Chapter 10

VAPOR & COMBINED POWER CYCLES

Thermodynamics: An Engineering Approach
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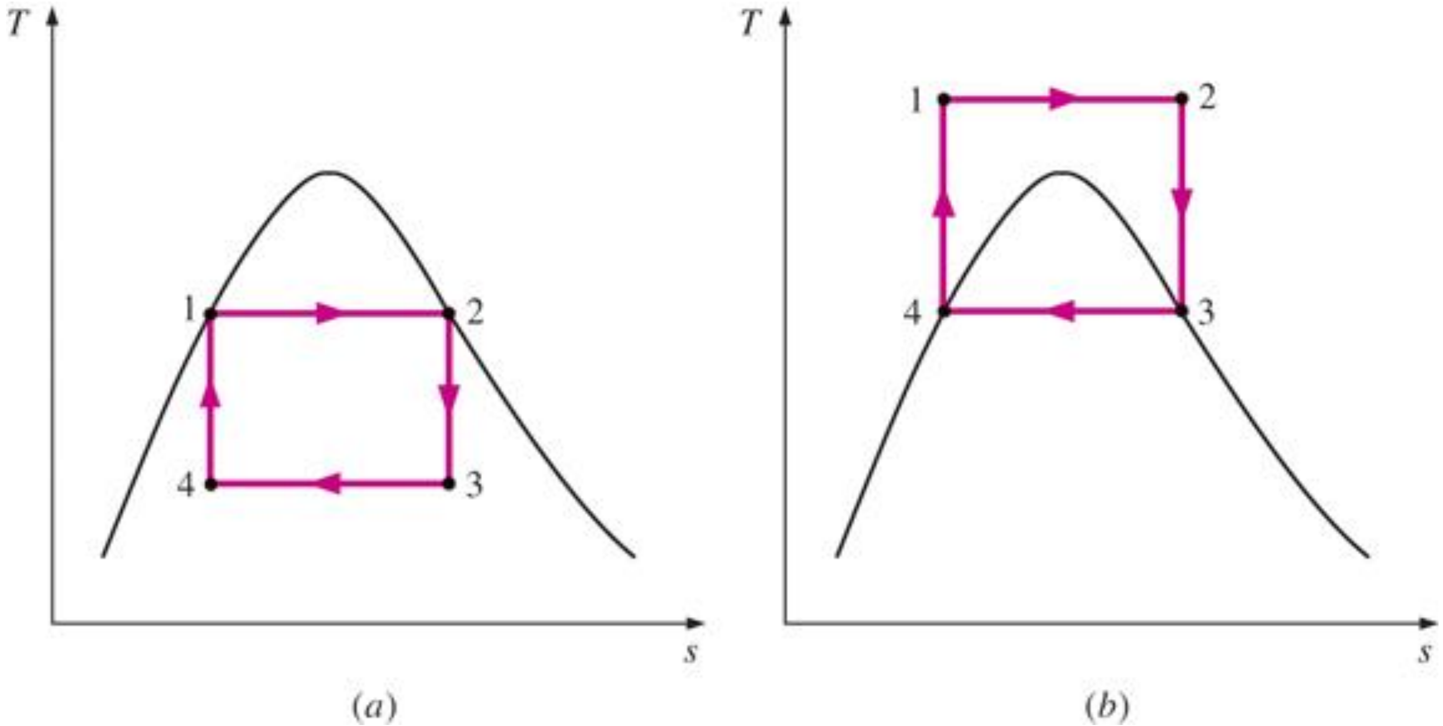
THE CARNOT VAPOR CYCLE



***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

THE CARNOT VAPOR CYCLE

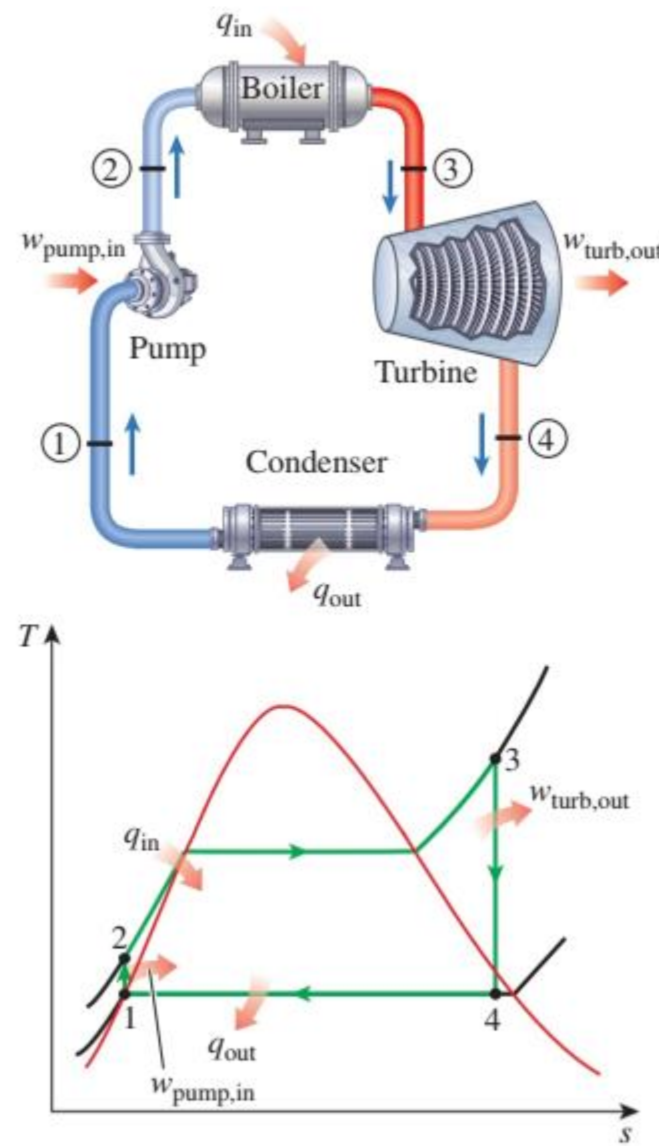
- The **Carnot cycle** is the most efficient cycle operating between two specified temperature limits, but it is *not a suitable model* for power cycles. Because:
  - **Process 1-2** Limiting the heat transfer processes to two-phase systems severely limits the maximum temperature that can be used in 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.

