ESO201A : THERMODYNAMICS
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IIT Kanpur

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Lecture 29

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 $T_{low,avg}$*)

It leads to increase in thermal efficiency

However, it also lead to higher moisture content in the final stages (near exit) of turbine, which is undesirable.

For example, if river water at 15 ^{0}C is available, the temperature in the condenser needs to be At least 25 ^{0}C and hence the condenser pressure will be equal to corresponding saturation pressure At 25 ^{0}C which is 3.2 kPa

Increase in w_{net}

$P'_4 < P_4$

Lowering condenser pressures leads to air leakage
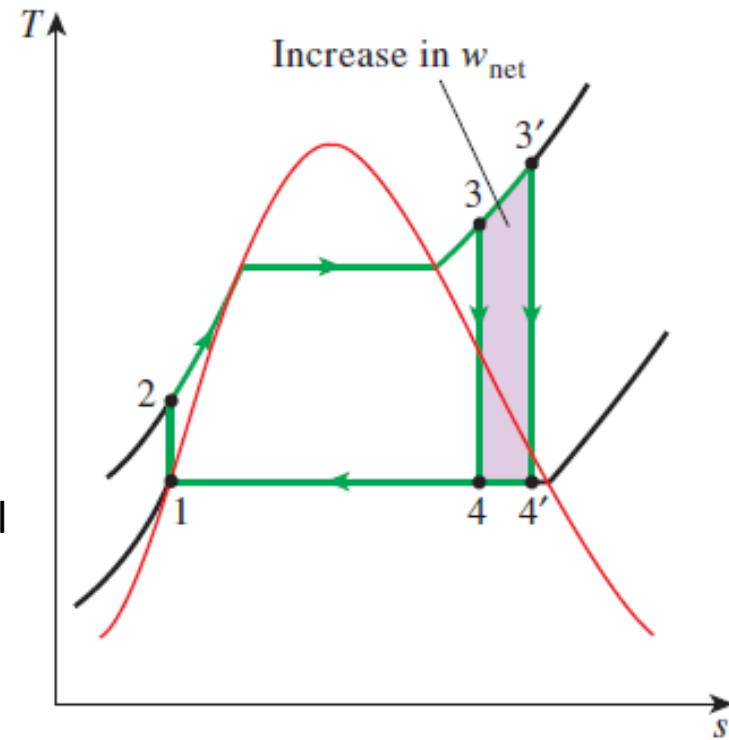
Ref. Cengel and Boles, 8th Edition (2015)
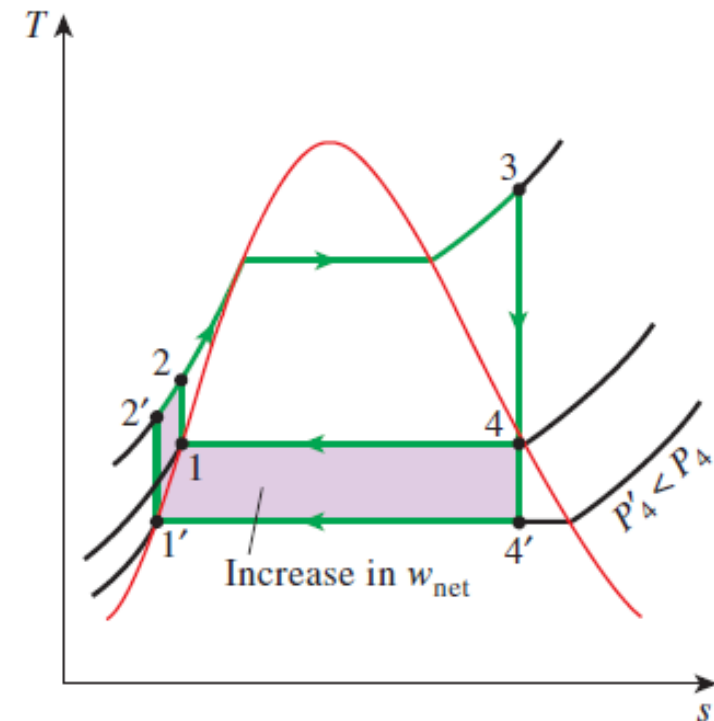
Superheating the Steam to High Temperatures (*Increases* $T_{high,avg}$)

It leads to increase in thermal efficiency

Further, it leads to reduction in moisture at turbine exit

However, maximum temperature is limited by metallurgical considerations. It cannot be more than 620 ^{0}C.

