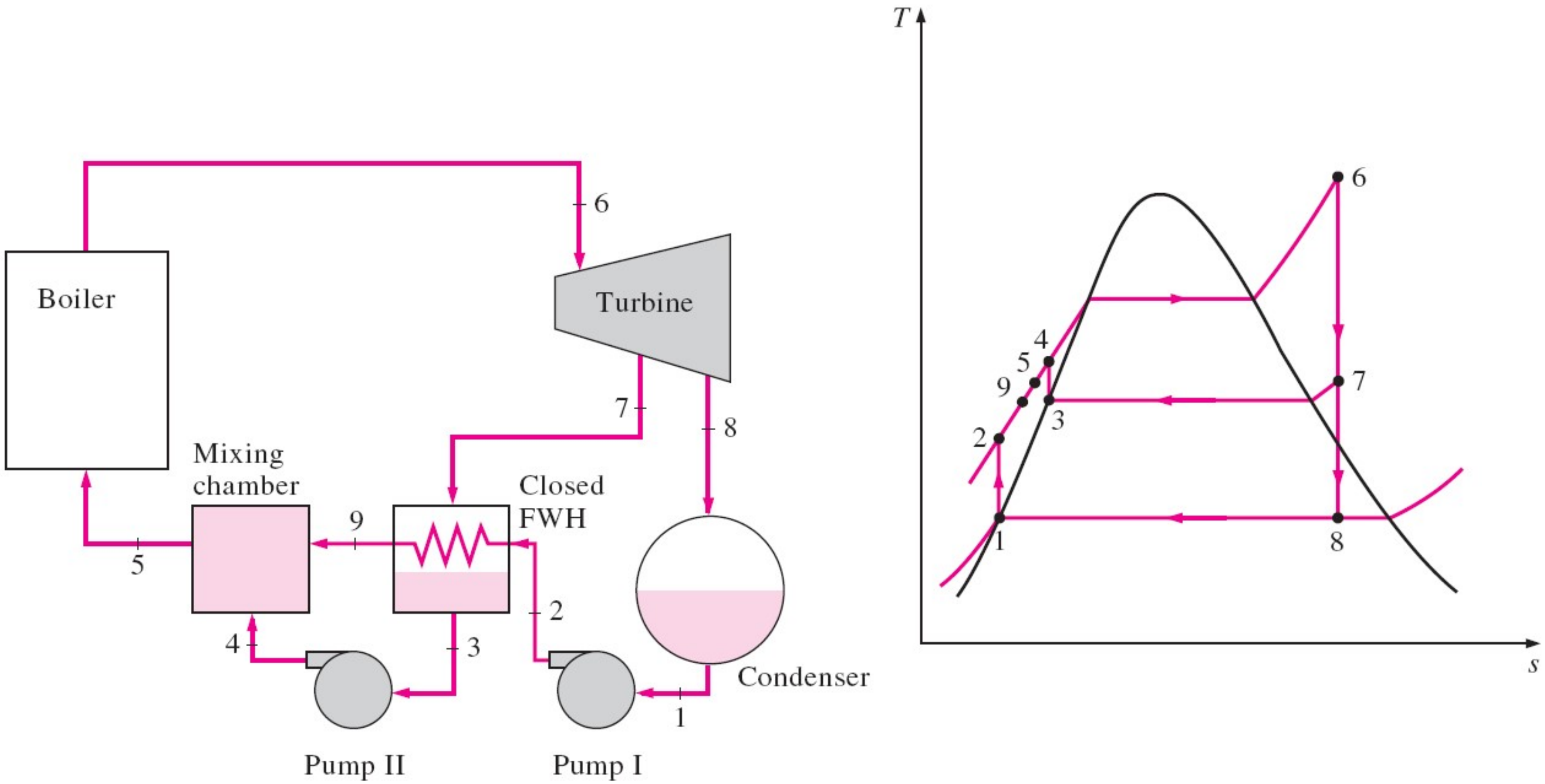


The ideal regenerative Rankine cycle with an open feedwater heater.

Rankine Cycles (Open Feedwater Heaters)

Closed Feedwater Heaters

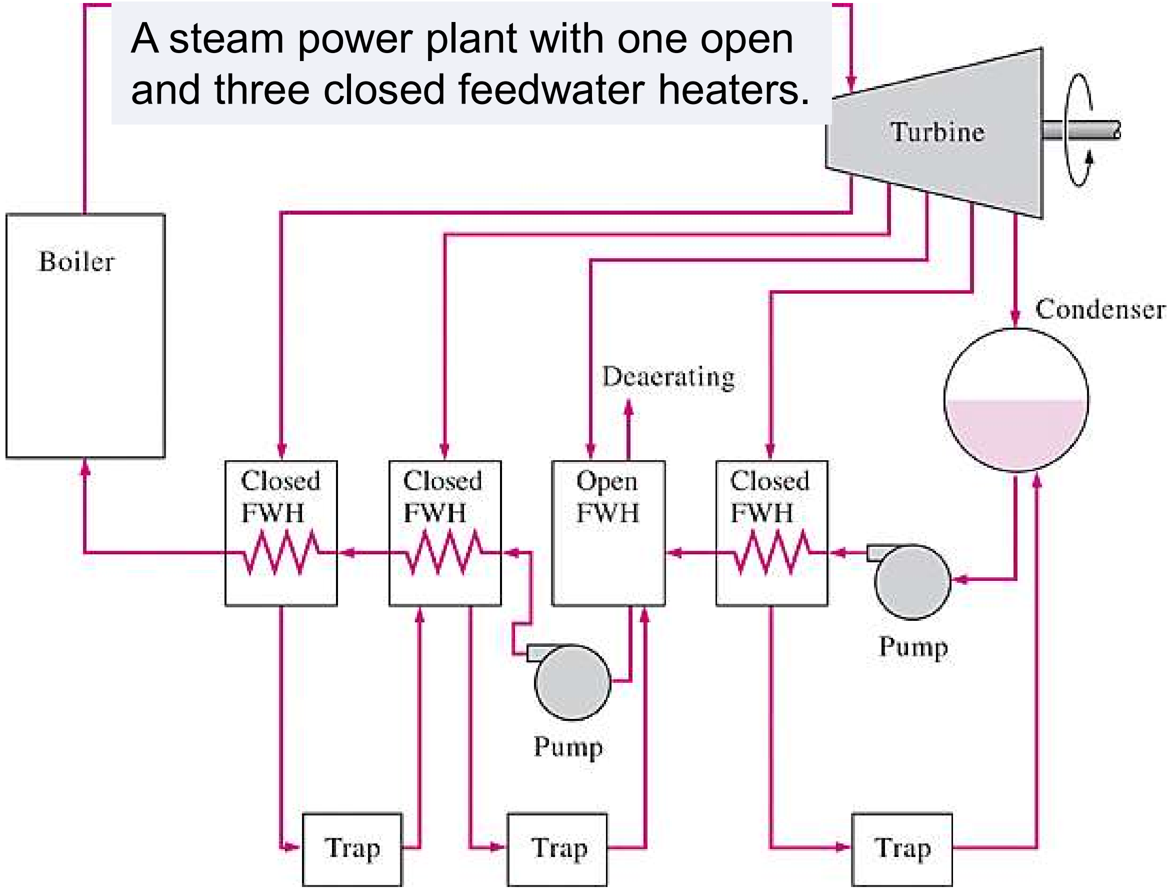
Another type of feedwater heater frequently used in steam power plants is the **closed feedwater heater**, in which heat is transferred from the extracted steam to the feedwater without any mixing taking place. The two streams now can be at different pressures, since they do not mix.



The ideal regenerative Rankine cycle with a closed feedwater heater.

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.

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



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.

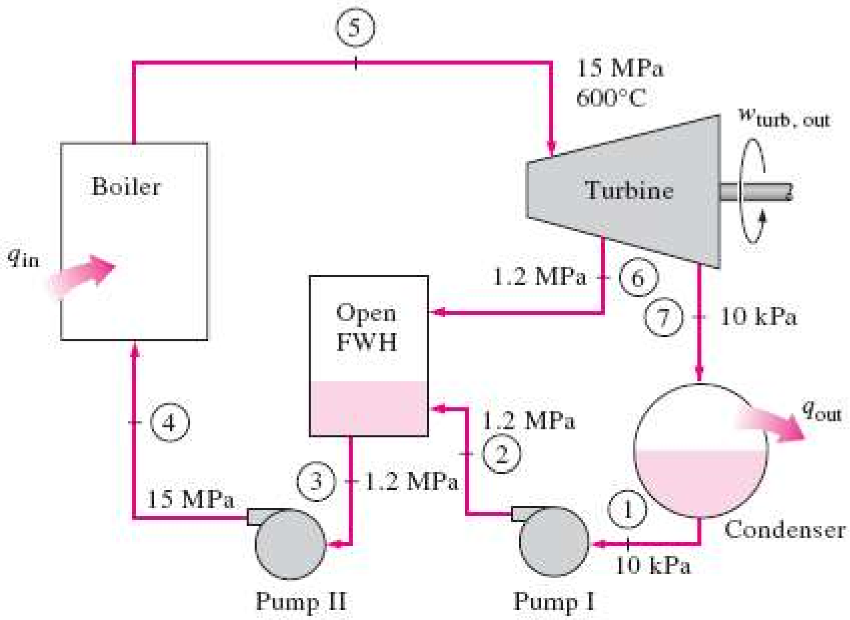
Most steam power plants use a combination of open and closed feedwater heaters.

