

CHAPTER 4

Vapor Power Cycles

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INTRODUCTION

ENERGY AND POWER

ENERGY:-

- Energy appears in many forms, but has one thing in common **energy is possessed of the ability to produce a dynamic, vital effect.**
- Energy is associated with physical substance, but is not a substance it self.
- It shows itself by excited, animated state assumed by material which receives energy.
- Energy exists in various forms:- ***mechanical, thermal, electrical.....***
- One form of energy can be converted into other by the use of suitable arrangements.

- Out of all these forms of energy, electrical energy is preferred due to the following advantages:-
 - ✓ **Can be easily transported from one place to another.**
 - ✓ **Losses in transport are minimum.**
 - ✓ **Can be easily subdivide.**
 - ✓ **Economical in use.**
 - ✓ **Easily converted to other form of energy.**
- **Power**.-any physical unit of energy when divided by a unit time automatically becomes a unit of power.
 - ✓ **The rate of production or consumption of heat energy and to a certain extent of radiation energy is not ordinary through of as power.**
 - ✓ **Power is primarily associated with mechanical and electrical energy.**
 - ✓ **Power can be define as the rate of flow of energy.**

- **Power plant** is a unit for the production and delivery of a flow of mechanical and electrical energy.
- In common usage, a machine or assemblage of equipment that produce and delivers a flow mechanical or electrical energy is a power plant.
- Example of power plants are: an internal combustion, water wheel.....

POWER PLANT CYCLES

- A thermal power station works on the basic principle that heat liberated by burning fuel is converted in to mechanical work by means of a suitable working fluid.

Fuel $\longrightarrow$ heat generated $\longrightarrow$ mechanical work $\longrightarrow$ electric energy.

- A working fluid goes through a repetitive cycle change and this cycle change involving heat and work is known as **thermodynamic cycle**.

CLASSIFICATION OF POWER PLANT CYCLES

➤ Thermal power plants in general may work on:-

- ✓ **vapor power cycles**
- ✓ **gas power cycles.**

❖ Vapor power cycles:-

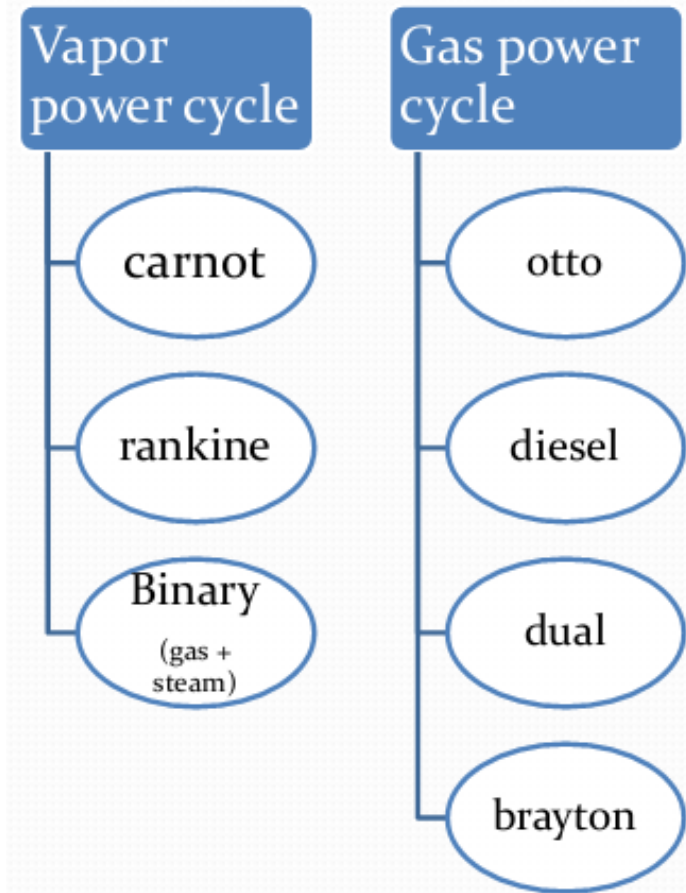
- ✓ **Rankine cycle,**
- ✓ **Reheat cycle,**
- ✓ **Regenerative cycle,**
- ✓ **Binary vapor cycle.....**

❖ Gas power cycles:-

- ✓ **Otto cycles,**
- ✓ **Duel combustion cycle,**
- ✓ **Diesel cycles,**
- ✓ **Gas turbine cycles.....**

❖ **Nuclear energy** has enlarged the world's power resources,

- ✓ the energy released by 1kg of uranium $\approx$ 4500 kg of high grade coal.



SOURCE OF ENERGY

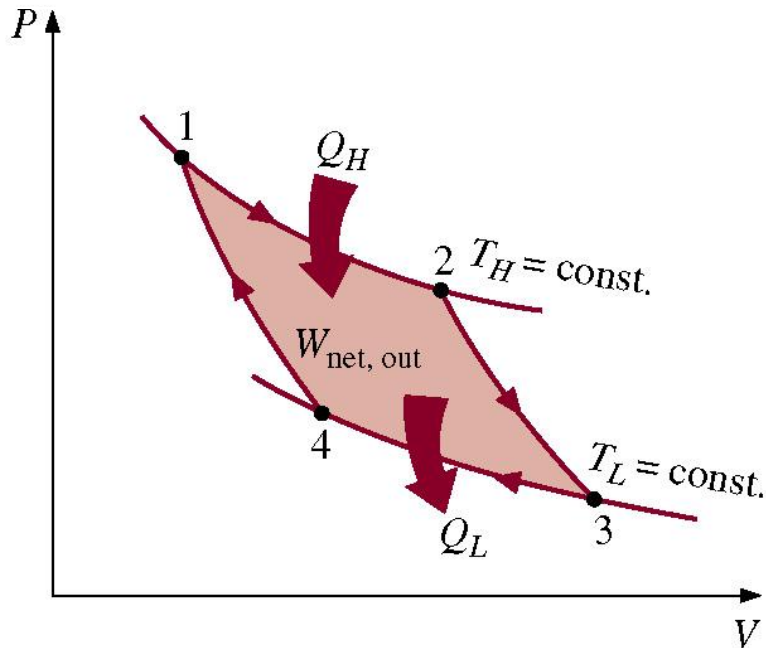
- The various source of energy are:-
- ✓ **Fuels:-** may be chemical or nuclear. **consider chemical fuel only.**
- A chemical fuel is a substance which release heat energy on combustion.
- The principal combustible elements of each fuel are *carbon* and *hydrogen*.

CLASSIFICATION OF FUELS

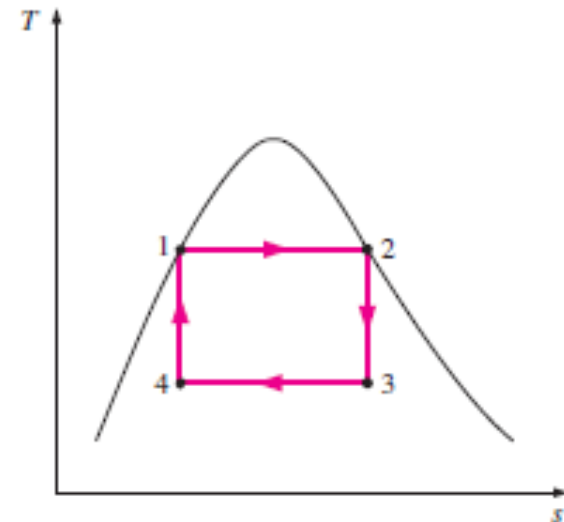
- fuels can be classified according to whether:
- ❖ They occur in *nature* (primary fuel) and are *prepared* (secondary fuel).
- ❖ They are in solid, liquid or gaseous state.