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

It leads to increase in thermal efficiency

In early days, typical boiler pressures were around 3 MPa. In modern times, boilers with pressures >30 MPa (supercritical cycles) are also used leading to thermal efficiencies as high as 40 % for fossil fuel plants.

However, increasing boiler pressure leads to an undesirable increase in moisture content at turbine exit

Ref. Cengel and Boles, 8th Edition (2015)
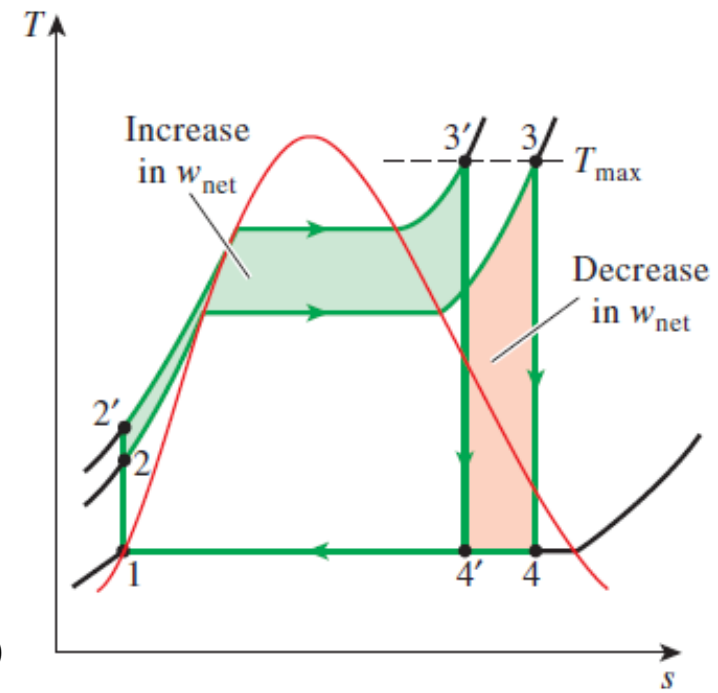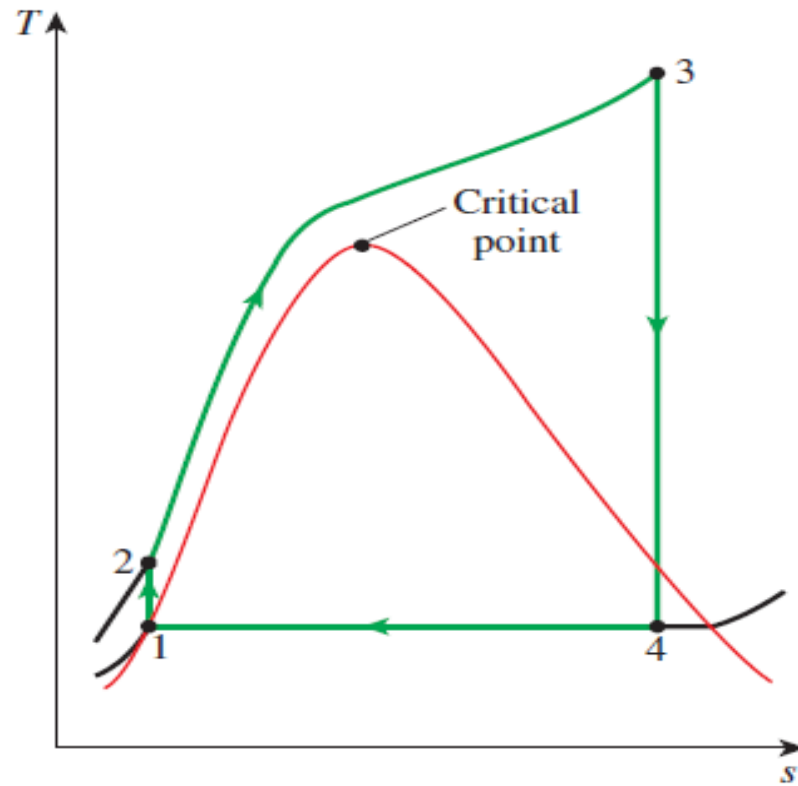
A supercritical Rankine cycle.