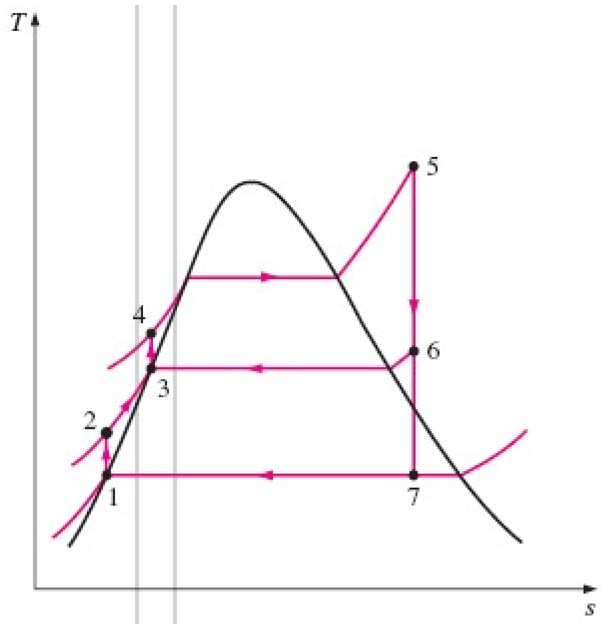
EXAMPLE 10–5

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

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

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





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

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

$$s_2 = s_1$$

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

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

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

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

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

$$s_4 = s_3$$

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

$$= (0.001138 \text{ m}^3/\text{kg})[(15{,}000 - 1200) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 15.70 \text{ kJ/kg}$$

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

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

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

$$\textit{State 7:}\quad P_7 = 10\ \text{kPa} \quad \begin{array}{l} s_7 = s_5 \qquad x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.6796 - 0.6492}{7.4996} = 0.8041 \\ h_7 = h_f + x_7 h_{fg} = 191.81 + 0.8041(2392.1) = 2115.3\ \text{kJ/kg} \end{array}$$

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

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

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

$$y h_6 + (1 - y) h_2 = 1(h_3)$$

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

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

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

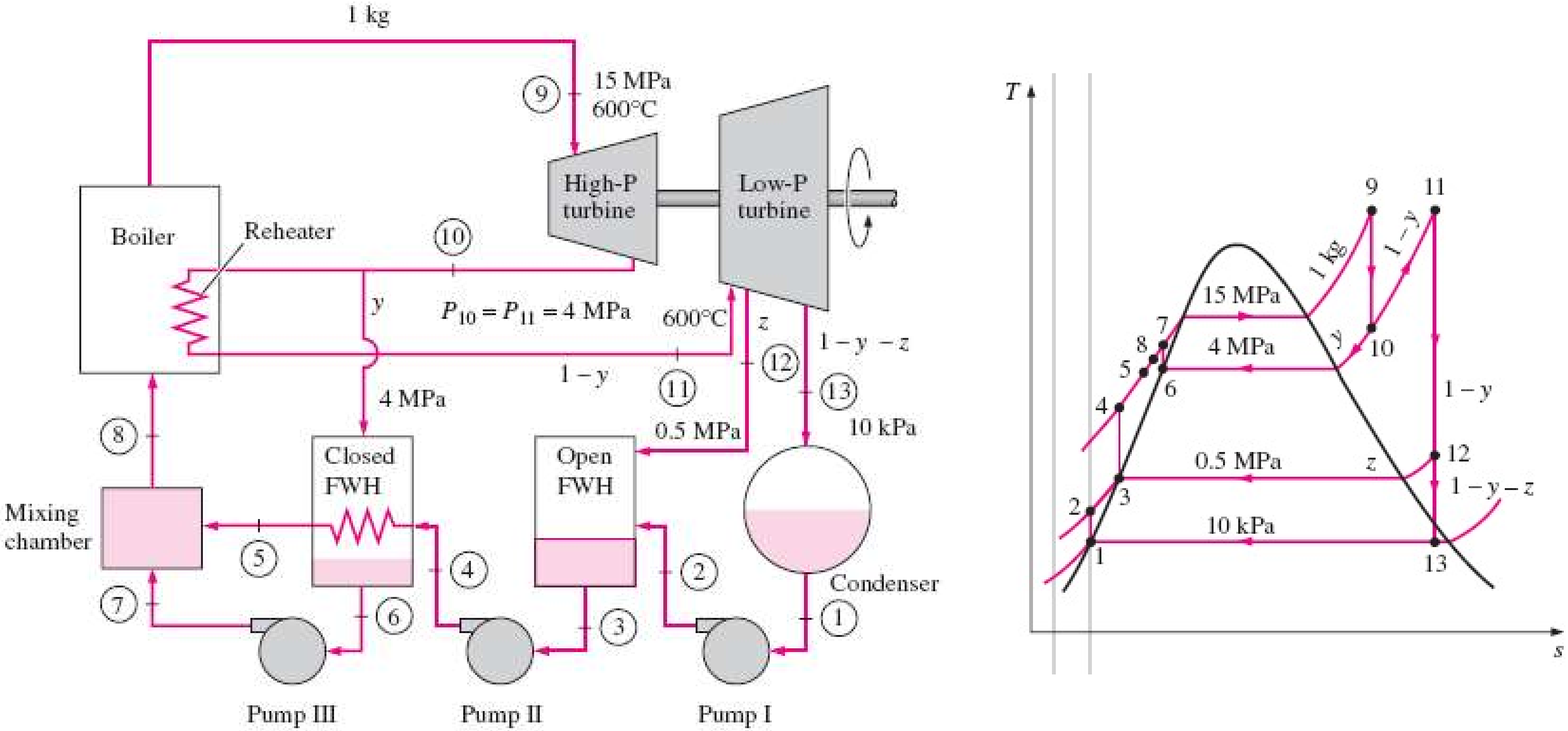
$$q_{out} = (1 - y)(h_7 - h_1) = (1 - 0.2270)(2115.3 - 191.81)\text{ kJ/kg}$$
$$= 1486.9\text{ kJ/kg}$$

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

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.

EXAMPLE 10–6

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



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

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

$h_1 = 191.81$ kJ/kg	$h_3 = 640.09$ kJ/kg
$h_2 = 192.30$ kJ/kg	$h_4 = 643.92$ kJ/kg
$h_5 = 1087.4$ kJ/kg	$h_{11} = 3674.9$ kJ/kg
$h_6 = 1087.4$ kJ/kg	$h_{12} = 3014.8$ kJ/kg
$h_7 = 1101.2$ kJ/kg	$h_{13} = 2335.7$ kJ/kg
$h_8 = 1089.8$ kJ/kg	$w_{\text{pump I, in}} = 0.49$ kJ/kg
$h_9 = 3583.1$ kJ/kg	$w_{\text{pump II, in}} = 3.83$ kJ/kg
$h_{10} = 3155.0$ kJ/kg	$w_{\text{pump III, in}} = 13.77$ kJ/kg

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

Closed feedwater heater:

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

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

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

Open feedwater heater:

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

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

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

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

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

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

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

$$= 1089.8\text{ kJ/kg}$$

$$q_{in} = (h_9 - h_8) + (1 - y)(h_{11} - h_{10})$$
$$= (3583.1 - 1089.8)\ \text{kJ/kg} + (1 - 0.1766)(3674.9 - 3155.0)\ \text{kJ/kg}$$
$$= 2921.4\ \text{kJ/kg}$$

$$q_{out} = (1 - y - z)(h_{13} - h_1)$$
$$= (1 - 0.1766 - 0.1306)(2335.7 - 191.81)\ \text{kJ/kg} = 1485.3\ \text{kJ/kg}$$