1.1. The Carnot Cycle

- We have mentioned repeatedly that the Carnot cycle is the most efficient cycle operating between two specified temperature limits.
- Consider a steady-flow Carnot cycle executed within the saturation dome of a pure substance.
- The fluid is heated **reversibly** and **isothermally** in a boiler (**process 1-2**), expanded **isentropically** in a turbine (**process 2-3**), condensed **reversibly** and **isothermally** in a condenser (**process 3-4**), and compressed **isentropically** by a compressor to the initial state (**process 4-1**).



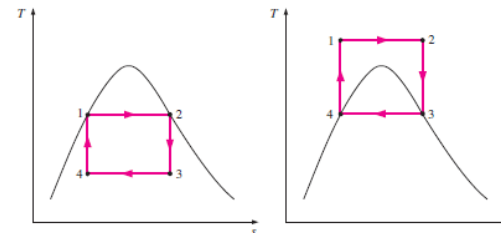
$$\eta_{\text{carnot}} = \frac{T_H - T_L}{T_H} = \left[1 - \frac{T_L}{T_H} \right]$$



8.1. The Carnot Cycle

- Several impracticalities are associated with this cycle:
 - Limiting the heat transfer processes to two-phase systems, severely limits the maximum temperature that can be used in the cycle - maximum temperature has to remain under the **critical-point value**.
 - The impingement of liquid droplets on the turbine blades **causes erosion** and is a major source of wear.
 - ✓ This problem could be eliminated by using a working fluid with a very steep saturated vapor line.
 - It is not easy to control the condensation process so precisely as to end up with desired quality at **state 4** and it is **not practical** to design a compressor that will handle two phases.
- Some of these problems could be eliminated by executing the Carnot cycle in a different way as shown in the second. This cycle however, presents other problems such as **isentropic compression to extremely high pressure** and **isothermal heat transfer at variable pressures**.

Conclusion: **Carnot cycle** cannot be approximated in actual devices and is not a realistic **model for vapor power cycles**.



8.2. The Rankine Cycle

- 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 as shown.
- 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** and consists of the following four processes:

- 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

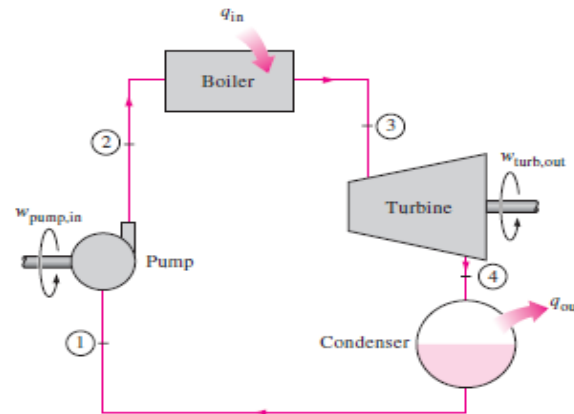
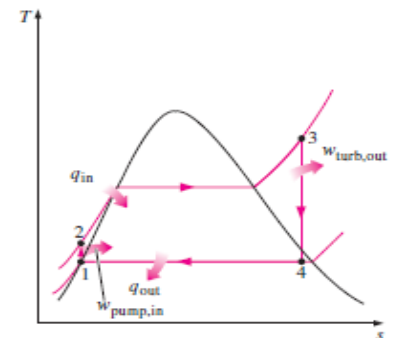


FIGURE
The simple ideal Rankine cycle.



Energy Analysis of the Rankine Cycle

- Rankine cycle can be analyzed as steady-flow processes.
- Neglecting changes in kinetic and potential energies.
- The *steady-flow energy equation* per unit mass of steam

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i$$

- The boiler and the condenser do not involve any work,
- The pump and the turbine are assumed to be isentropic.
- The conservation of energy relation for each device can be expressed as follows:

$$\begin{aligned} \text{Pump}(q = 0): \quad w_{pump,in} &= h_2 - h_1 \\ w_{pump,in} &= v(P_2 - P_1) \end{aligned}$$

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

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

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

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

The *thermal efficiency* of the **Rankine cycle** is

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

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

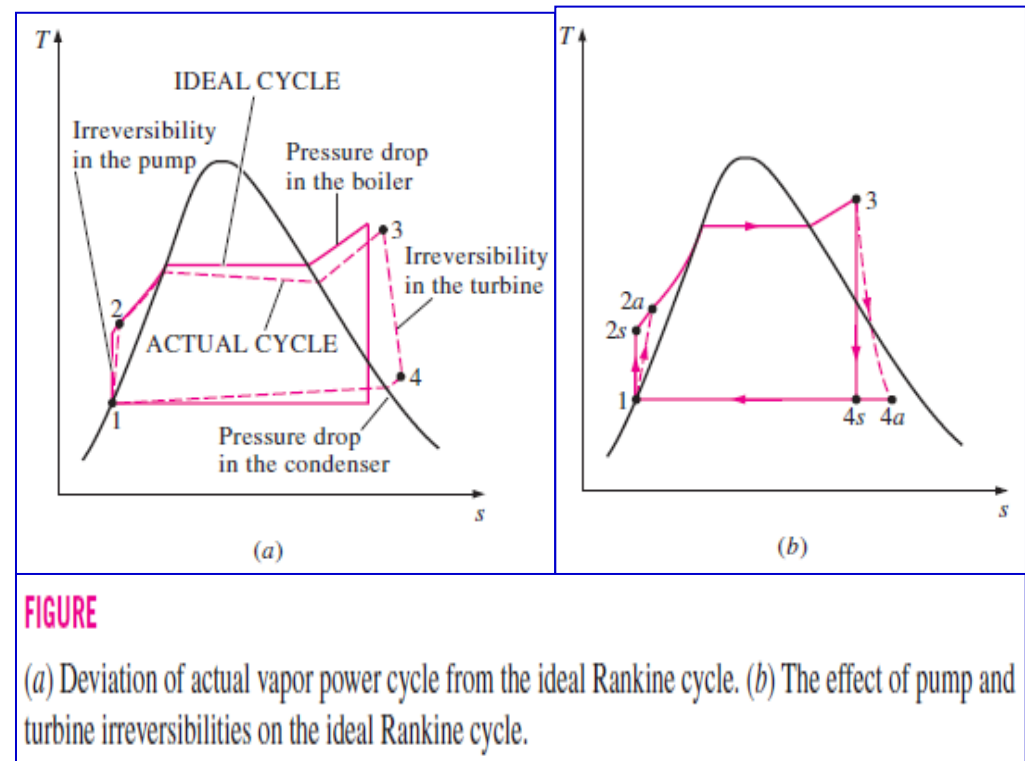
Example 1.1

Consider a steam power plant that operates on a simple ideal Rankine cycle and has a net power output of 45 MW. Steam enters the turbine at 7 MPa and 500 °C and is cooled in the condenser at a pressure of 10 kPa by running cooling water from a lake through the tubes of the condenser at a rate of 2000 kg/sec.