RANKINE CYCLE:

THE IDEAL CYCLE FOR VAPOR POWER CYCLES

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

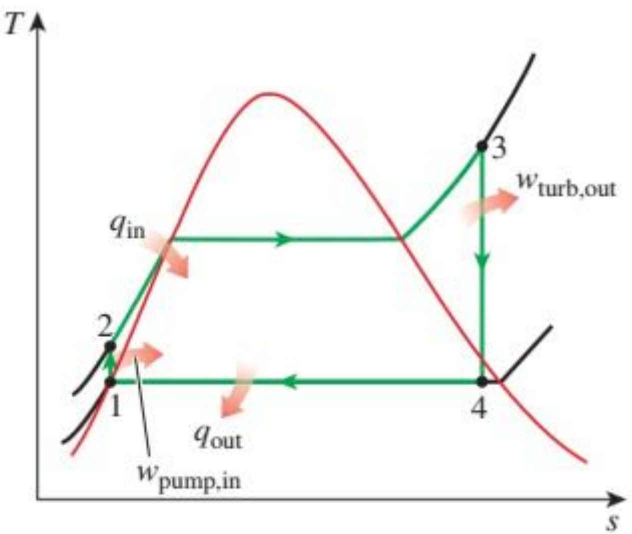
1-2	Isentropic compression in a pump
2-3	Constant pressure heat addition in a boiler
3-4	Isentropic expansion in a turbine
4-1	Constant pressure heat rejection in a condenser



The simple ideal **Rankine cycle**.

Energy Analysis of the Ideal Rankine Cycle

Steady-flow energy equation



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

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

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

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

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

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

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

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

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

The thermal efficiency can be interpreted as the ratio of the area enclosed by the cycle on a *T-s* diagram to the area under the heat-addition process.

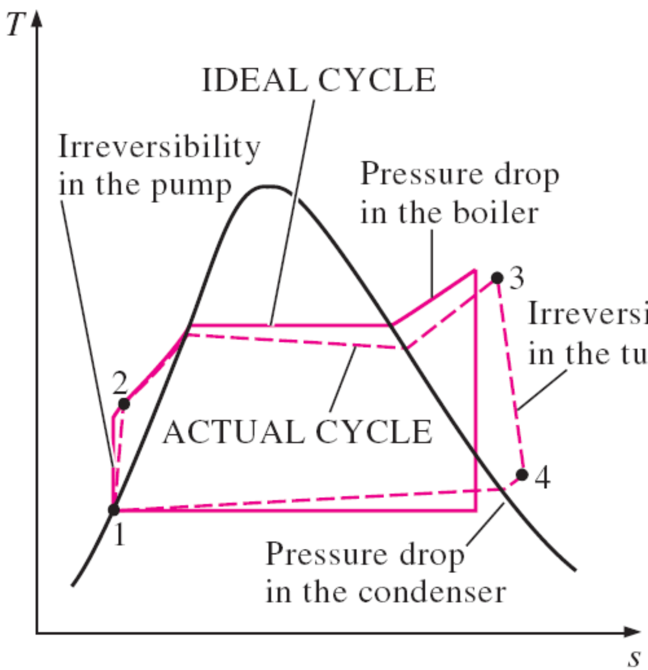
DEVIATION OF ACTUAL VAPOR POWER CYCLES FROM IDEALIZED ONES

- 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.

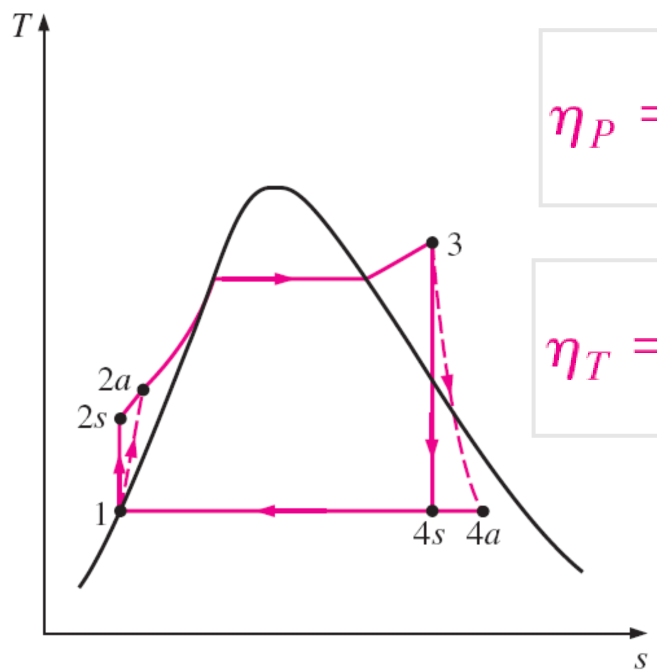
Isentropic efficiencies:

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

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



(a)



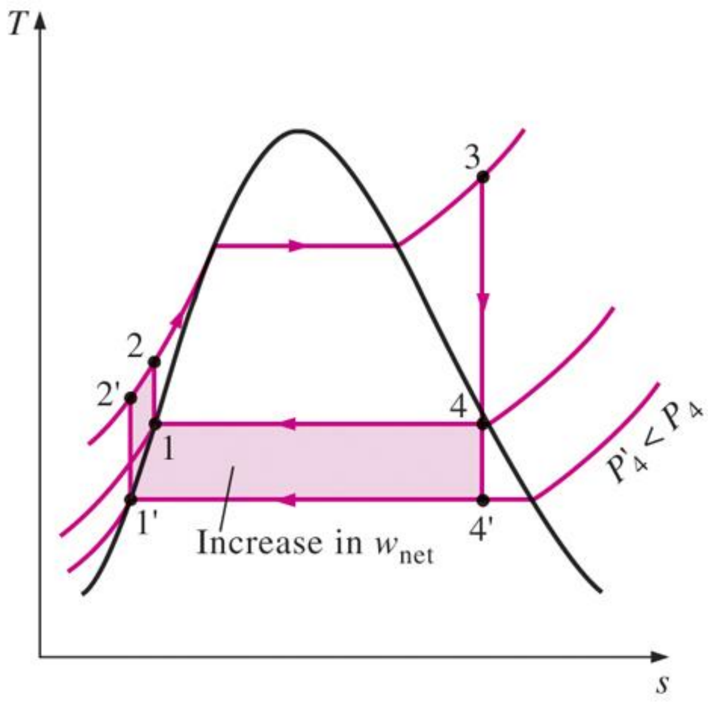
(b)