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

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

The thermal efficiency of this cycle could also be determined from

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

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

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

Exergy destruction for a steady-flow system

$$\dot{X}_{\text{dest}} = T_0 \dot{S}_{\text{gen}} = T_0(\dot{S}_{\text{out}} - \dot{S}_{\text{in}}) = T_0\left(\sum \dot{m}_e s_e + \frac{\dot{Q}_{\text{out}}}{T_{b,\text{ out}}} - \sum \dot{m}_i s_i - \frac{\dot{Q}_{\text{in}}}{T_{b,\text{ in}}}\right)$$

Steady-flow, one-inlet, one-exit

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

Exergy destruction of a cycle

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

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

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

Stream exergy

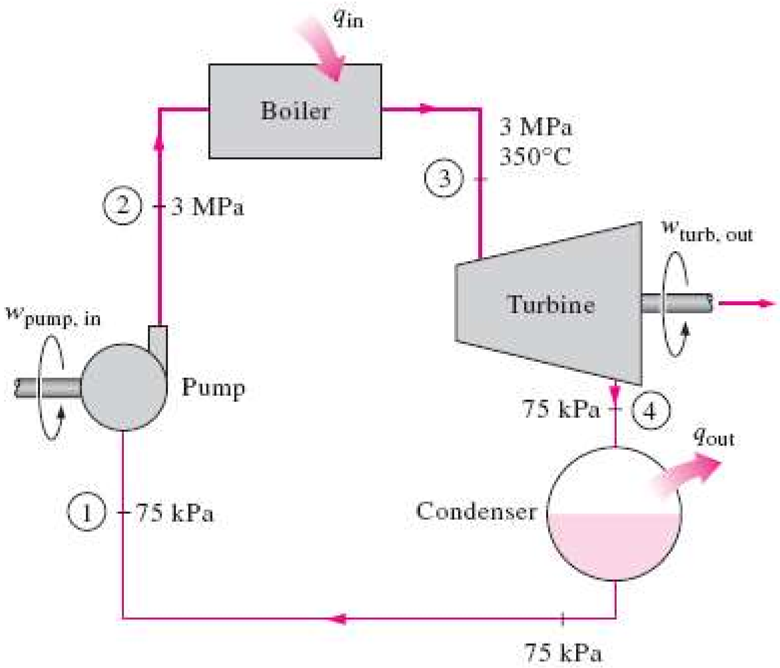
$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \qquad (\text{kJ/kg})$$

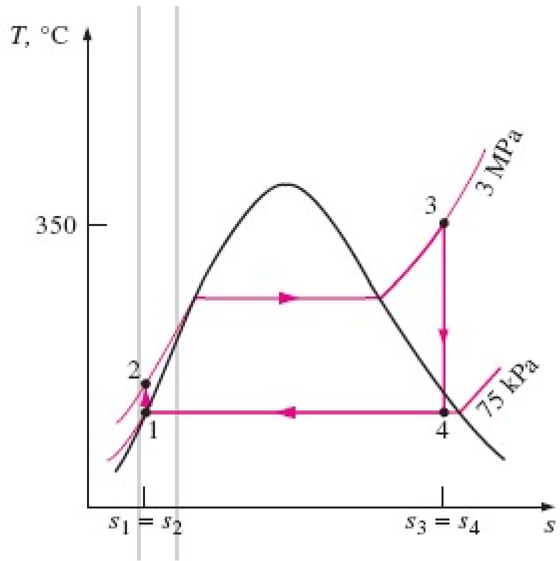
A second-law analysis of vaporpower cycles reveals where the largest irreversibilities occur and where to start improvements.

EXAMPLE 10–7

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

Solution The Rankine cycle analyzed in Example 10–1 is reconsidered. For specified source and sink temperatures, the exergy destruction associated with the cycle and exergy of the steam at turbine exit are to be determined.





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,

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

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

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

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

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

$$= \mathbf{1109.7 \text{ kJ/kg}}$$

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

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

$$= \mathbf{414.4 \text{ kJ/kg}}$$

Therefore, the irreversibility of the cycle is

$$
\begin{aligned}
x_{\text{dest, cycle}} &= x_{\text{dest, 12}} + x_{\text{dest, 23}} + x_{\text{dest, 34}} + x_{\text{dest, 41}} \\
&= 0 + 1109.7 \text{ kJ/kg} + 0 + 414.4 \text{ kJ/kg} \\
&= \mathbf{1524.1 \text{ kJ/kg}}
\end{aligned}
$$

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

$$
\begin{aligned}
\psi_4 &= (h_4 - h_0) - T_0(s_4 - s_0) + \cancelto{0}{\frac{V_4^2}{2}} + \cancelto{0}{gz_4} \\
&= (h_4 - h_0) - T_0(s_4 - s_0)
\end{aligned}
$$

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

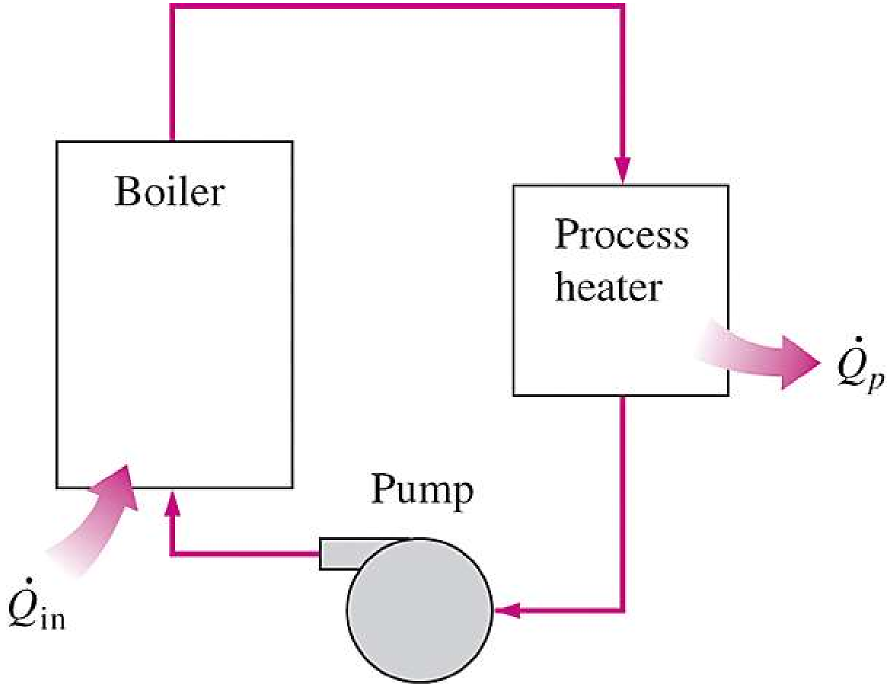
$$s_0 = s_{@\ 290\text{ K},\ 100\text{ kPa}} \cong s_{f\ @\ 290\text{ K}} = 0.2533 \text{ kJ/kg} \cdot \text{K}$$

$$
\begin{aligned}
\psi_4 &= (2403.0 - 71.355) \text{ kJ/kg} - (290 \text{ K})[(6.7450 - 0.2533) \text{ kJ/kg} \cdot \text{K}] \\
&= \mathbf{449.1 \text{ kJ/kg}}
\end{aligned}
$$

Discussion Note that 449.1 kJ/kg of work could be obtained from the steam leaving the turbine if it is brought to the state of the surroundings in a reversible manner.

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.



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.

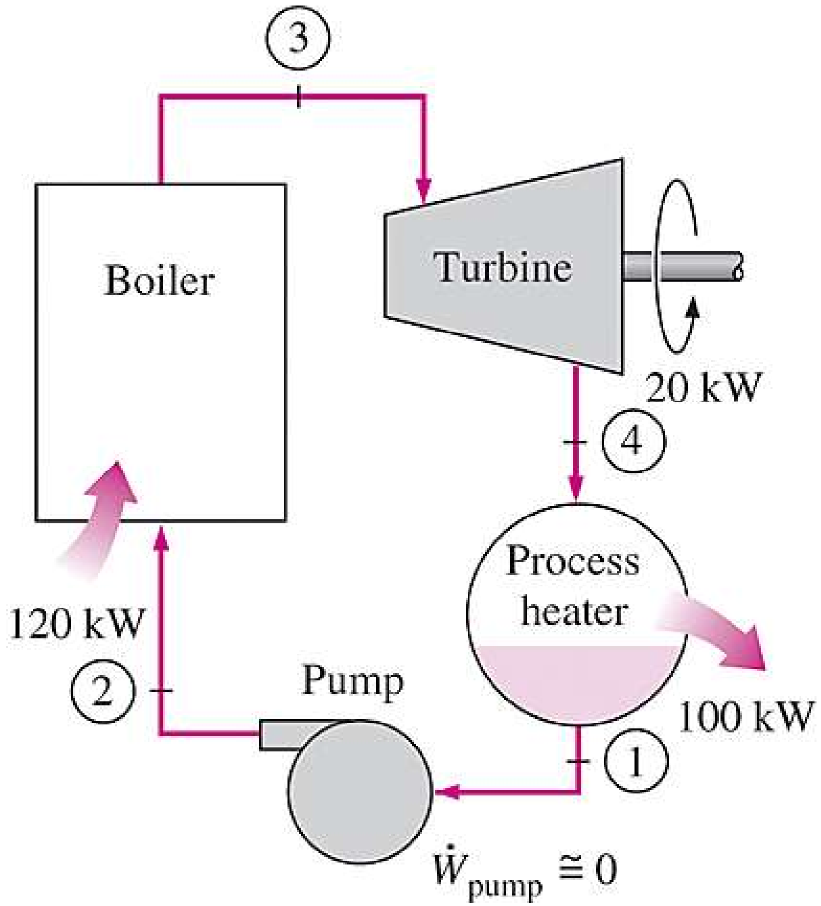
The result is a plant that produces electricity while meeting the process-heat requirements of certain industrial processes (cogeneration plant)