- (a) Show the cycle on a T-s diagram with respect to saturation lines, and determine
- (b) the thermal efficiency of the cycle
- (c) the mass flow rate of the steam, and
- (d) the temperature rise of the cooling water.

8.3. Deviation of Actual Vapor Power Cycles from Idealized Ones

- The actual vapor power cycle differs from ideal Rankine cycle, as illustrated in the figure, as a result of irreversibilities in various components.
- **Fluid friction** and **undesired heat loss to the surroundings** are the two most common sources of irreversibilities.



- **Fluid friction** causes pressure drops in the **boiler**, the **condenser**, and the **piping** between various components.
 - As a result, steam leaves the boiler at a somewhat lower pressure.
 - the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to the **pressure drop** in the **connecting pipes**.
 - The pressure drop in the condenser is usually very small. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle calls for. This requires a **larger pump** and larger work input to the pump.
- The other major source of irreversibility is the **heat loss from the steam to the surroundings** as the steam flows through various components.
 - To maintain the same level of net work output, more heat needs to be transferred to the steam in the boiler to compensate for these undesired heat losses.
 - As a result, cycle efficiency decreases.

- Of particular importance are the *irreversibilities* occurring within the *pump* and the *turbine*.
- A pump requires a greater work input, and a turbine produces a smaller work output as a result of irreversibilities.
- Under ideal conditions, the flow through these devices is *isentropic*.
- Internal irreversibility of Rankine cycle is caused by fluid friction, throttling and mixing.
- As the flow rates in the steam turbine as well as in the pumps are large, and the expansion and compression processes are quite rapid, the heat loss per unit mass may be considered negligible.
- Though the assumption of adiabatic flow in them is still valid, due to fluid friction the expansion and compression processes are not reversible and entropy of the fluid in both increases.
- The internal or isentropic efficiency (η_T) of the turbine is given by

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

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

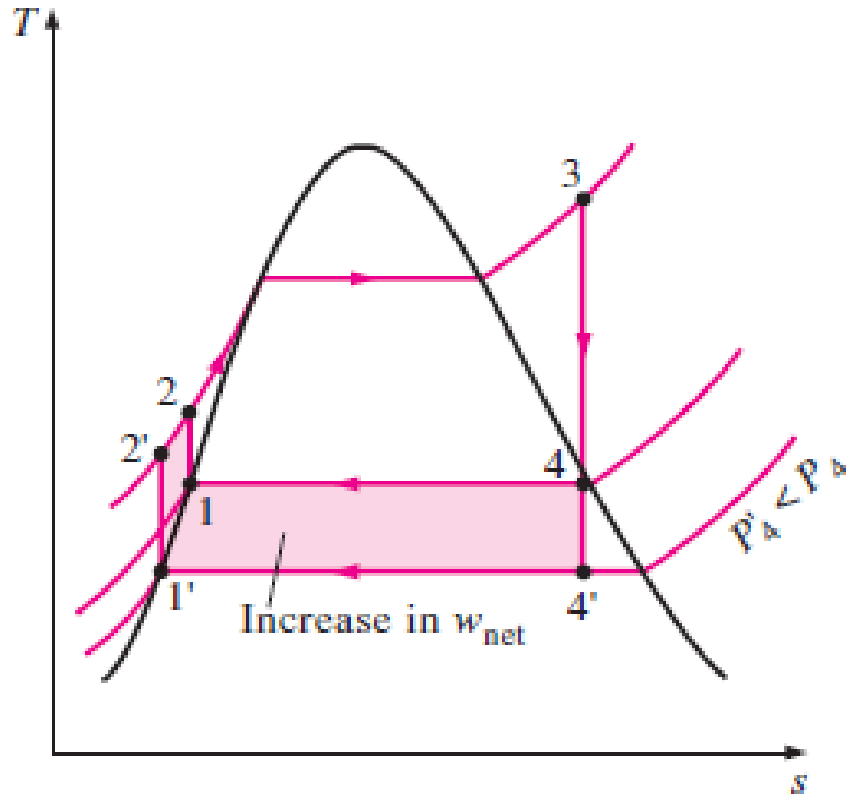
How can we increase the efficiency of the Rankine Cycle?

- 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
- Superheating the steam to high temperatures
- Increasing the boiler pressure

Lowering the Condenser Pressure (Lowers $T_{\text{low,av}}$)



- The colored area on this diagram represents the increase in *net work output as a result of lowering the condenser* pressure from P_4 to P'_4 .
- The heat input requirements also increase, but this increase is very small.
- The *overall effect* of lowering the condenser pressure is an *increase* in the *thermal efficiency* of the cycle.

Decrease condenser pressure

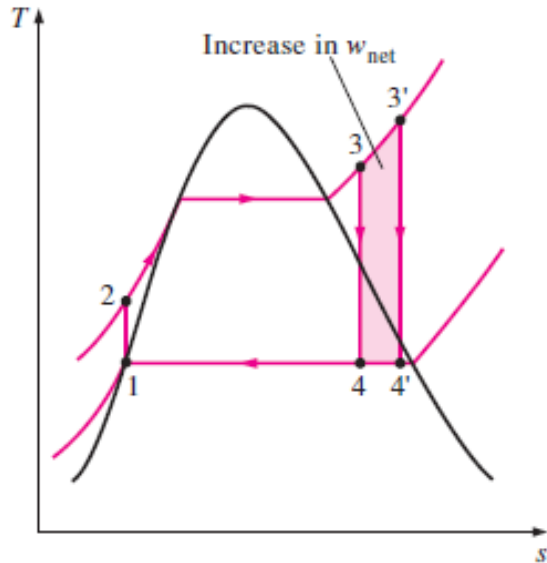
Good news

- Increase in the net Work.
- Increase in the efficiency.

Bad news

- Increase in Q_H (but this is small compared to the increase in the net work).
- Limited by the saturation pressure corresponding to the cooling medium (ex: river, sea, lake)
- More moisture within the turbine.

Superheating the Steam to High Temperatures (Increases $T_{\text{high,av}}$)



- The colored area on this diagram represents the increase in the **net work**.
 - The total area under the process curve 3-3' represents the increase in the **heat input**.
 - Thus, 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**.
-
- The temperature to which **steam** can be **superheated** is limited by **metallurgical considerations**.
 - Presently, the **highest steam temperature allowed** at the turbine inlet is about 620 °C(1150 °F).