(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.**

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.*

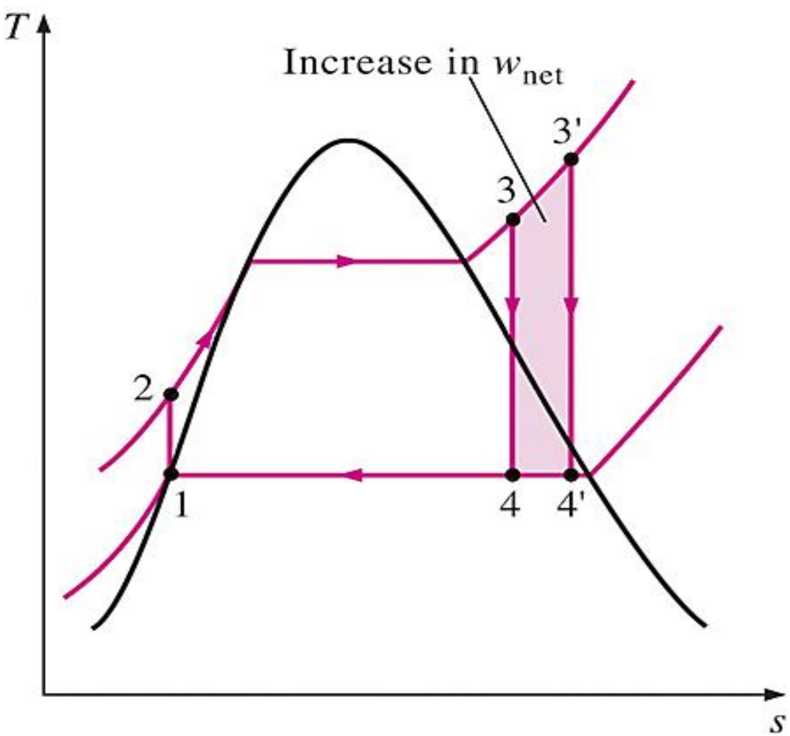
Lowering the Condenser Pressure (*Lowers $T_{low,avg}$*)



1) 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
2) **Side effect:** Lowering the condenser pressure increases the moisture content of the steam at the final stages of the turbine.

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

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

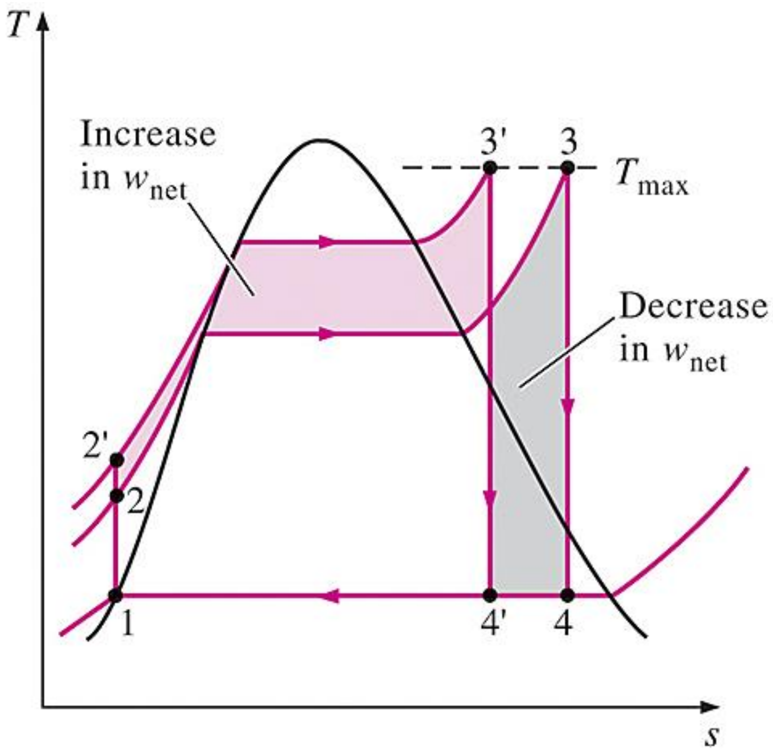


The effect of superheating the steam to higher temperatures on the ideal Rankine cycle.

- ❑ 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.

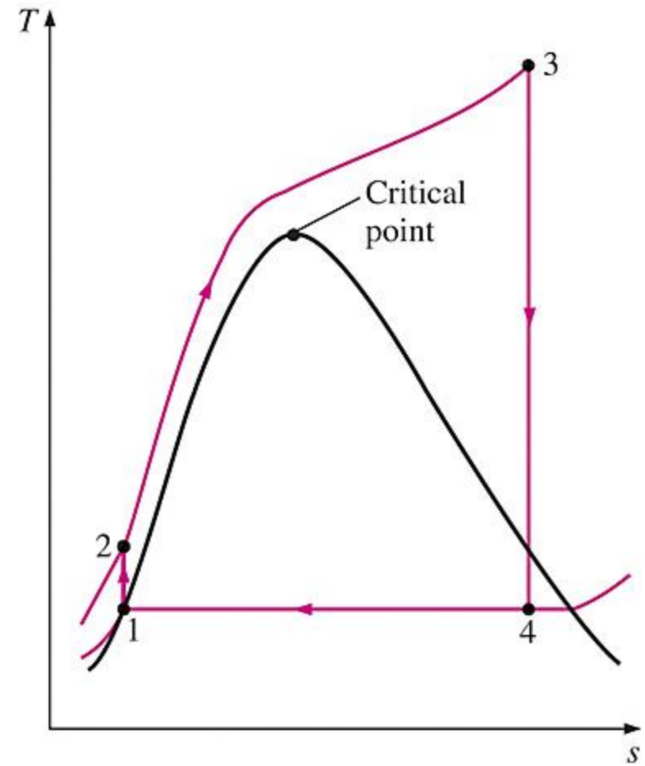
Increasing the Boiler Pressure (*Increases $T_{high,avg}$*)

- 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.