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

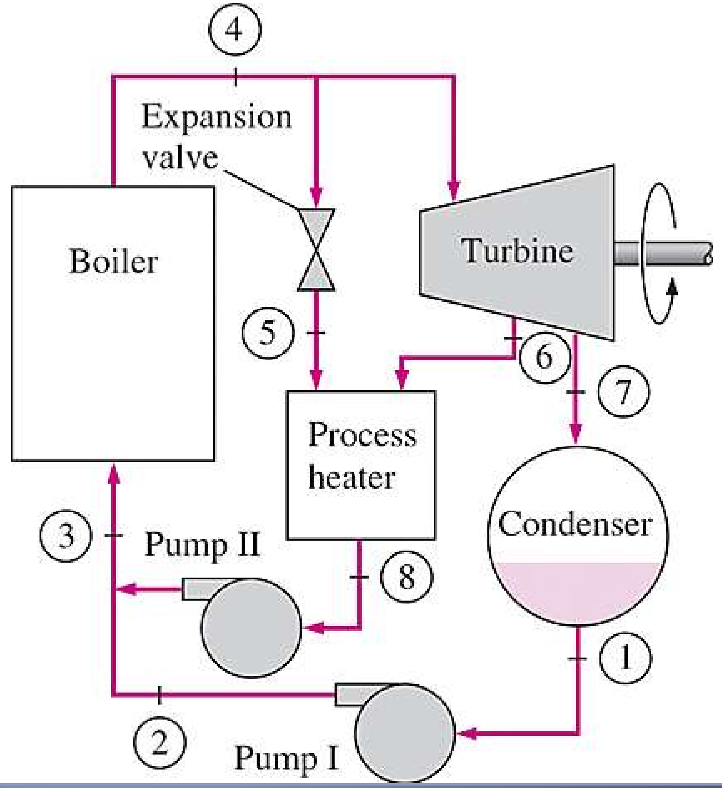
$$\epsilon_u = \frac{\text{Net work output} + \text{Process heat delivered}}{\text{Total heat input}} = \frac{\dot{W}_{net} + \dot{Q}_p}{\dot{Q}_{in}}$$

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



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.



A cogeneration plant with adjustable loads.

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

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

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

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

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.

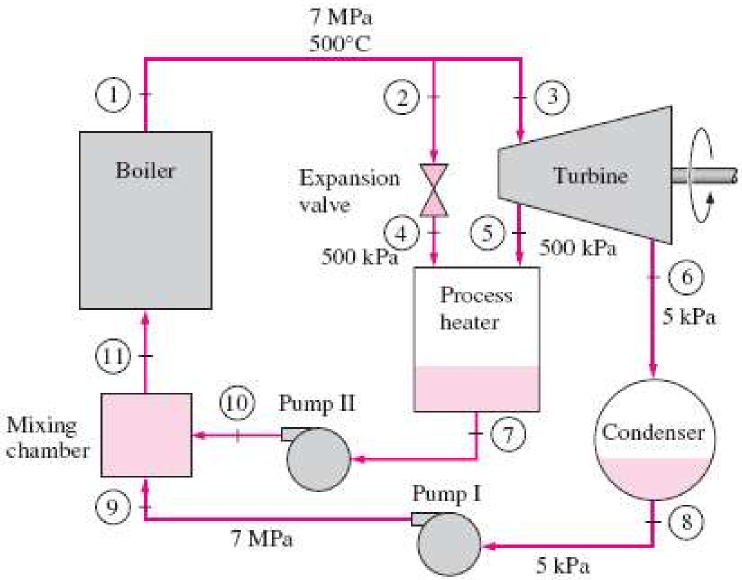
If this is not sufficient, some steam leaving the boiler is throttled by an expansion or pressure-reducing valve to the extraction pressure P_6 and is directed to the process-heating unit.

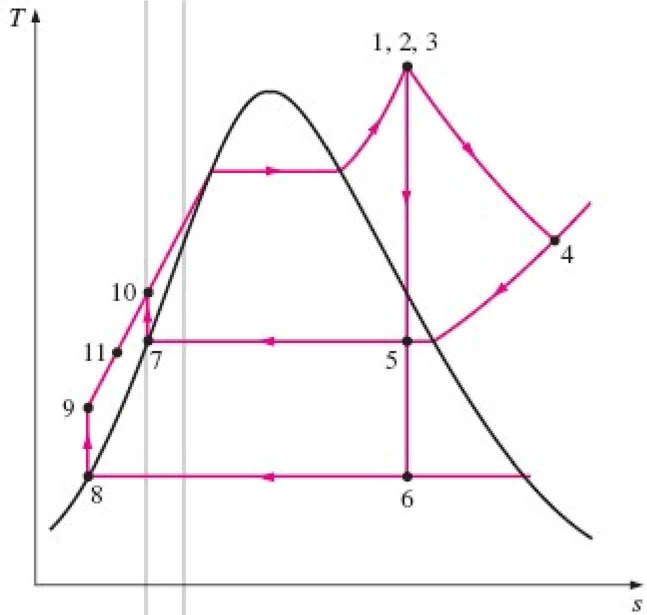
Maximum process heating is realized when all the steam leaving the boiler passes through the PRV (m_5= m_4). No power is produced in this mode.

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.

EXAMPLE 10–8

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





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

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

$$w_{\text{pump I, in}} = v_8(P_9 - P_8) = (0.001005\ \text{m}^3/\text{kg})[(7000 - 5)\text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 7.03\ \text{kJ/kg}$$

$$w_{\text{pump II, in}} = v_7(P_{10} - P_7) = (0.001093\ \text{m}^3/\text{kg})[(7000 - 500)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 7.10\ \text{kJ/kg}$$

$$h_1 = h_2 = h_3 = h_4 = 3411.4\ \text{kJ/kg}$$ $$h_5 = 2739.3\ \text{kJ/kg}$$ $$h_6 = 2073.0\ \text{kJ/kg}$$

$$h_7 = h_{f\ @\ 500\ \text{kPa}} = 640.09\ \text{kJ/kg}$$ $$h_8 = h_{f\ @\ 5\ \text{kPa}} = 137.75\ \text{kJ/kg}$$

$$h_9 = h_8 + w_{\text{pump I, in}} = (137.75 + 7.03)\ \text{kJ/kg} = 144.78\ \text{kJ/kg}$$
$$h_{10} = h_7 + w_{\text{pump II, in}} = (640.09 + 7.10)\ \text{kJ/kg} = 647.19\ \text{kJ/kg}$$