Superheating the steam

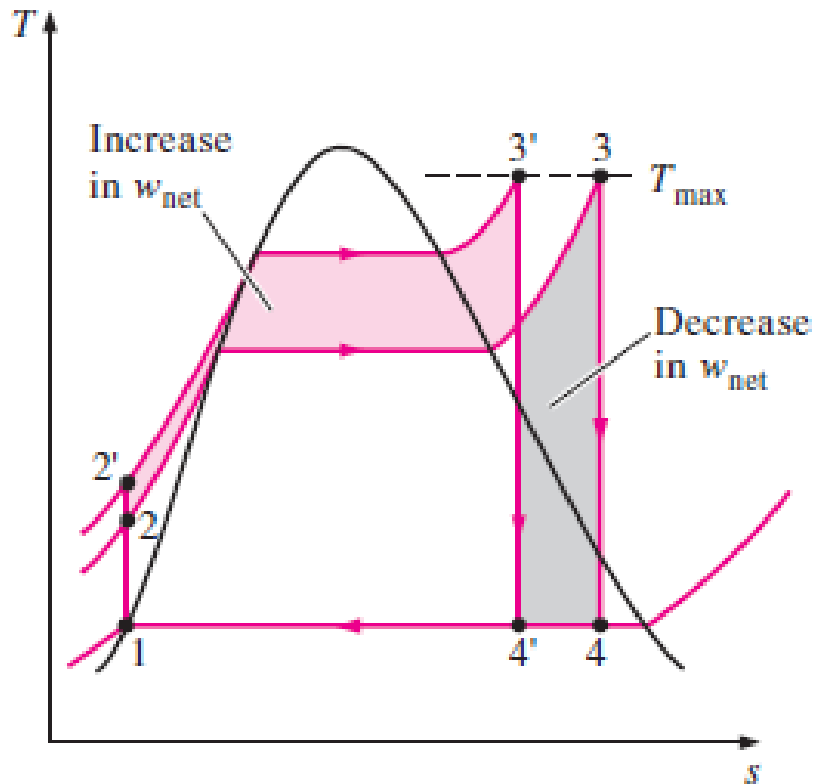
Good news

- Increase in the net Work.
- Increase in the efficiency.
- Decrease moisture.

Bad news

- Significant increase in Q_H .
- Metallurgical limitations ($T_3 < 620\text{ }^{\circ}\text{C}$) to protect turbine blades.

Increasing the Boiler Pressure (Increase $T_{\text{high,av}}$)



- Another way of increasing the average temperature during the heat-addition process is to increase the operating pressure of the boiler, which automatically raises the temperature at which boiling takes place.
- This, in turn, raises the average temperature at which heat is added to the steam and thus **raises** the **thermal efficiency** of the cycle.

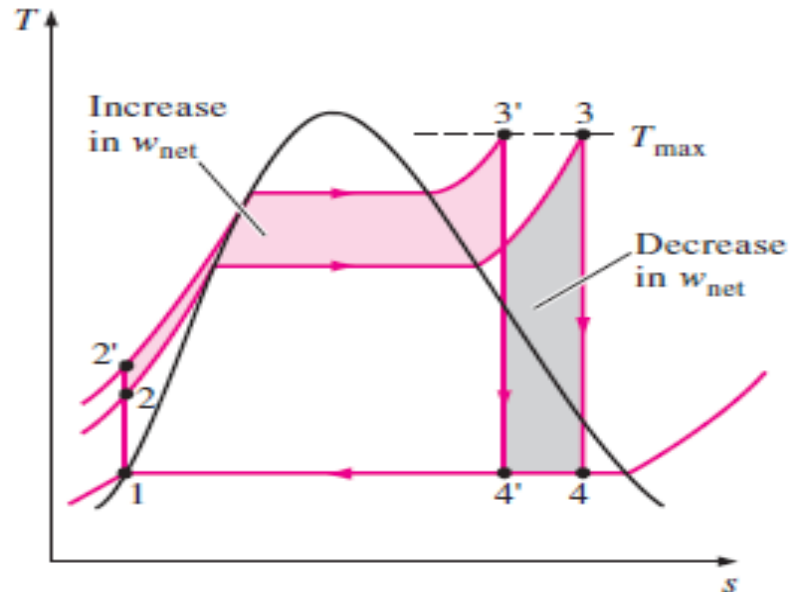
Increase boiler pressure

Good news	Bad news
- Increase Temperature at which boiling take place (reduce Q_H). - Increase in the efficiency.	- increase moisture.

8.4. The Ideal Reheat Rankine Cycle

- ***Increasing the boiler pressure*** increases the thermal efficiency of the Rankine cycle, but it also ***increases the moisture content of the steam*** to unacceptable levels.
- Then, it is natural to ask the following question:

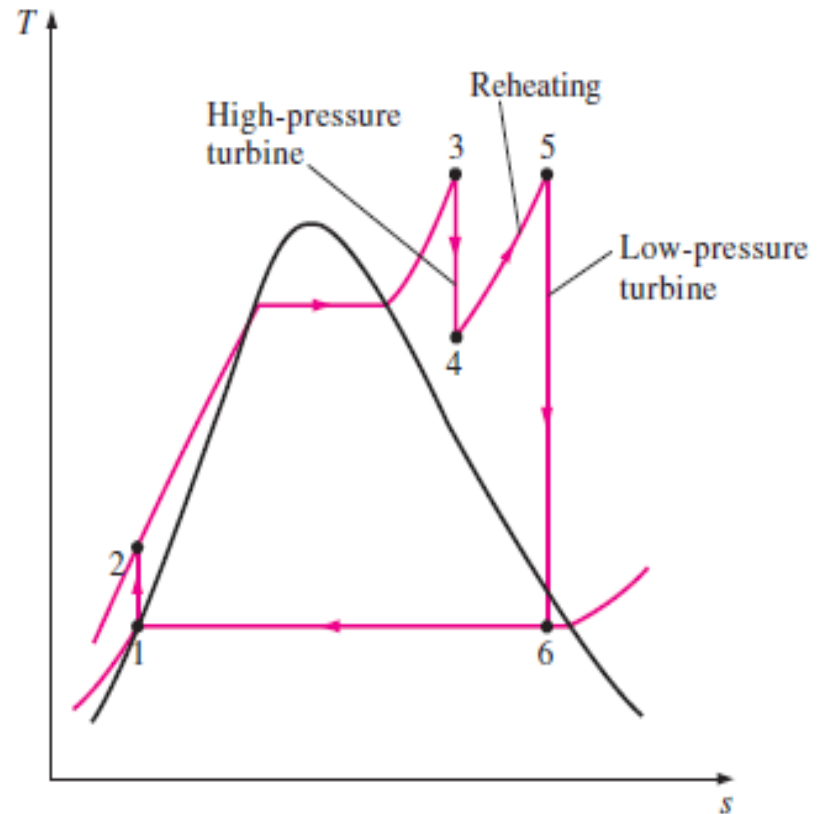
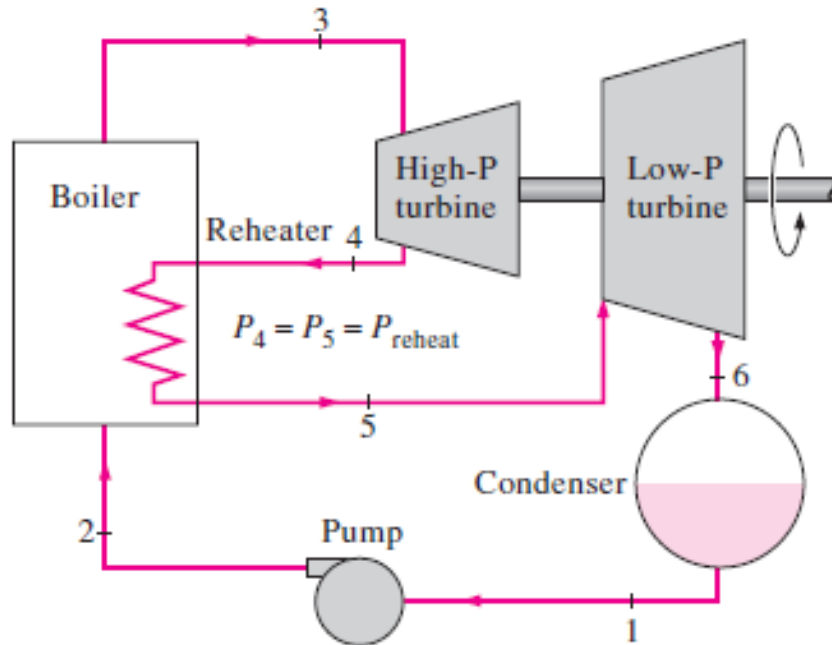
How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?