Ref. Cengel and Boles, 8th Edition (2015)

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.

Ref. Cengel and Boles, 8th Edition (2015)
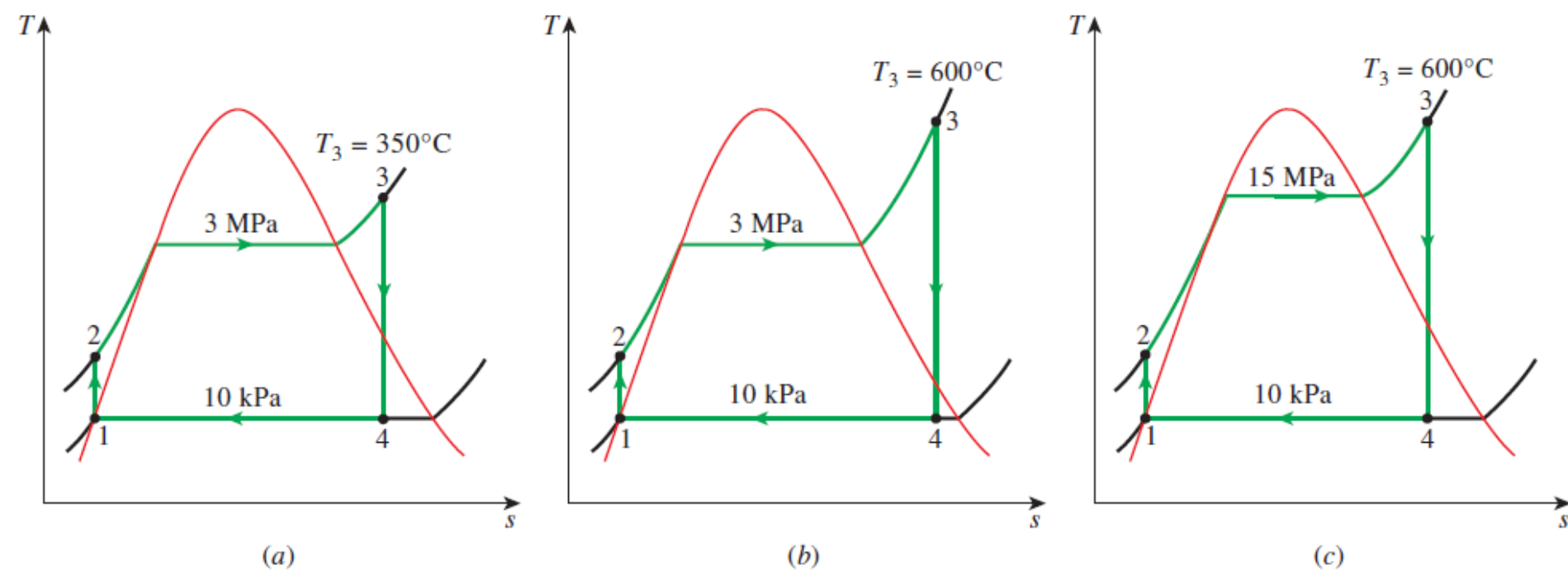
(a) This is the steam power plant except that the condenser pressure is lowered to 10 kPa. The thermal efficiency is determined in a similar manner:

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

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

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

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

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

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

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

            $s_4 = s_3$

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

Ref. Cengel and Boles, 8th Edition (2015)

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

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

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

Ref. Cengel and Boles, 8th Edition (2015)

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

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

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

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

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

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

Ref. Cengel and Boles, 8th Edition (2015)

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

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

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).

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?

Ref. Cengel and Boles, 8th Edition (2015)

Two possibilities come to mind:

1. Superheat the steam to very high temperatures before it enters the turbine. This would be the desirable solution since the average temperature at which heat is added would also increase, thus increasing the cycle efficiency. This is not a viable solution, however, since it requires raising the steam temperature to metallurgically unsafe levels.