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

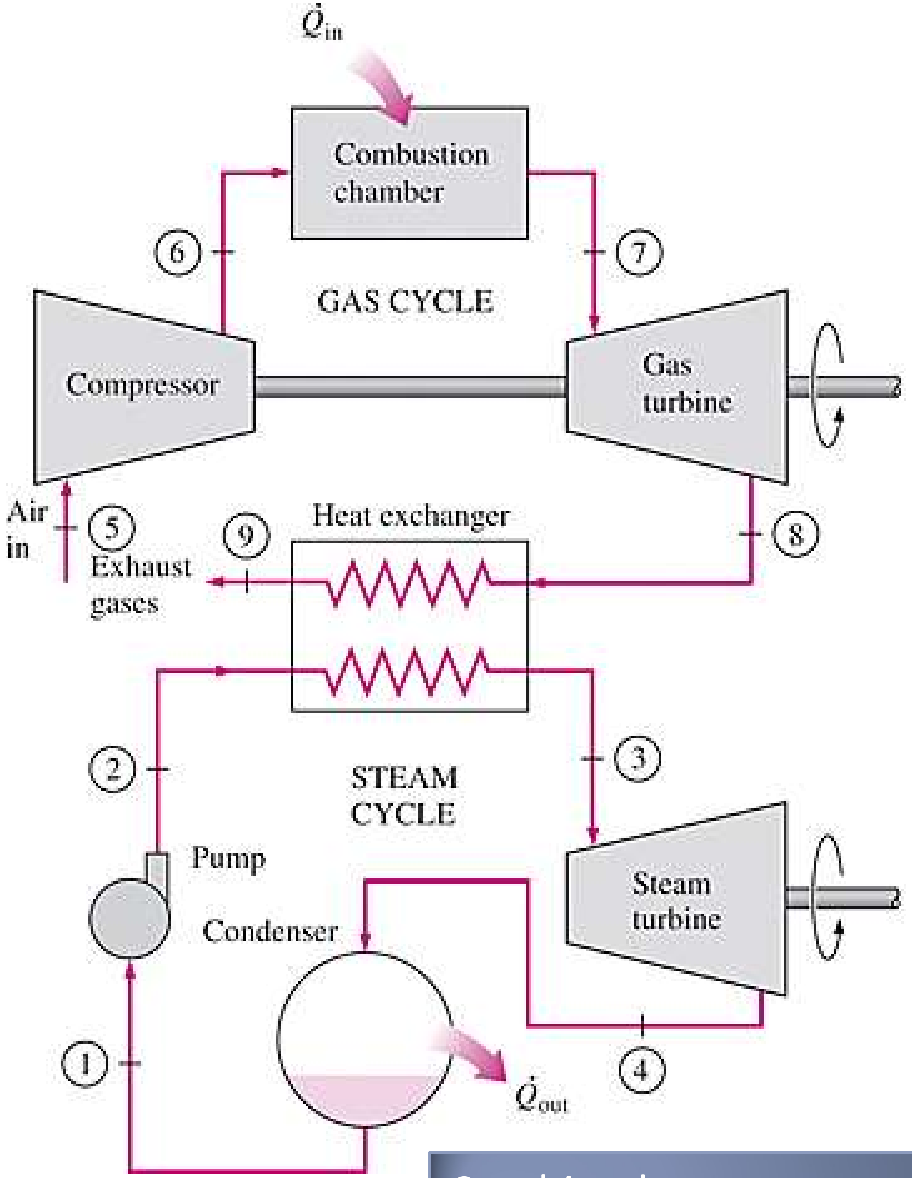
$$\dot{m}_7 = \dot{m}_4 + \dot{m}_5 = 1.5 + 10.5 = 12\ \text{kg/s}$$

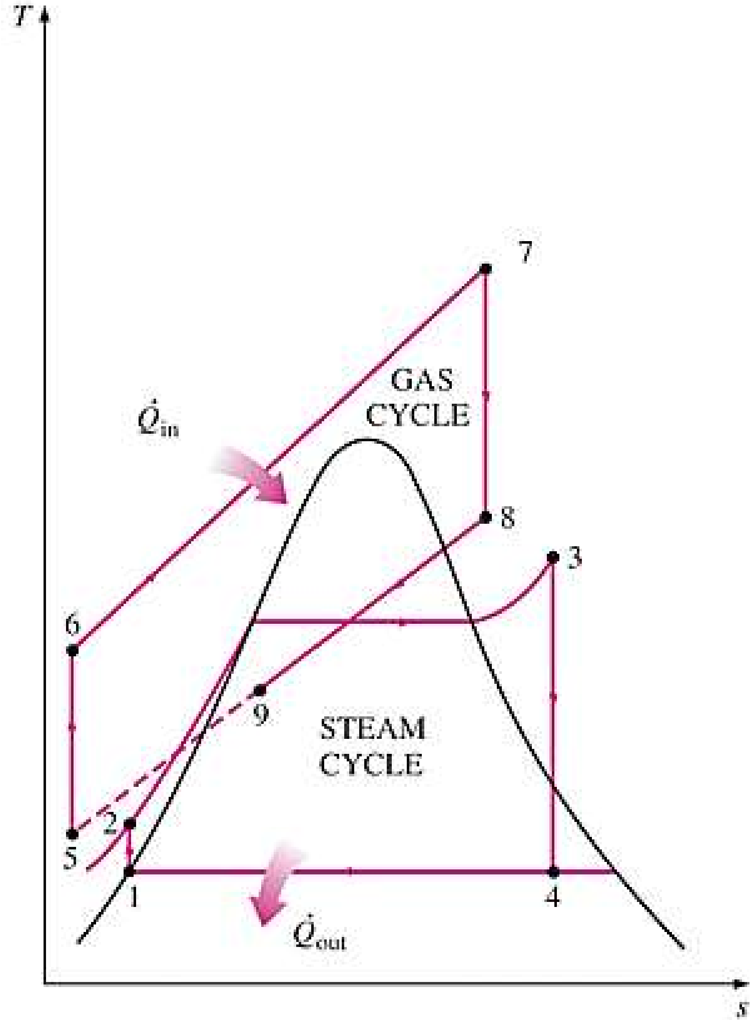
$$\begin{aligned}\dot{Q}_{p,\,out} &= (1.5\ \text{kg/s})(3411.4\ \text{kJ/kg}) + (10.5\ \text{kg/s})(2739.3\ \text{kJ/kg}) \\ &\quad - (12\ \text{kg/s})(640.09\ \text{kJ/kg}) \\ &= \mathbf{26{,}199\ kW}\end{aligned}$$

Discussion Note that 26,199 kW of the heat transferred will be utilized in the process heater. We could also show that 10,966 kW of power is produced in this case, and the rate of heat input in the boiler is 42,970 kW. Thus the utilization factor is 86.5 percent.

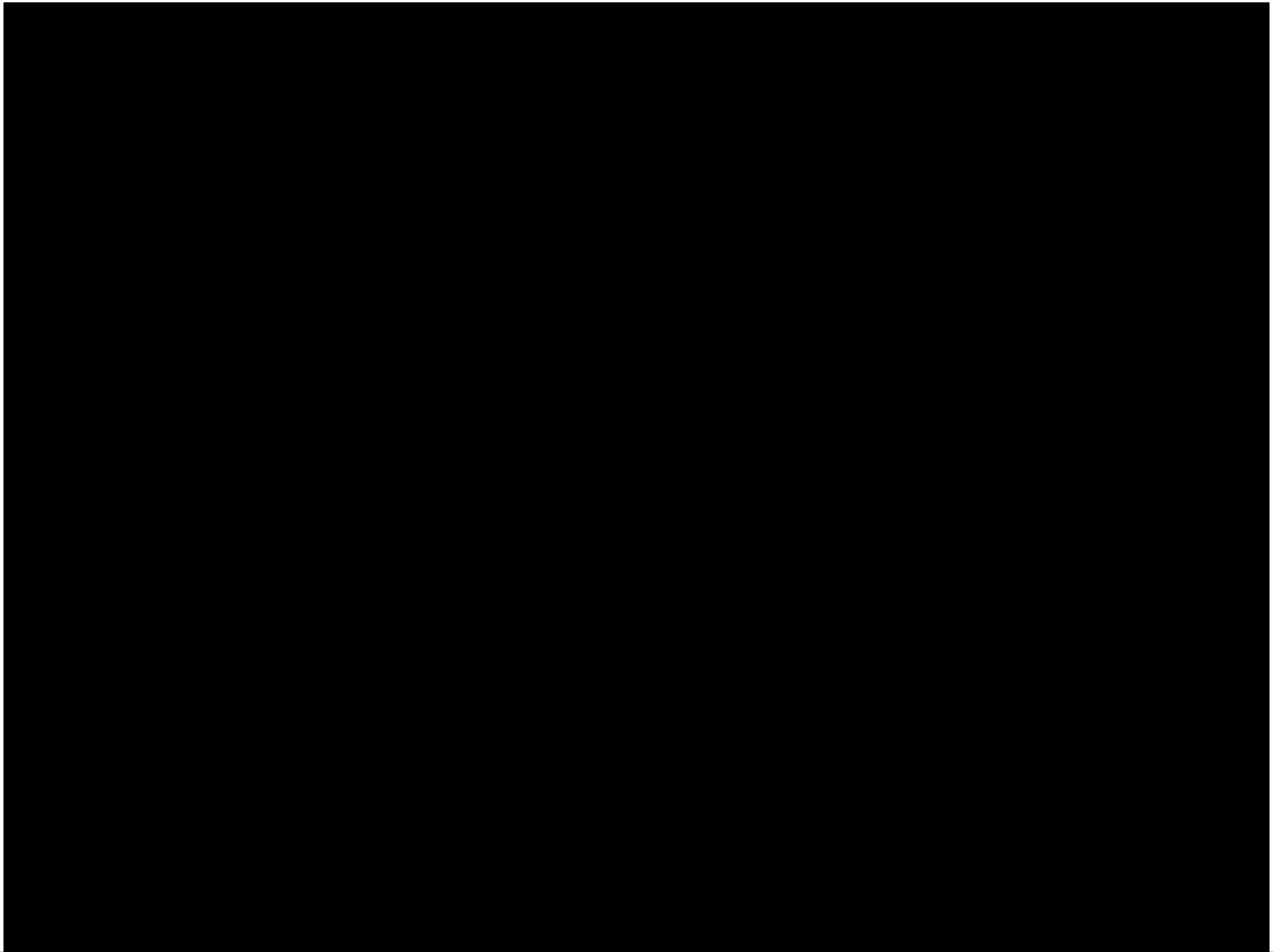
COMBINED GAS-VAPOR POWER CYCLES

- 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.