Two possibilities 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 will require 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 used frequently in modern steam power plants.**

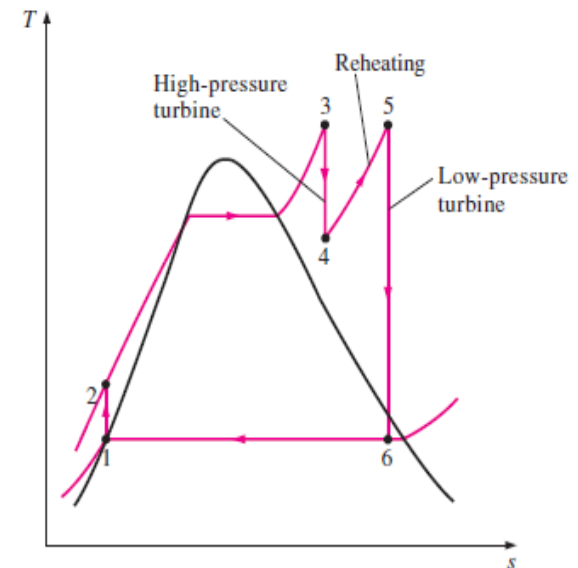
The T-s diagram of the *ideal reheat Rankine cycle* and the schematic of the power plant operating on this cycle are shown below:



- 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 (**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.
- The total heat input and the total turbine work output for a reheat cycle become

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

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



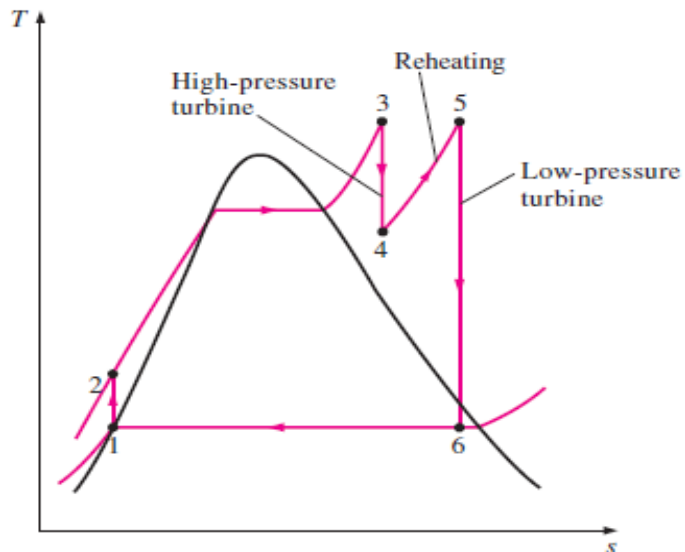
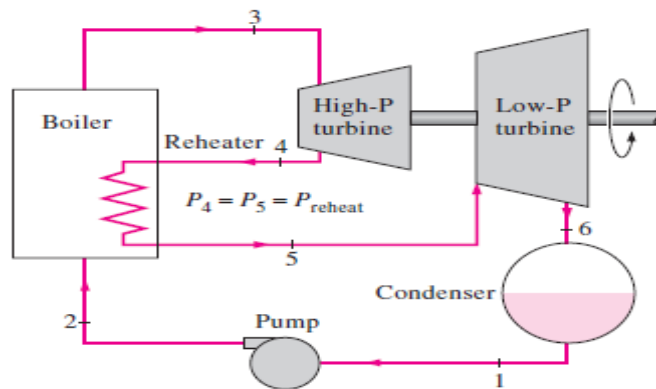
Advantages & disadvantages

• Advantages

1. *Increases thermal eff.*
2. *Increases dryness fraction of the steam at turbine exhaust, this will reduce blade erosion.*
3. *Increase work done per unit mass of steam, thus reducing boiler size.*

• Disadvantages

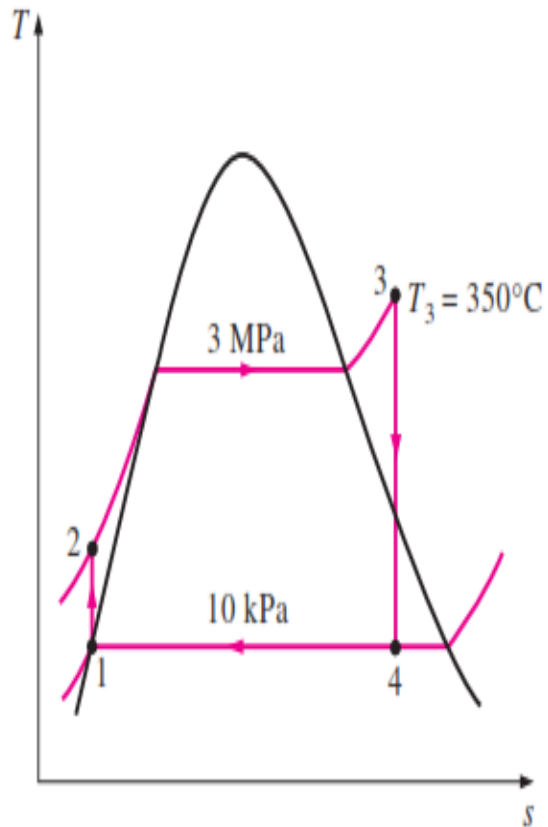
1. *Increases plant cost due to re-heater requirement and it's long piping system.*
2. *Increases condenser capacity due to the increase in steam dryness fraction.*



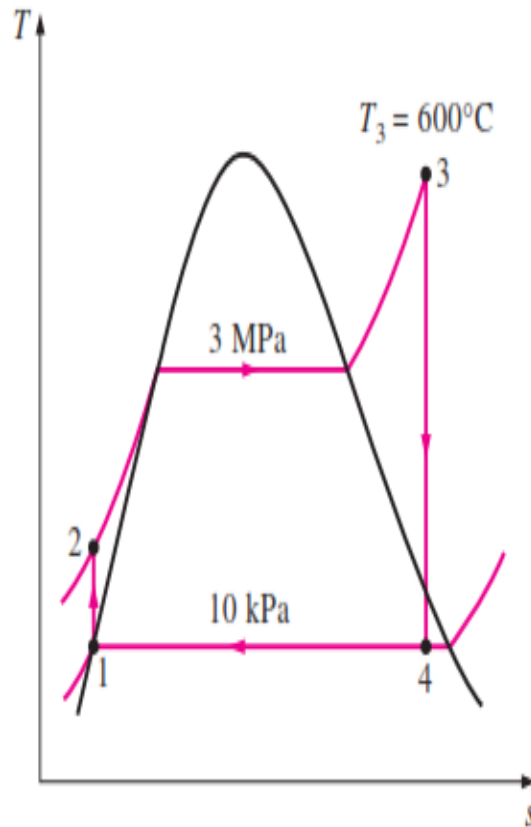
Example 1.2

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.