2. Expand the steam in the turbine in two stages, and reheat it in between. In other words, modify the simple ideal Rankine cycle with a **reheat** process. Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants.
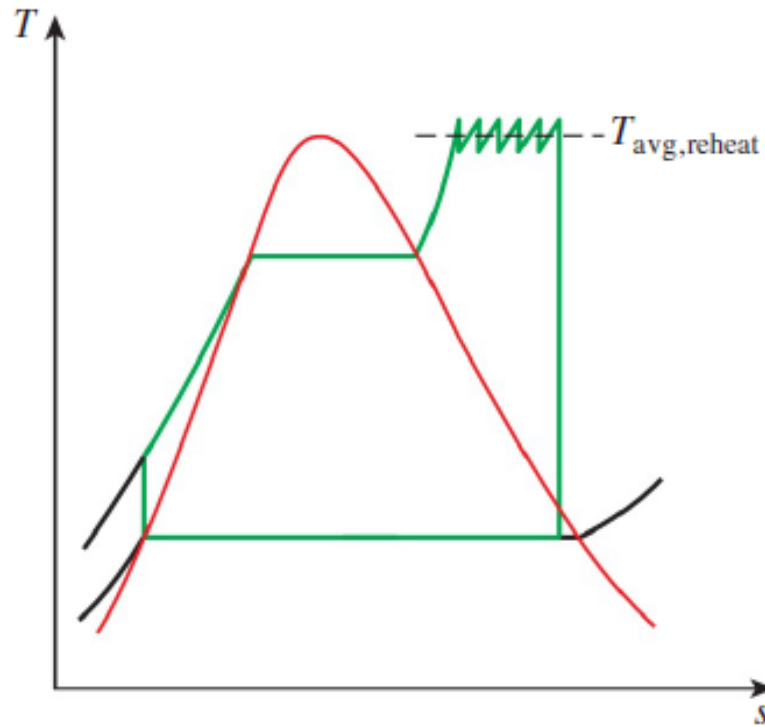
The *T-s* diagram of the ideal reheat Rankine cycle and the schematic of the power plant operating on this cycle are shown in Fig. _____ . The ideal reheat Rankine cycle differs from the simple ideal Rankine cycle in that the expansion process takes place in two stages. In the first stage (the high-pressure turbine), steam is expanded isentropically to an intermediate pressure and sent back to the boiler where it is reheated at constant pressure, usually to the inlet temperature of the first turbine stage. Steam then expands isentropically in the second stage (low-pressure turbine) to the condenser pressure.