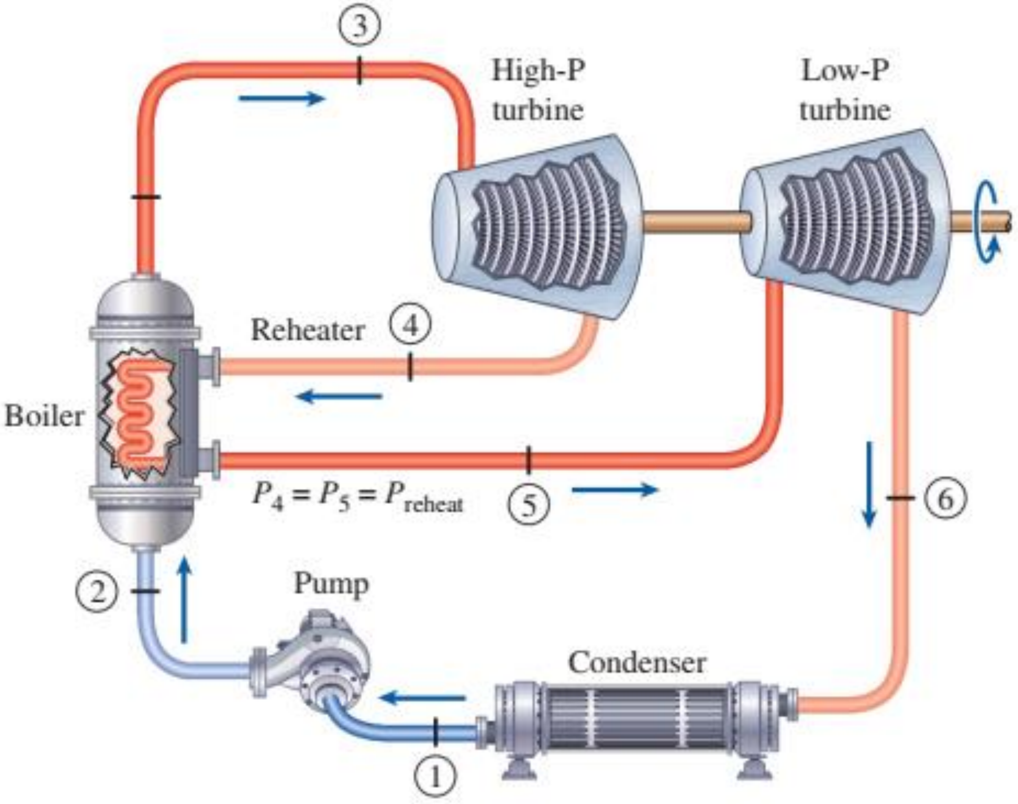
THE IDEAL REHEAT RANKINE CYCLE

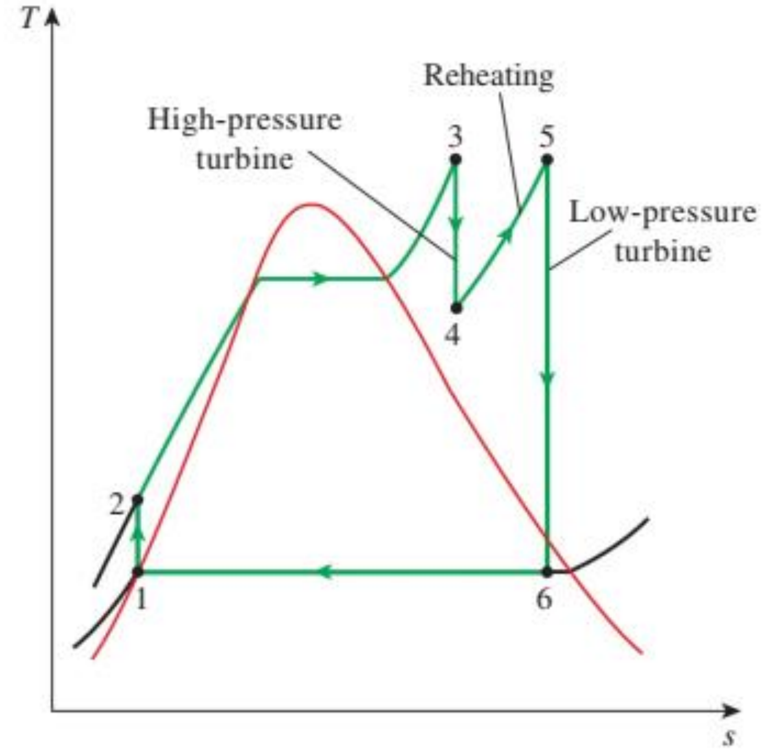
- How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?
  1. Superheat the steam to very high temperatures. It is limited metallurgically.
  2. Expand the steam in the turbine in two stages and reheat it in between (**reheat**).