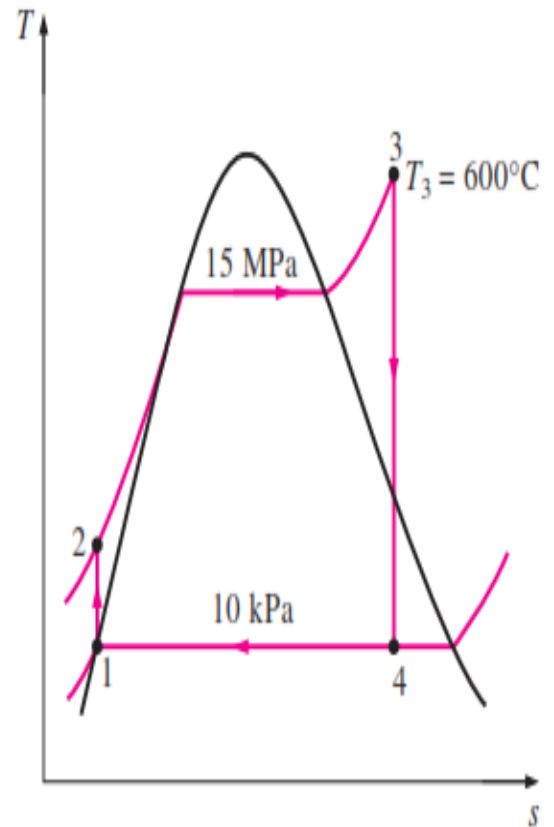
Con't...



(a)



(b)



(c)

Cont...

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

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

$$\begin{aligned} w_{\text{pump,in}} &= v_1 (P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg}) [(3000 - 10) \text{ kPa}] \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= 3.02 \text{ kJ/kg} \end{aligned}$$

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

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

$$\text{State 4: } \begin{array}{l} P_4 = 10 \text{ kPa} \quad (\text{sat. mixture}) \\ s_4 = s_3 \end{array}$$

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

Con't...

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

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

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

and

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

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

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

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

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

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

Con't...

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

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

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

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

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

Thus,

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

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

and

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

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).

Example 1.3

Consider a steam power plant that operates on a reheat Rankine cycle and has a net power output of 80 MW. Steam enters the high-pressure turbine at 10 MPa and 500 °C and the low-pressure turbine at 1 MPa and 500 °C. Steam leaves the condenser as saturated liquid at a pressure of 10 kPa.

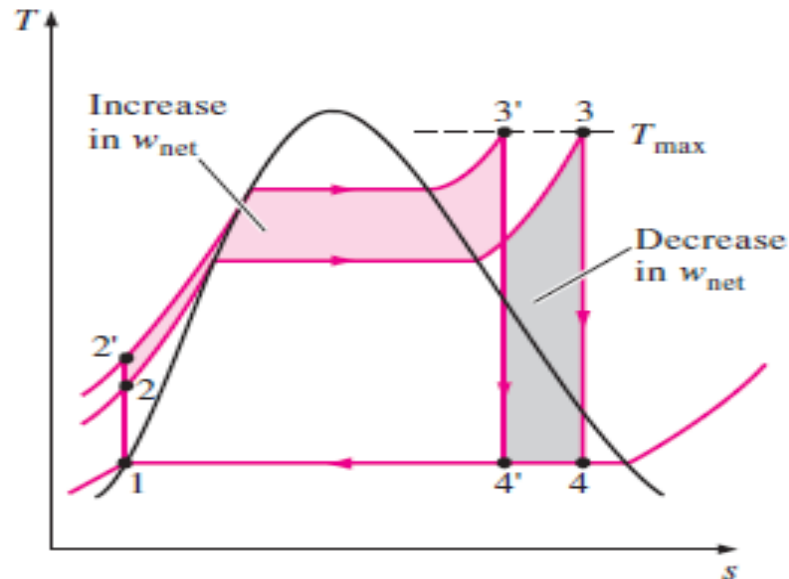
The isentropic efficiency of the turbine is 80 percent, and that of the pump is 95 percent.

- (a) show the cycle on a T-s diagram with respect to saturation lines, and determine
- (b) the quality (or temperature, if superheated) of the steam at the turbine exit,
- (c) the mass flow rate of the steam.

The Ideal Reheat Rankine Cycle

- *Increasing the boiler pressure* increases the thermal efficiency of the Rankine cycle, but it also *increases the moisture content of the steam* to unacceptable levels.
- Then, it is natural to ask the following question:

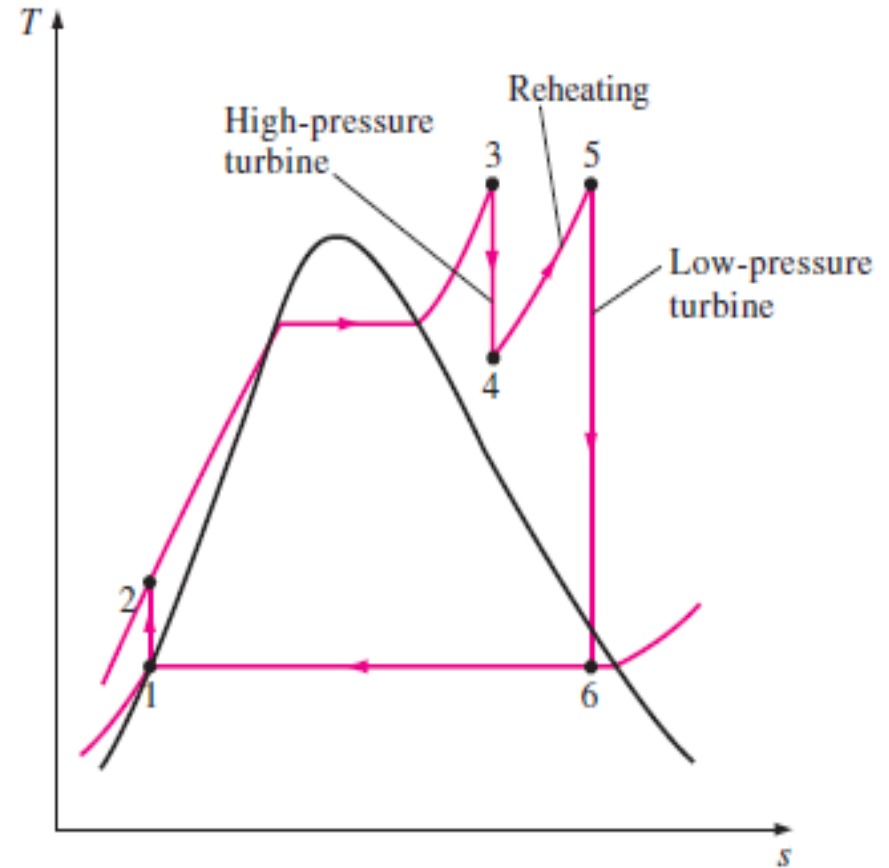
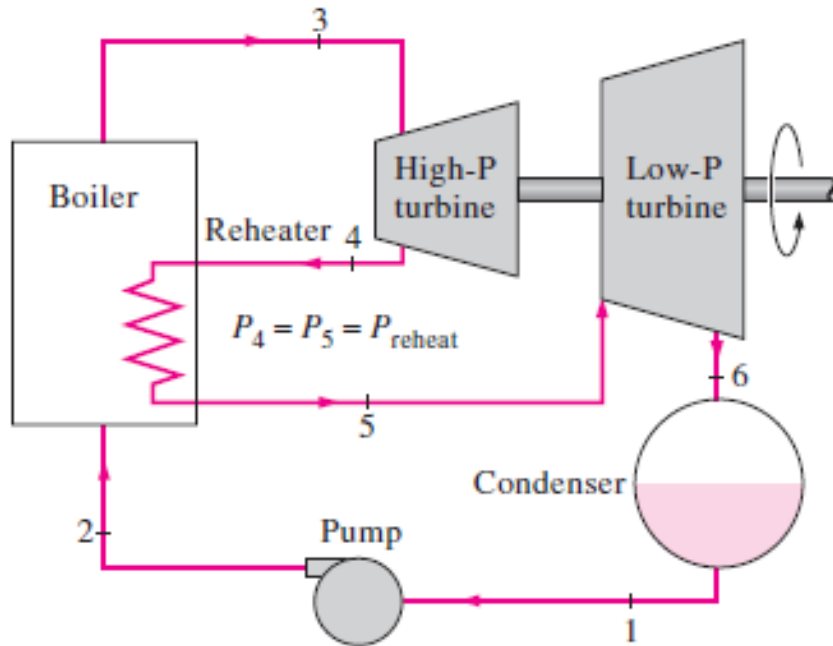
How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?