Ref. Cengel and Boles, 8th Edition (2015)
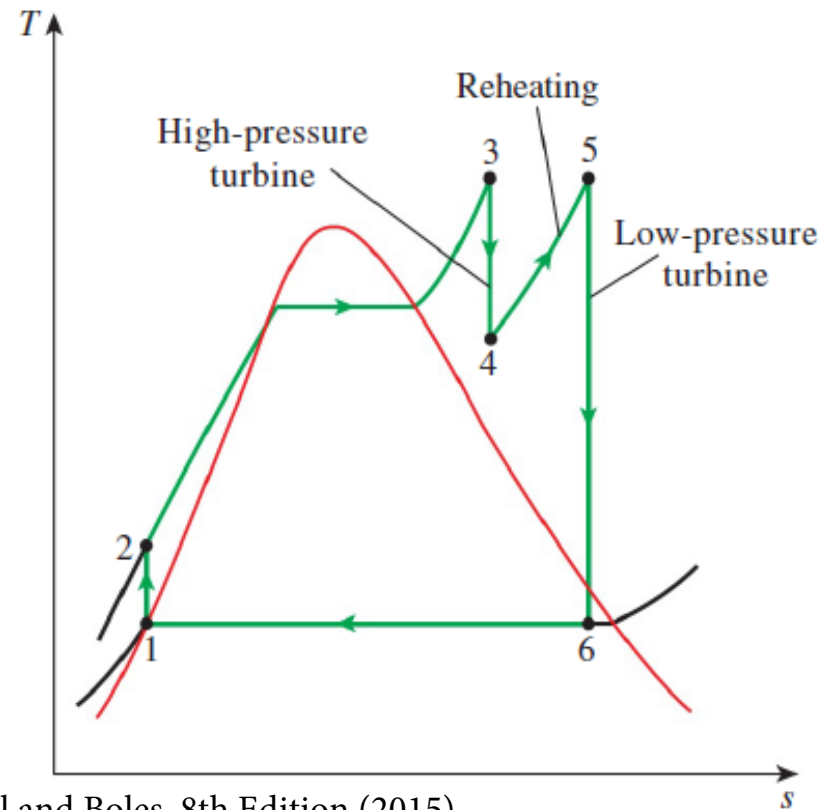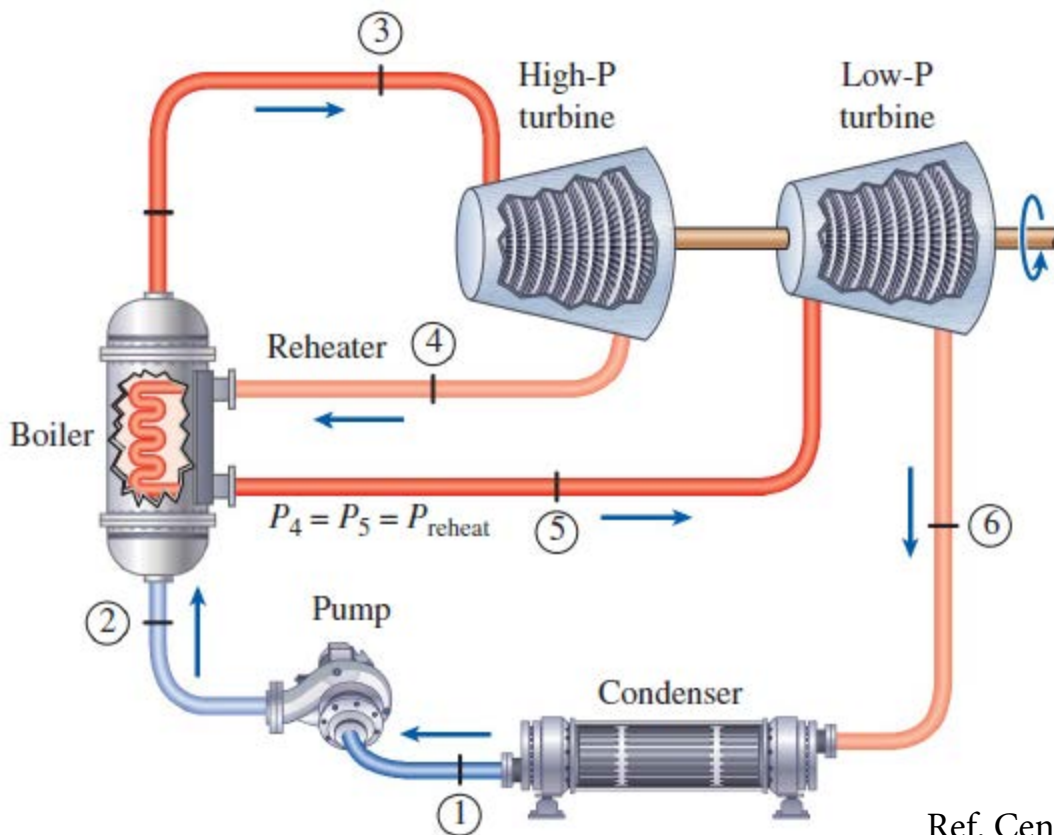
$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4)$$

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

The incorporation of the single reheat in a modern power plant improves the cycle efficiency by 4 to 5 percent 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, as seen in Fig. on next slide. The use of more than two reheat stages, however, is not practical. The theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat. If the turbine inlet pressure is not high enough, double reheat would result in superheated exhaust. This is undesirable as it would cause the average temperature for heat rejection to increase and thus the cycle efficiency to decrease. Therefore, double reheat is used only on supercritical-pressure ($P > 22.06$ MPa) power plants. A third reheat stage would increase the cycle efficiency by about half of the improvement attained by the second reheat. This gain is too small to justify the added cost and complexity.

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

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.