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

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

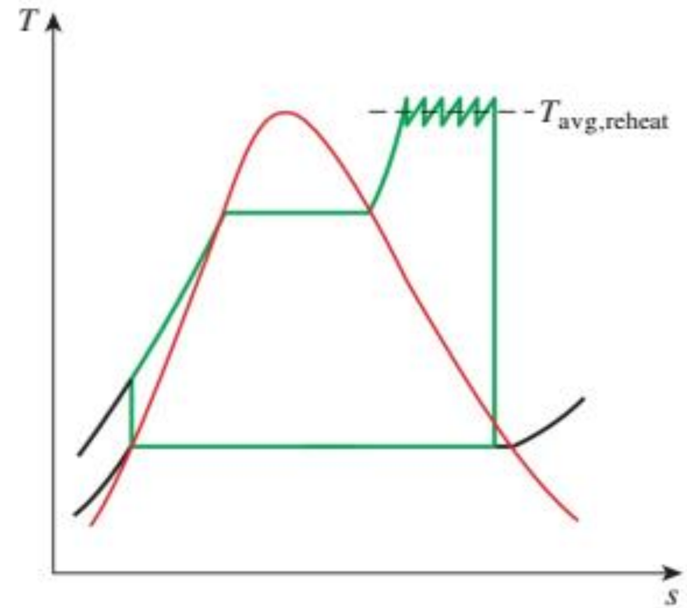
THE IDEAL REHEAT RANKINE CYCLE





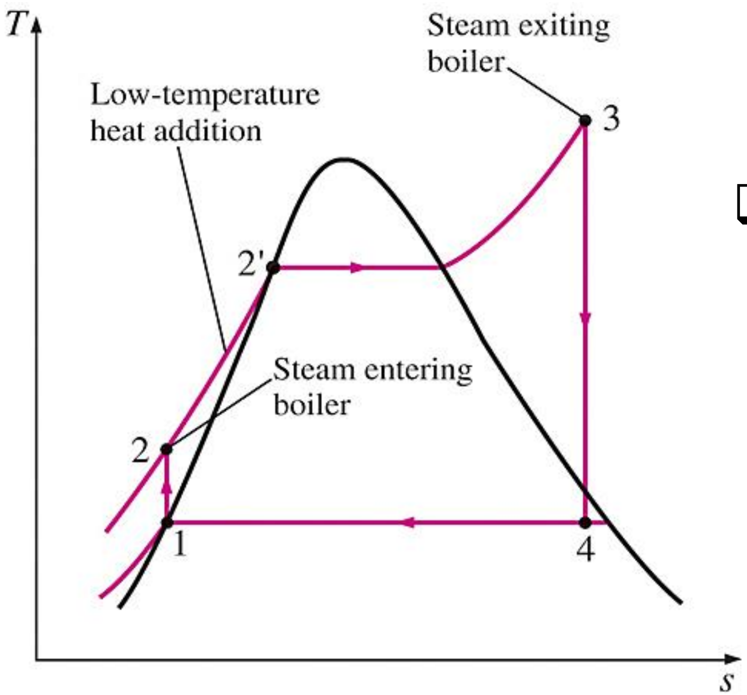
THE IDEAL REHEAT RANKINE CYCLE

- 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.

THE IDEAL REGENERATIVE RANKINE CYCLE

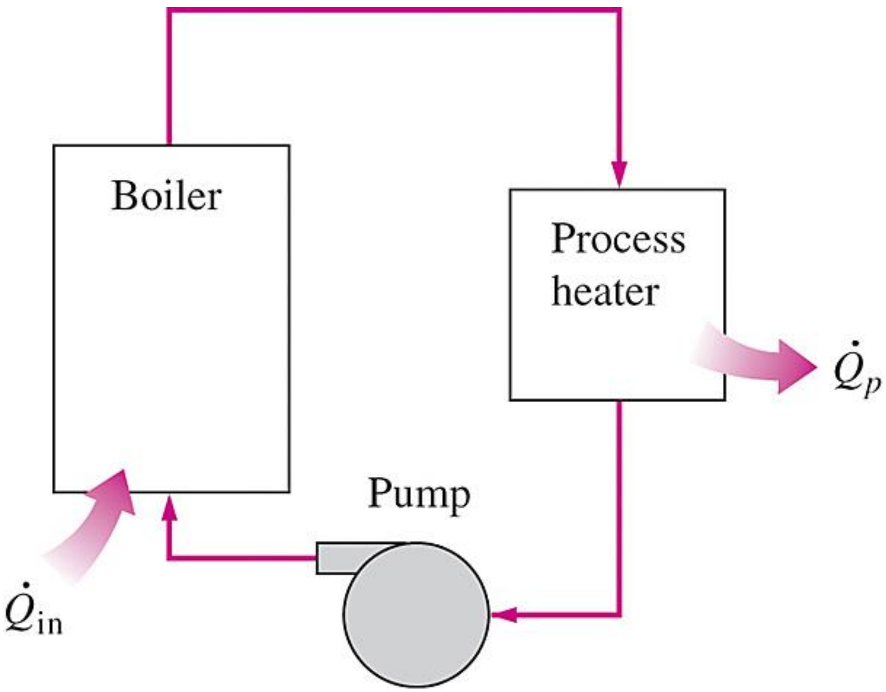


The first part of the heat-addition process in the boiler takes place at relatively low temperatures.

- 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).

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.

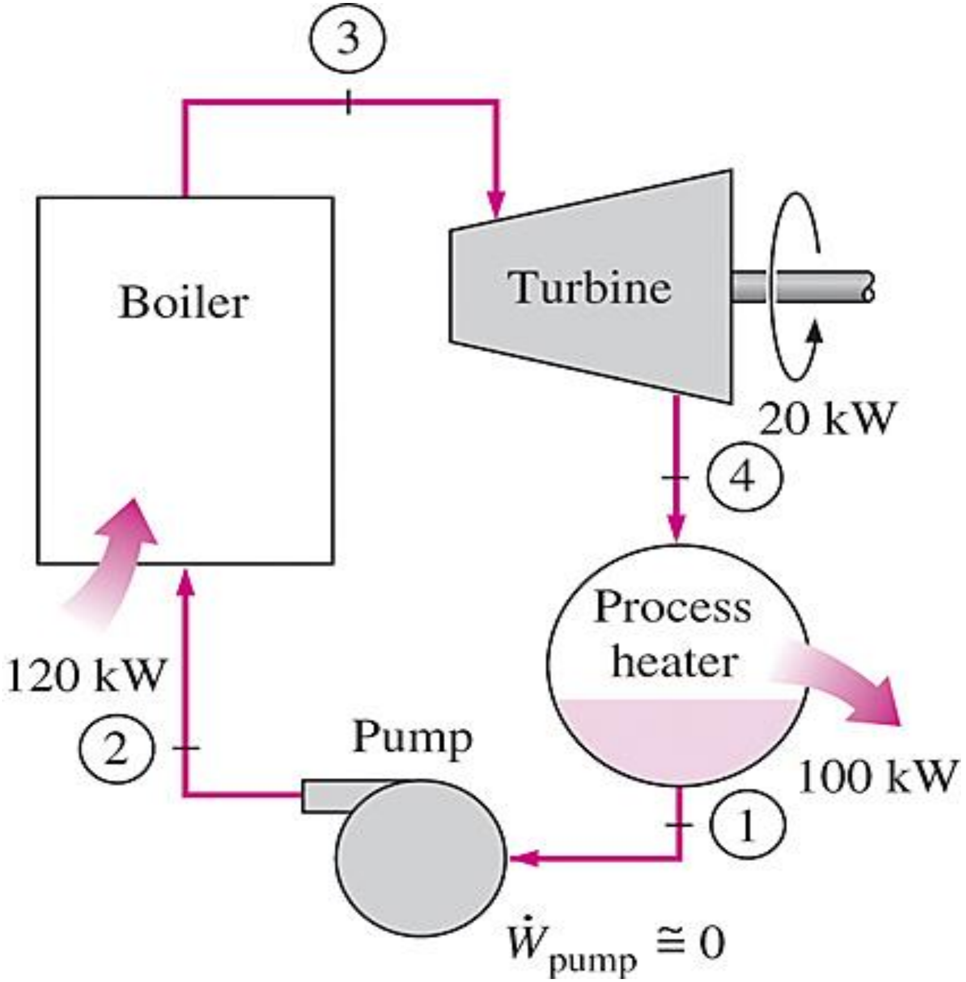
- Industries that use large amounts of process heat also consume a large amount of electric power.
- It makes sense to use the already-existing work potential to produce power instead of letting it go to waste.
- The result is a plant that produces electricity while meeting the process-heat requirements 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:

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

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

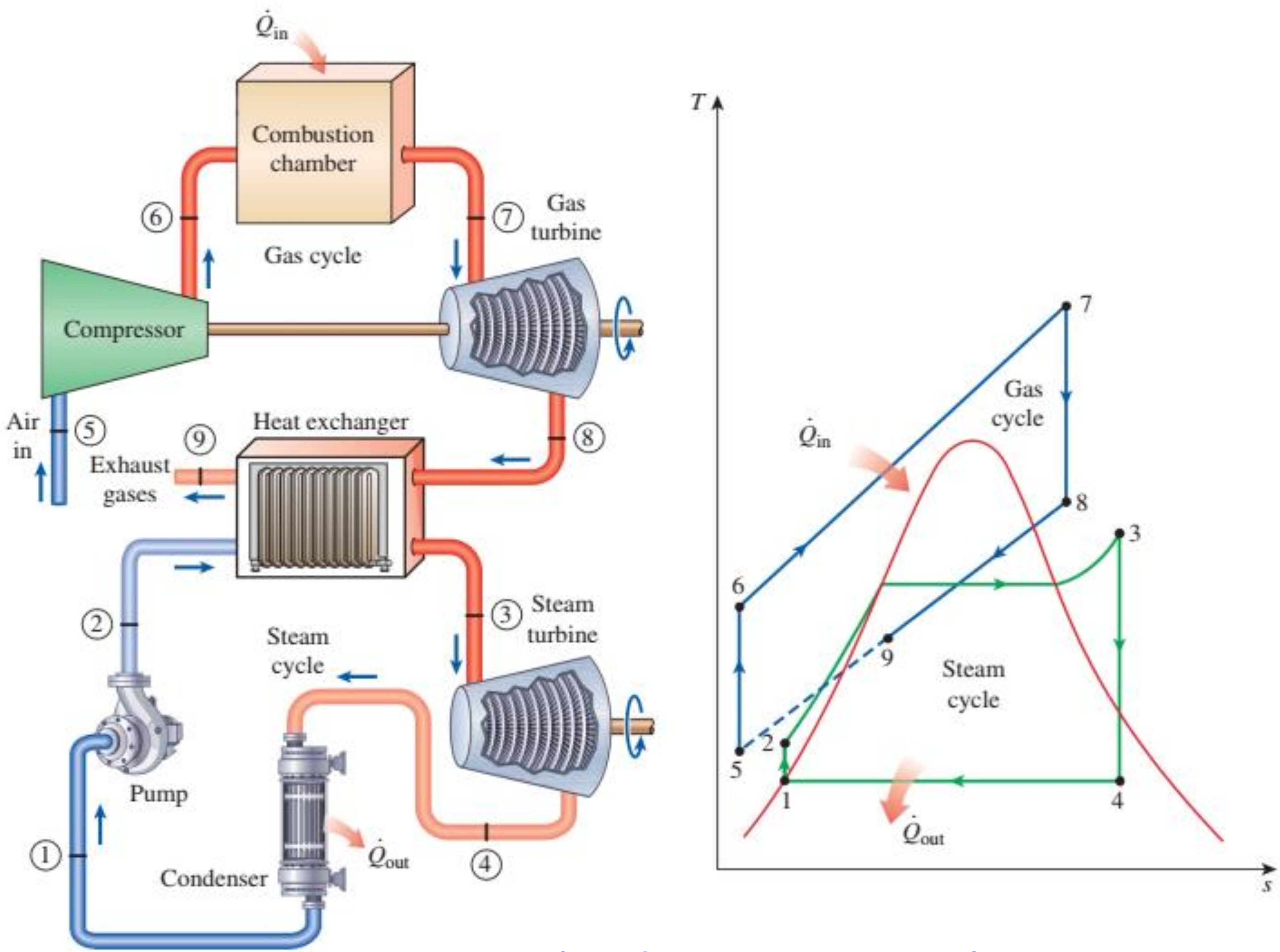


An ideal cogeneration plant.

- 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.

COMBINED GAS–VAPOR POWER CYCLES

1) The continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants.
2) 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**.
3) 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.
4) 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.
5) Recent developments in gas-turbine technology have made the combined gas–steam cycle economically very attractive.
6) 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.
7) Thermal efficiencies over 50% are reported.



Combined gas–steam power plant.

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.