Two possibilities 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 will require 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 used frequently in modern steam power plants.**

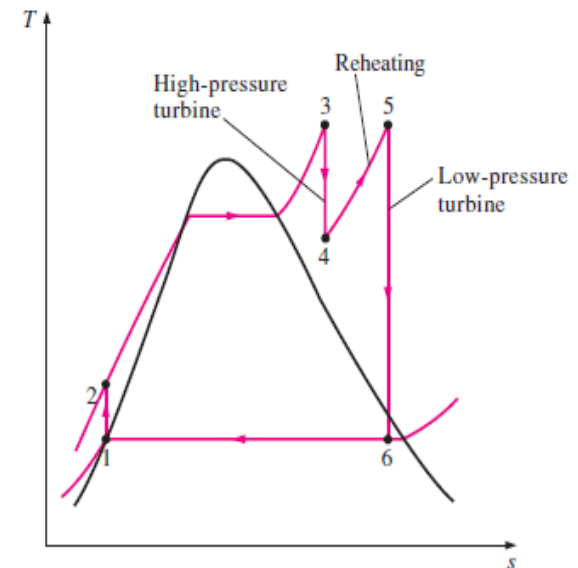
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- The ideal Reheat Rankine cycle differs from the simple ideal Rankine cycle in that the expansion process takes place in two stages.
- In the first stage (**high-pressure turbine**), steam is expanded isentropically to an intermediate pressure and 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.
- The total heat input and the total turbine work output for a reheat cycle become

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

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



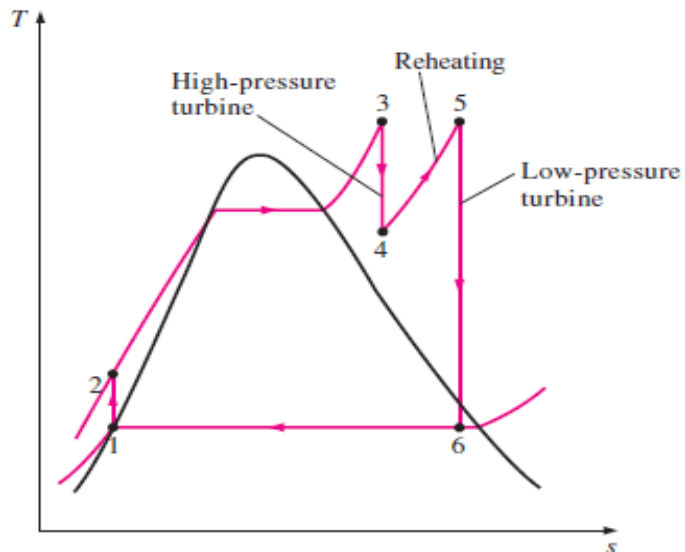
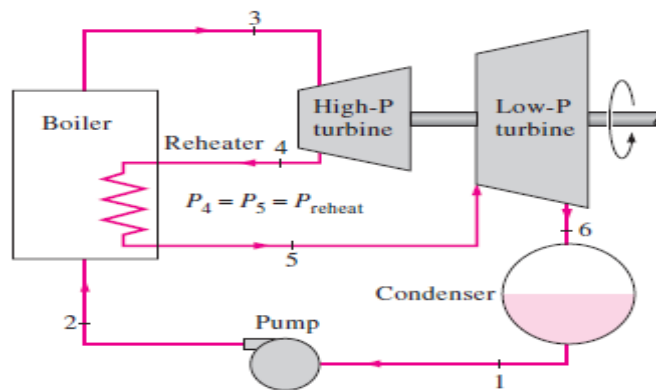
Advantages & disadvantages

• Advantages

1. *Increases thermal eff.*
2. *Increases dryness fraction of the steam at turbine exhaust, this will reduce blade erosion.*
3. *Increase work done per unit mass of steam, thus reducing boiler size.*