Combined gas–steam power plant.



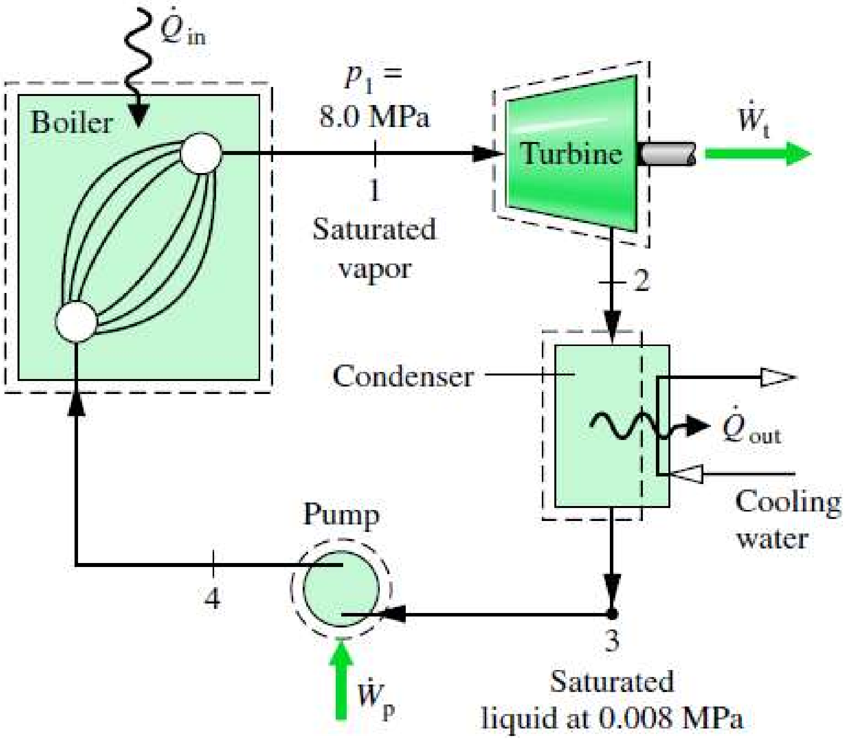
SUMMARY

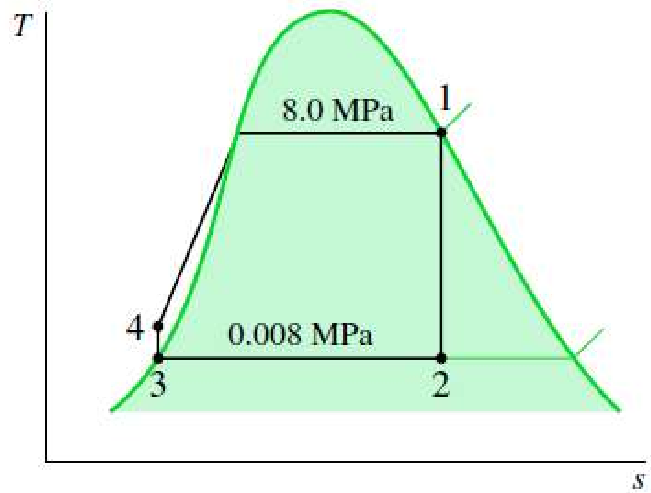
- The Carnot vapor cycle
- Rankine cycle: The ideal cycle for vapor power cycles
  - Energy analysis of the ideal Rankine cycle
- Deviation of actual vapor power cycles from idealized ones
- How can we increase the efficiency of the Rankine cycle?
  - Lowering the condenser pressure (*Lowers* $T_{low,avg}$)
  - 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

EXAMPLE 1

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,
(f) the mass flow rate of the condenser cooling water, in kg/ h, if cooling water enters the condenser at 15C and exits at 35C.





h_1= 2758.0 kJ/kg and s_1= 5.7432 kJ/kg K.

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

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

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

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

$$h_4 = 173.88 \text{ kJ/kg} + (1.0084 \times 10^{-3} \text{ m}^3\text{/kg})(8.0 - 0.008)\text{MPa}$$

$$= 173.88 + 8.06 = 181.94 \text{ kJ/kg}$$

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

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

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

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

$$\eta = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_{in}} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

$$= \frac{[(2758.0 - 1794.8) - (181.94 - 173.88)]\ \text{kJ/kg}}{(2758.0 - 181.94)\ \text{kJ/kg}}$$

$$= 0.371\ (37.1\%)$$

(b) The back work ratio is

$$\text{bwr} = \frac{\dot{W}_p}{\dot{W}_t} = \frac{h_4 - h_3}{h_1 - h_2} = \frac{(181.94 - 173.88)\ \text{kJ/kg}}{(2758.0 - 1794.8)\ \text{kJ/kg}}$$

$$= \frac{8.06}{963.2} = 8.37 \times 10^{-3}\ (0.84\%)$$

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

$$\dot{m} = \frac{\dot{W}_{cycle}}{(h_1 - h_2) - (h_4 - h_3)}$$

$$= \frac{(100\ \text{MW})|10^3\ \text{kW/MW}||3600\ \text{s/h}|}{(963.2 - 8.06)\ \text{kJ/kg}}$$

$$= 3.77 \times 10^5\ \text{kg/h}$$

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

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

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

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

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

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

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

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

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

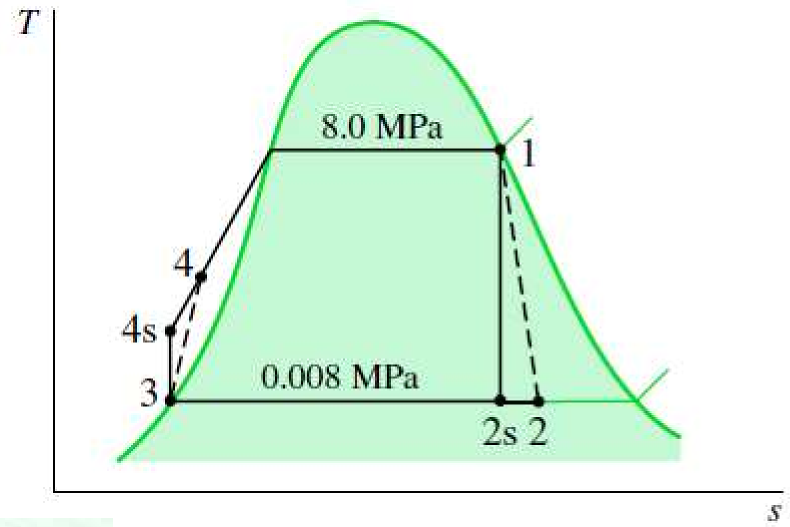
- ❖ Note that the back work ratio is relatively low for the Rankine cycle. In the present case, the work required to operate the pump is less than 1% of the turbine output.
- ❖ In this example, 62.9% of the energy added to the working fluid by heat transfer is subsequently discharged to the cooling water. 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.

EXAMPLE 2

Reconsider the vapor power cycle of Example 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 into the working fluid as it passes through the boiler, in MW,
(d) the rate of heat transfer 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 15C and exits as 35C. Discuss the effects on the vapor cycle of irreversibilities within the turbine and pump.