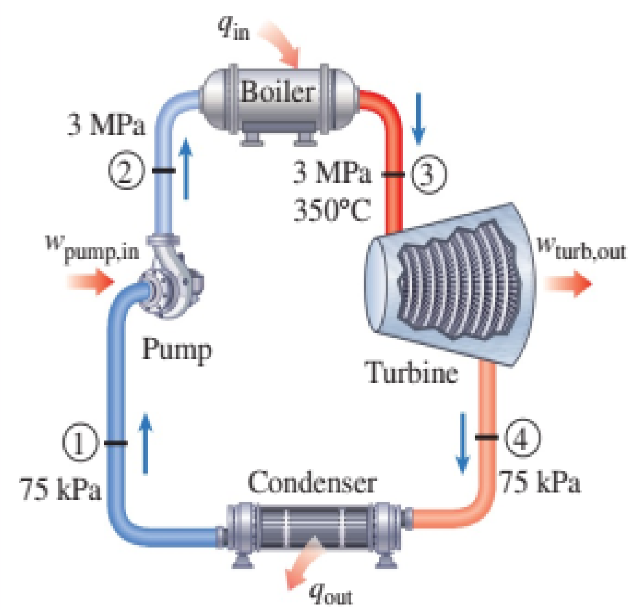
SOLUTION A steam power plant operating on the simple ideal Rankine cycle is considered. The thermal efficiency of the cycle is to be determined.
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.
Analysis The schematic of the power plant and 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.

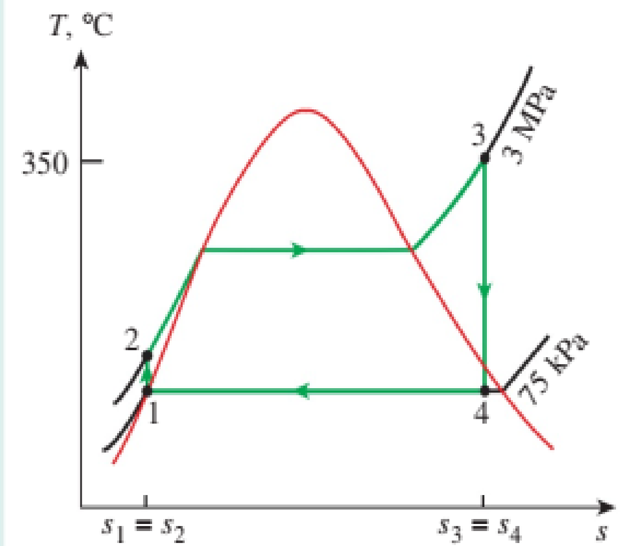
First we determine the enthalpies at various points in the cycle, using data from steam tables (Tables A–4, A–5, and A–6):

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

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

$s_2 = s_1$





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

$$= 3.03\ \text{kJ/kg}$$

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

State 3:

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

State 4:

$$P_4 = 75\ \text{kPa} \quad \text{(sat. mixture)}$$

$$s_4 = s_3$$

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

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

Thus,

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

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

and

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

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

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

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

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

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 combustion gases in the furnace.

EXAMPLE 10–2 An Actual Steam Power Cycle

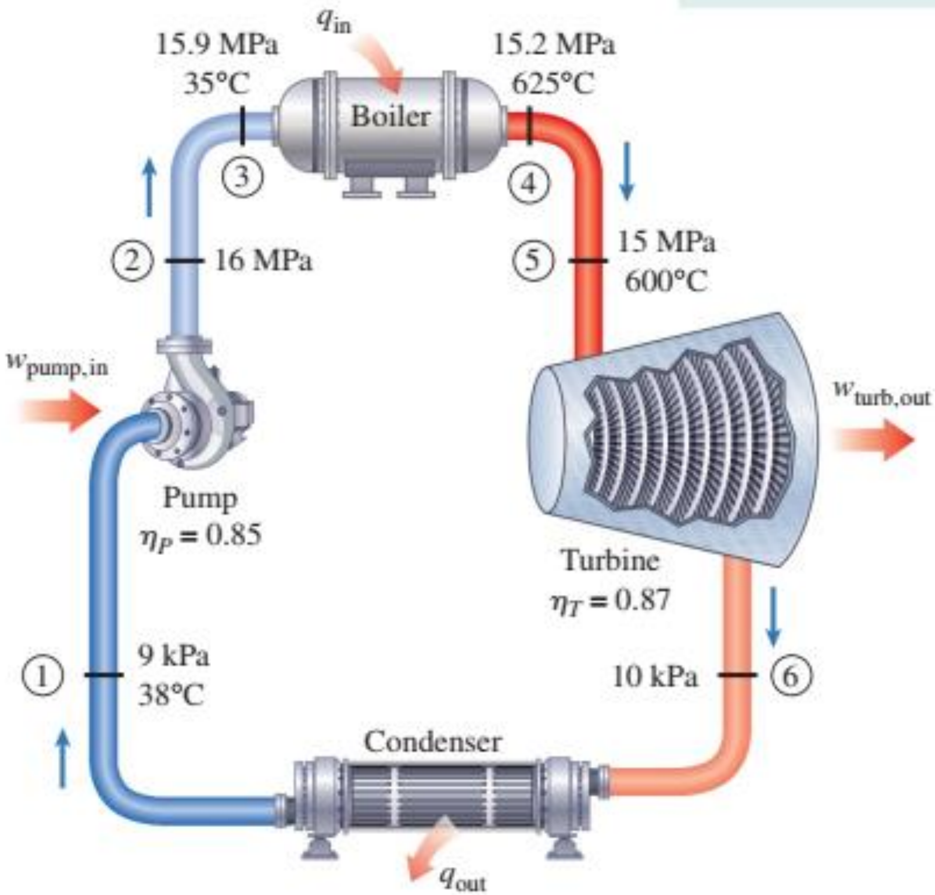
A steam power plant operates on the cycle shown in Fig. 10–5. If the isentropic efficiency of the turbine is 87 percent and the isentropic efficiency of the pump is 85 percent, determine (*a*) the thermal efficiency of the cycle and (*b*) the net power output of the plant for a mass flow rate of 15 kg/s.