Ref. Cengel and Boles, 8th Edition (2015)
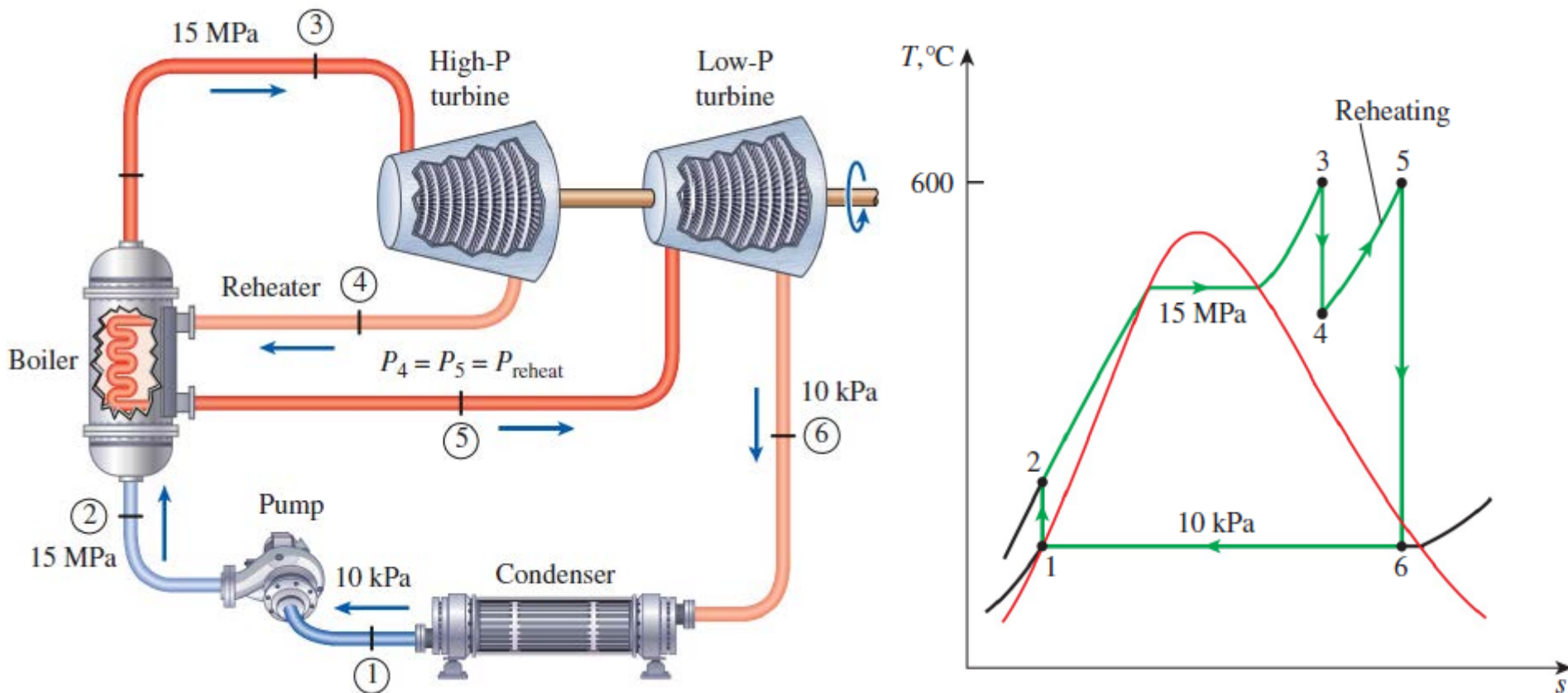
(a) The reheat pressure is determined from the requirement that the entropies at states 5 and 6 be the same:

State 6: $P_6 = 10 \text{ kPa}$

$x_6 = 0.896$ (sat. mixture)

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

Also,

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

Thus,

State 5: $T_5 = 600°\text{C}$ $P_5 = 4.0 \text{ MPa}$
            $s_5 = s_6$ $h_5 = 3674.9 \text{ kJ/kg}$

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

Ref. Cengel and Boles, 8th Edition (2015)

State 2: $P_2 = 15 \text{ MPa}$

$s_2 = s_1$

$w_{pump,in} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg})$

$$\times [(15,000 - 10)\text{kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa·m}^3}\right)$$

$$= 15.14 \text{ kJ/kg}$$

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

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

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

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

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

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

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

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

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

Discussion This problem was solved in part (c) of the previous problem 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.

Ref. Cengel and Boles, 8th Edition (2015)