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



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

$$h_3 = 173.88 \text{ kJ/kg.}$$

$$h_4 = h_3 + \dot{W}_p/\dot{m} \qquad \eta_p = \frac{(\dot{W}_p/\dot{m})_s}{(\dot{W}_p/\dot{m})}$$

$$\frac{\dot{W}_p}{\dot{m}} = \frac{v_3(p_4 - p_3)}{\eta_p} \qquad \frac{\dot{W}_p}{\dot{m}} = \frac{8.06\,\text{kJ/kg}}{0.85} = 9.48 \text{ kJ/kg}$$

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

(a) the thermal efficiency

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

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

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

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

(b) the mass flow rate of the steam is

$$\dot{m} = \frac{\dot{W}_{cycle}}{(h_1 - h_2) - (h_4 - h_3)}$$

$$= \frac{(100\ \text{MW})|3600\ \text{s/h}|\,|10^3\ \text{kW/MW}|}{(818.7 - 9.48)\ \text{kJ/kg}} = 4.449 \times 10^5\ \text{kg/h}$$

(c) the rate of heat transfer into the working fluid as it passes through the boiler,

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

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

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

$$\dot{Q}_{out} = \dot{m}(h_2 - h_3)$$

$$= \frac{(4.449 \times 10^5 \text{ kg/h})(1939.3 - 173.88) \text{ kJ/kg}}{|3600 \text{ s/h}||10^3 \text{ kW/MW}|} = 218.2 \text{ MW}$$

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

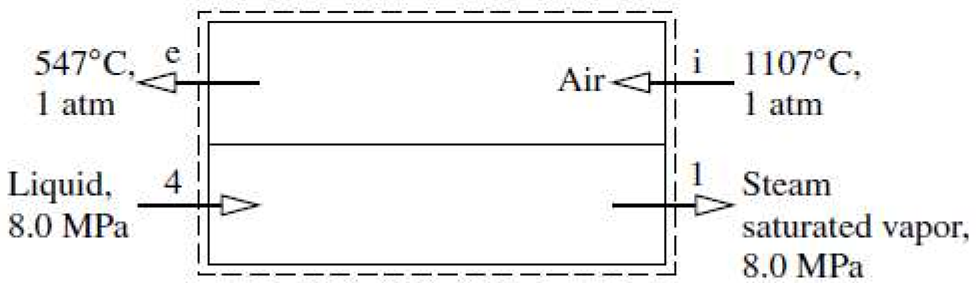
$$\dot{m}_{cw} = \frac{\dot{m}(h_2 - h_3)}{(h_{cw,out} - h_{cw,in})}$$

$$= \frac{(218.2 \text{ MW})|10^3 \text{ kW/MW}||3600 \text{ s/h}|}{(146.68 - 62.99) \text{ kJ/kg}} = 9.39 \times 10^6 \text{ kg/h}$$

EXAMPLE 3 HEAT EXCHANGER EXERGY ANALYSIS

The heat exchanger unit of the boiler of Example 2 has a stream of water entering as a liquid at 8.0 MPa and exiting as a saturated vapor at 8.0 MPa. In a separate stream, gaseous products of combustion cool at a constant pressure of 1 atm from 1107 to 547C. The gaseous stream can be modeled as air as an ideal gas. Let *T*0 22C, *p*0 1 atm. Determine

(a) the net rate at which exergy is carried into the heat exchanger unit by the gas stream, in MW,
(b) the net rate at which exergy is carried from the heat exchanger by the water stream, in MW,
(c) the rate of exergy destruction, in MW,
(d) the exergetic efficiency given by Eq. 7.45.



$$\dot{m}_i = \dot{m}_e \qquad \text{(air)}$$
$$\dot{m}_4 = \dot{m}_1 \qquad \text{(water)}$$

1. The control volume shown in the accompanying figure operates at steady state with .
2. Kinetic and potential energy effects can be ignored.
3. The gaseous combustion products are modeled as air as an ideal gas.
4. The air and the water each pass through the steam generator at constant pressure.
5. Only 69% of the exergy entering the plant with the fuel remains after accounting for the stack loss and combustion exergy destruction.
6. T_0= 22C, p_0=1 atm.

$$0 = \cancelto{0}{\dot{Q}_{cv}} - \cancelto{0}{\dot{W}_{cv}} + \dot{m}_a(h_i - h_e) + \dot{m}(h_4 - h_1)$$

$$h_1 = 2758 \text{ kJ/kg and } h_4 = 183.36 \text{ kJ/kg.}$$

$$\frac{\dot{m}_a}{\dot{m}} = \frac{h_1 - h_4}{h_i - h_e} \qquad \frac{\dot{m}_a}{\dot{m}} = \frac{2758 - 183.36}{1491.44 - 843.98} = 3.977 \frac{\text{kg (air)}}{\text{kg (steam)}}$$

(a) The net exergy rate into the heat exchanger;

$$\begin{bmatrix} \text{net rate at which exergy} \\ \text{is carried in by the} \\ \text{gaseous stream} \end{bmatrix} = \dot{m}_a(\mathsf{e}_{fi} - \mathsf{e}_{fe})$$
$$= \dot{m}_a[h_i - h_e - T_0(s_i - s_e)]$$

$$\dot{m}_a(\mathsf{e}_{fi} - \mathsf{e}_{fe}) = (17.694 \times 10^5 \text{ kg/h})[(1491.44 - 843.98) \text{ kJ/kg} - (295 \text{ K})(3.34474 - 2.74504)$$
$$= \frac{8.326 \times 10^8 \text{ kJ/h}}{|3600 \text{ s/h}|}\left|\frac{1 \text{ MW}}{10^3 \text{ kJ/s}}\right| = 231.28 \text{ MW}$$

(b) The net exergy rate of the boiler;

$$\begin{bmatrix} \text{net rate at which exergy} \\ \text{is carried out by the} \\ \text{water stream} \end{bmatrix} = \dot{m}(\mathrm{e}_{f1} - \mathrm{e}_{f4})$$

$$= \dot{m}[h_1 - h_4 - T_0(s_1 - s_4)]$$

$s_1 = 5.7432\ \text{kJ/kg} \cdot \text{K}$ $h_4 = 183.36\ \text{kJ/kg}$ $s_4 = 0.5957\ \text{kJ/kg} \cdot \text{K}$

$$\dot{m}(\mathrm{e}_{f1} - \mathrm{e}_{f4}) = (4.449 \times 10^5)[(2758 - 183.36) - 295(5.7432 - 0.5957)]$$

$$= \frac{4.699 \times 10^8\ \text{kJ/h}}{|3600\ \text{s/h}|} \left| \frac{1\ \text{MW}}{10^3\ \text{kJ/s}} \right| = 130.53\ \text{MW}$$

(c) The rate of exergy destruction can be evaluated by reducing the exergy rate balance to obtain

$$\dot{\mathrm{E}}_d = \dot{m}_a(\mathrm{e}_{fi} - \mathrm{e}_{fe}) + \dot{m}(\mathrm{e}_{f4} - \mathrm{e}_{f1})$$

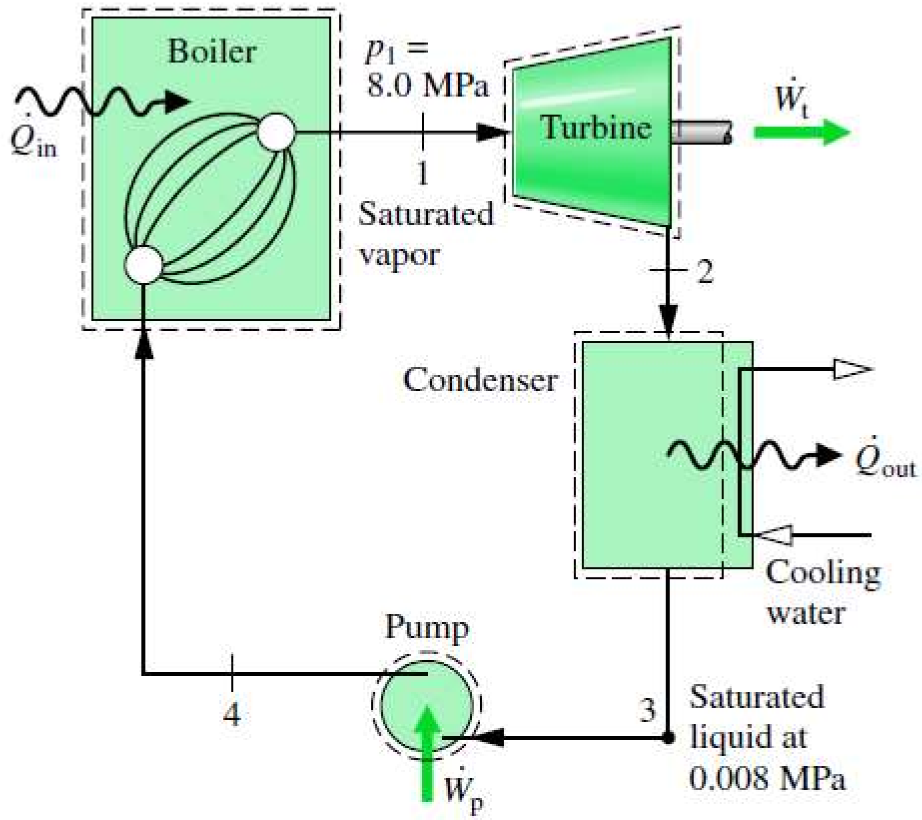
$$\dot{\mathrm{E}}_d = 231.28\ \text{MW} - 130.53\ \text{MW} = 100.75\ \text{MW}$$