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:



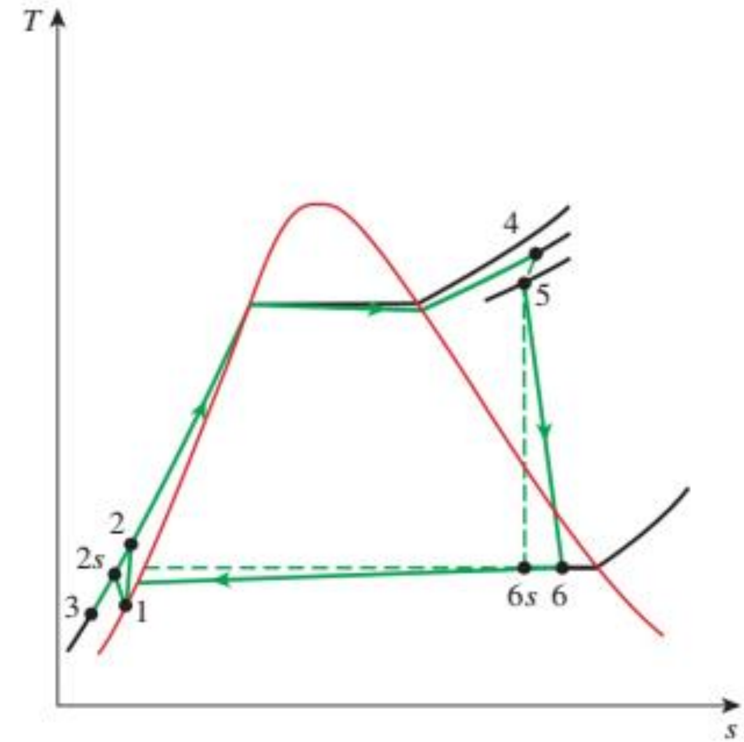


FIGURE 10–5
Schematic and *T*-*s* diagram for Example 10–2.

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

Thus,

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

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

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

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

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

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: $P_2 = 3\text{ MPa}$

$s_2 = s_1$

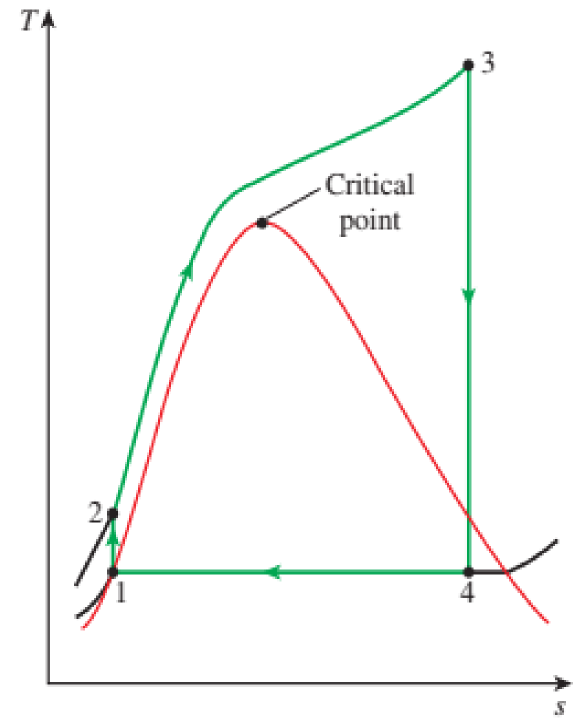


FIGURE 10–9
A supercritical Rankine cycle.

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

$$= 3.02\ kJ/kg$$

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

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

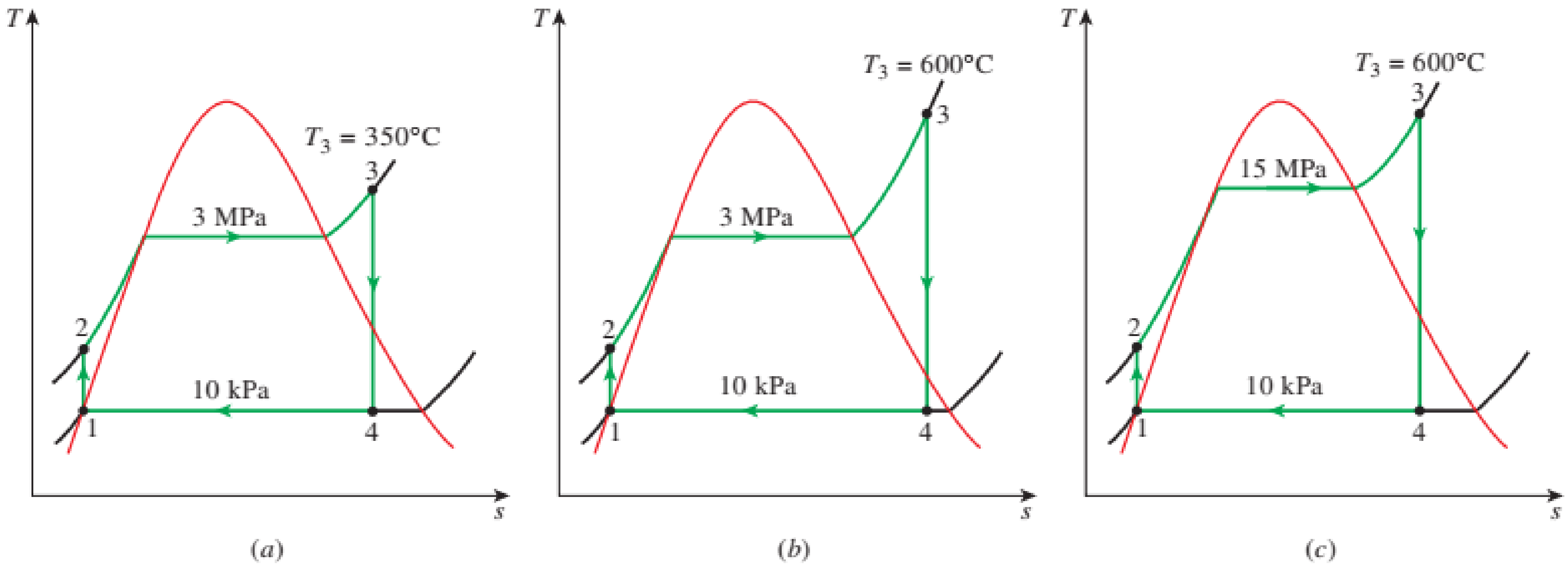


FIGURE 10–10
T-s diagrams of the three cycles discussed in Example 10–3.

State 4: $P_4 = 10 \text{ kPa}$ (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

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

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

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

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

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

Thus,

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

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

and

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

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

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

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

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

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

Thus,

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

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

and

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

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).