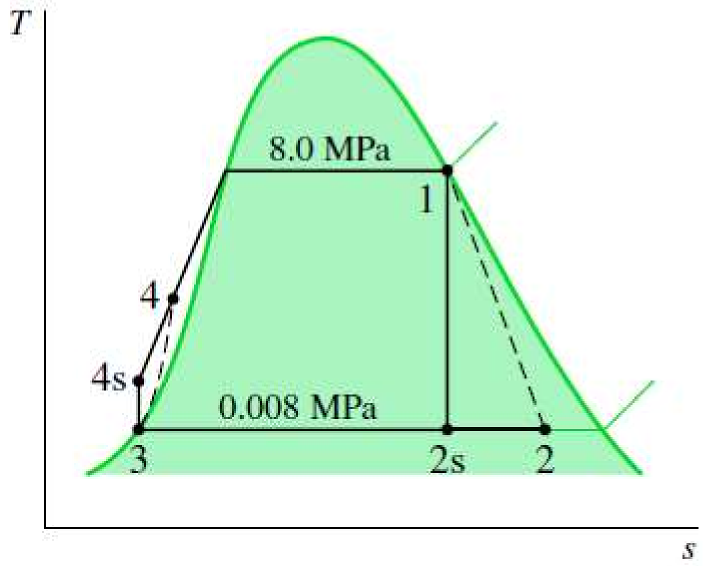
(d) The exergetic efficiency

$$\varepsilon = \frac{\dot{m}(\mathrm{e}_{f1} - \mathrm{e}_{f4})}{\dot{m}_a(\mathrm{e}_{fi} - \mathrm{e}_{fe})} = \frac{130.53\ \text{MW}}{231.28\ \text{MW}} = 0.564\ (56.4\%)$$

EXAMPLE 4 TURBINE AND PUMP EXERGY ANALYSIS

Reconsider the turbine and pump of Example 2. Determine for each of these components the rate at which exergy is destroyed, in MW. Express each result as a percentage of the exergy entering the plant with the fuel.





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

the rate of exergy destruction for the turbine

$$\dot{E}_d = \dot{m}T_0(s_2 - s_1)$$

$s_1 = 5.7432$ kJ/kg · K. $s_2 = 6.2021$ kJ/kg · K $h_2 = 1939.3$ kJ/kg

$$\dot{E}_d = (4.449 \times 10^5 \text{ kg/h})(295 \text{ K})(6.2021 - 5.7432)(\text{kJ/kg} \cdot \text{K})$$

$$= \left(0.602 \times 10^8 \frac{\text{kJ}}{\text{h}}\right)\left|\frac{1 \text{ h}}{3600 \text{ s}}\right|\left|\frac{1 \text{ MW}}{10^3 \text{ kJ/s}}\right| = 16.72 \text{ MW}$$

the net rate at which exergy is supplied by the cooling combustion gases is 231.28 MW. The turbine rate of exergy destruction expressed as a percentage of this is (16.72/231.28)(100%)= **7.23** %.

the exergy destruction rate for the pump

$$\dot{E}_d = \dot{m}T_0(s_4 - s_3)$$

$$\dot{E}_d = (4.449 \times 10^5 \text{ kg/h})(295 \text{ K})(0.5957 - 0.5926)(\text{kJ/kg} \cdot \text{K})$$

$$= \left(4.07 \times 10^5 \frac{\text{kJ}}{\text{h}}\right)\left|\frac{1 \text{ h}}{3600 \text{ s}}\right|\left|\frac{1 \text{ MW}}{10^3 \text{ kJ/s}}\right| = 0.11 \text{ MW}$$

(0.11/231.28)(69%) = **0.03**%.

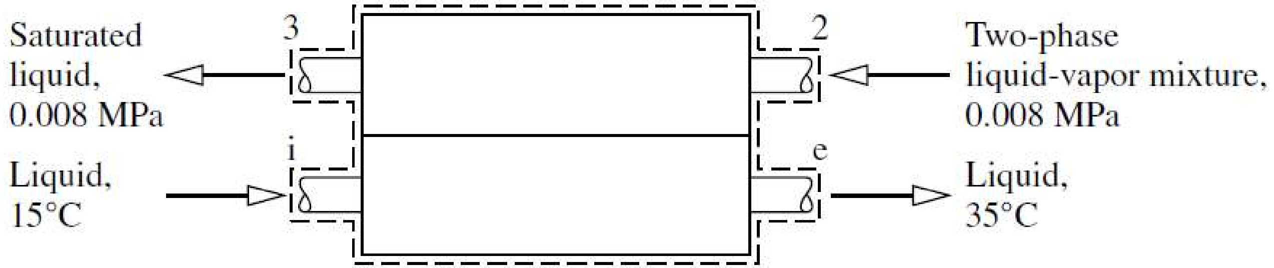
The net power output of the vapor power plant of Example 2 is 100 MW. Expressing this as a percentage of the rate at which exergy is carried into the plant with the fuel, (100/231.28)(69%) = **30**%

EXAMPLE 5 CONDENSER EXERGY ANALYSIS

The condenser of Example 2 involves two separate water streams. In one stream a two-phase liquid–vapor mixture enters at 0.008 MPa and exits as a saturated liquid at 0.008 MPa. In the other stream, cooling water enters at 15C and exits at 35C.

(a) Determine the net rate at which exergy is carried from the condenser by the cooling water, in MW. Express this result as a percentage of the exergy entering the plant with the fuel.

(b) Determine for the condenser the rate of exergy destruction, in MW. Express this result as a percentage of the exergy entering the plant with the fuel.



(a) The net rate at which exergy is carried out of the condenser

$$\begin{bmatrix}\text{net rate at which exergy}\\ \text{is carried out by the}\\ \text{cooling water}\end{bmatrix} = \dot{m}_{cw}(e_{fe} - e_{fi})$$

$$= \dot{m}_{cw}[h_e - h_i - T_0(s_e - s_i)]$$

$$\dot{m}_{cw}(e_{fe} - e_{fi}) = (9.39 \times 10^6\ \text{kg/h})[(146.68 - 62.99)\ \text{kJ/kg} - (295\ \text{K})(0.5053 - 0.2245)\ \text{kJ/kg} \cdot \text{K}]$$

$$= \frac{8.019 \times 10^6\ \text{kJ/h}}{|3600\ \text{s/h}|}\left|\frac{1\ \text{MW}}{10^3\ \text{kJ/s}}\right| = 2.23\ \text{MW}$$

As a percentage of the exergy entering the plant with the fuel; (2.23/231.28)(69%) = **1**%.

(b) The rate of exergy destruction for the condenser can be evaluated by reducing the exergy rate balance. Alternatively, the relationship $\dot{E}_d = T_o \dot{\sigma}_{cv}$ can be employed, where $\dot{\sigma}_{cv}$ is the time rate of entropy production for the condenser determined from
an entropy rate balance. With either approach, the rate of exergy destruction can be expressed as

$$\dot{E}_d = T_0\left[\dot{m}(s_3 - s_2) + \dot{m}_{cw}(s_e - s_i)\right]$$

$$\dot{E}_d = 295\left[(4.449 \times 10^5)(0.5926 - 6.2021) + (9.39 \times 10^6)(0.5053 - 0.2245)\right]$$

$$= \frac{416.1 \times 10^5 \text{ kJ/h}}{|3600 \text{ s/h}|}\left|\frac{1 \text{ MW}}{10^3 \text{ kJ/s}}\right| = 11.56 \text{ MW}$$

Expressing this as a percentage of the exergy entering the plant with the fuel, (11.56/231.28)(69%) = **3** %

Vapor Power Plant Exergy Accounting

Outputs			
	Net power out		30%
	Losses		
		Condenser cooling water	1%
		Stack gases (assumed)	1%
Exergy Destruction			
	Boiler		
		Combustion unit (assumed)	30%
		Heat exchanger unit	30%
		Turbine	5%
		Pump	—
	Condenser		3%
Total			**100%**