• Disadvantages

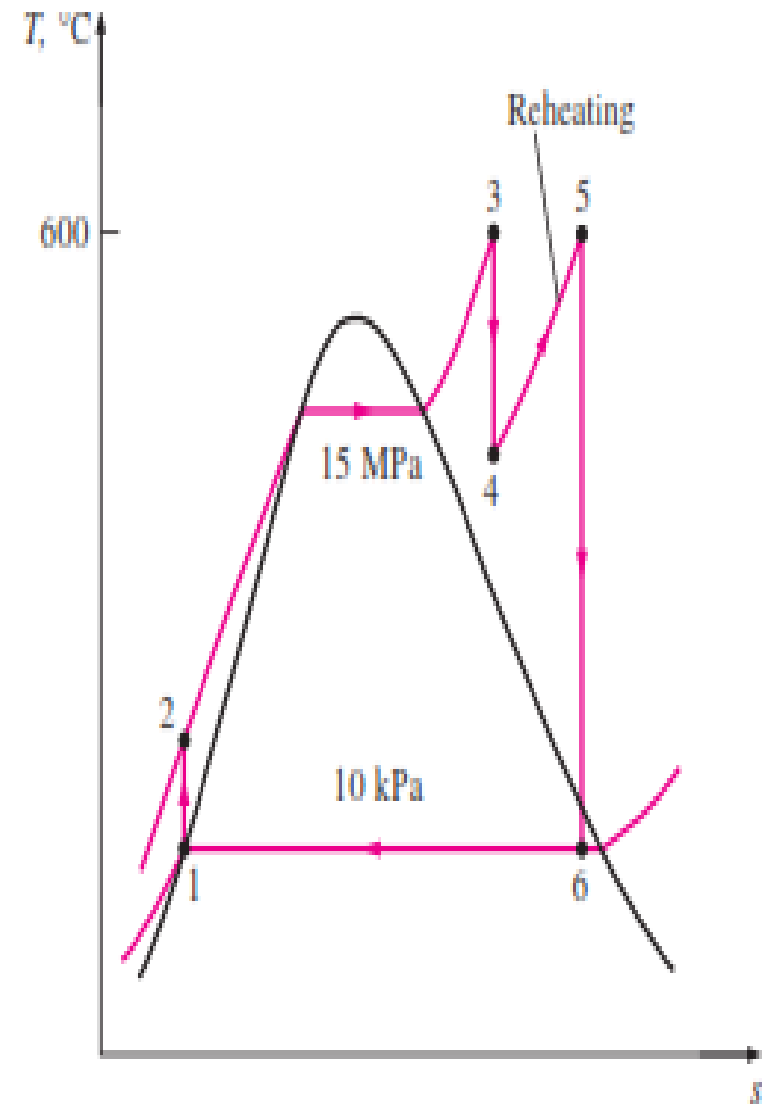
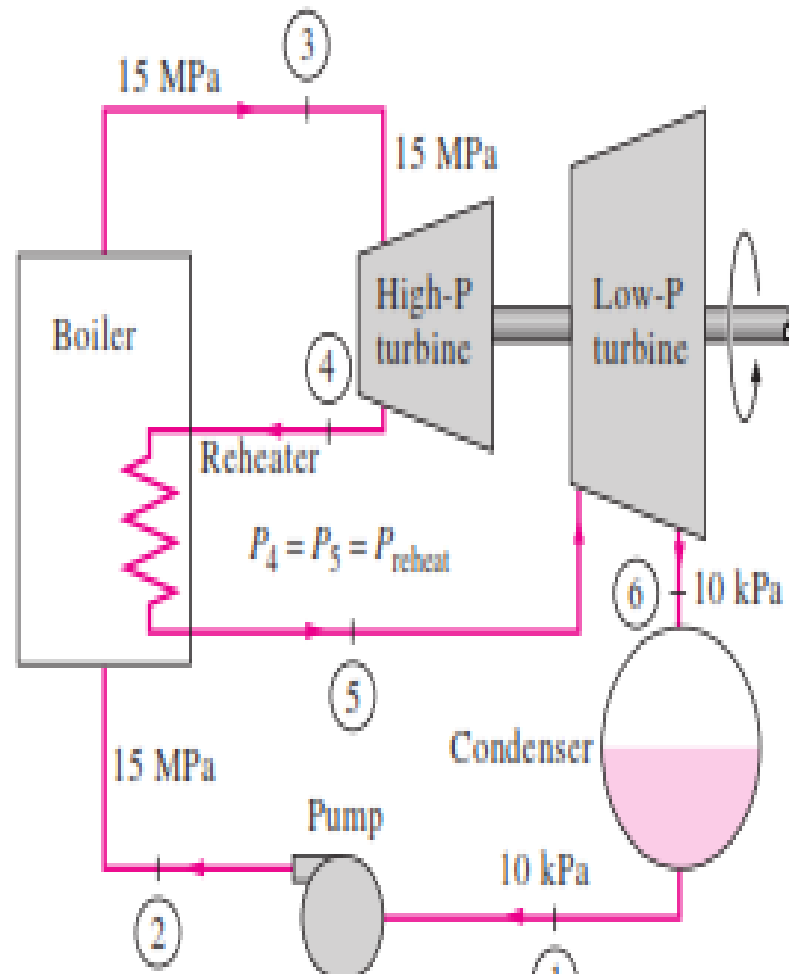
1. *Increases plant cost due to re-heater requirement and it's long piping system.*
2. *Increases condenser capacity due to the increase in steam dryness fraction.*



Example 1.4

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.

Con't....



Cont...

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

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

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

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

Also,

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

Thus,

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

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

Con't...

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

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

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

$$s_2 = s_1$$

$$\begin{aligned} w_{\text{pump,in}} &= v_1 (P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg}) \\ &\quad \times [(15,000 - 10) \text{ kPa}] \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) \\ &= 15.14 \text{ kJ/kg} \\ h_2 &= h_1 + w_{\text{pump,in}} = (191.81 + 15.14) \text{ kJ/kg} = 206.95 \text{ kJ/kg} \end{aligned}$$

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

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

Thus

$$\begin{aligned} q_{\text{in}} &= (h_3 - h_2) + (h_5 - h_4) \\ &= (3583.1 - 206.95) \text{ kJ/kg} + (3674.9 - 3155.0) \text{ kJ/kg} \\ &= 3896.1 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{\text{out}} &= h_6 - h_1 = (2335.1 - 191.81) \text{ kJ/kg} \\ &= 2143.3 \text{ kJ/kg} \end{aligned}$$

Con't...

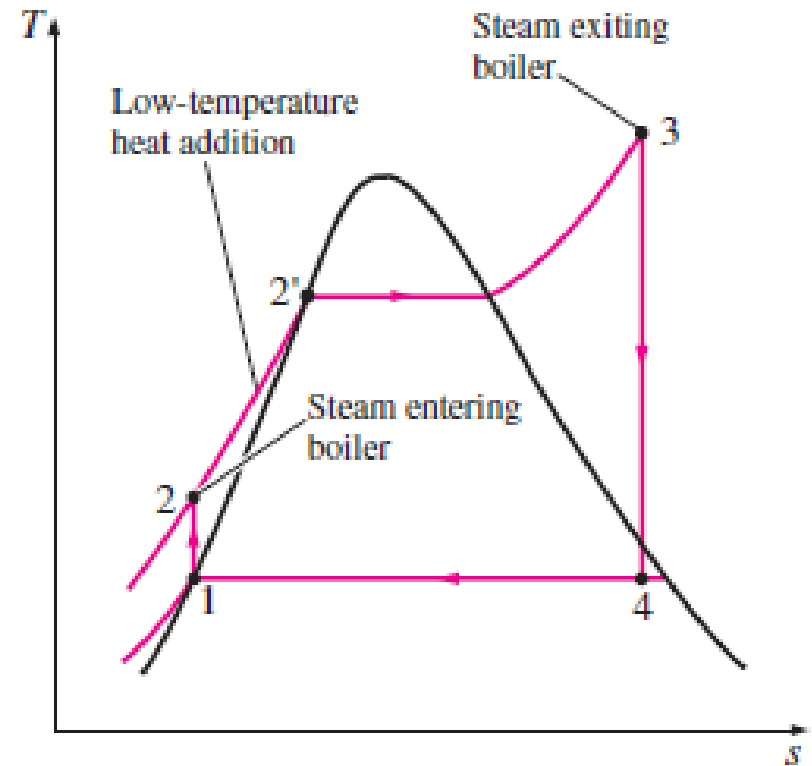
and

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

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

8.5. The Ideal Regenerative Rankine Cycle

- A careful examination of the T-s diagram of the Rankine cycle shown reveals that heat is added to the working fluid during process 2-2' at a **relatively low temperature**.
- This lowers the average temperature at which heat is added and thus the cycle efficiency.



- To remedy this shortcoming, we look for ways to raise the temperature of the liquid leaving the pump (called the *feedwater*) before it enters the boiler.
1. One such possibility is to *compress the feed water* isentropically to a high temperature, as in the Carnot cycle.
 - This, however, would involve extremely high pressures and is therefore *impractical*.
 2. Another possibility is to transfer heat to the feedwater from the *expanding steam in a counter flow heat exchanger built into the turbine*, that is, to use **regeneration**.
 - This solution is also *impractical* because it is difficult to design such a heat exchanger and because **it would increase the moisture content of the steam at the final stages of the turbine**.

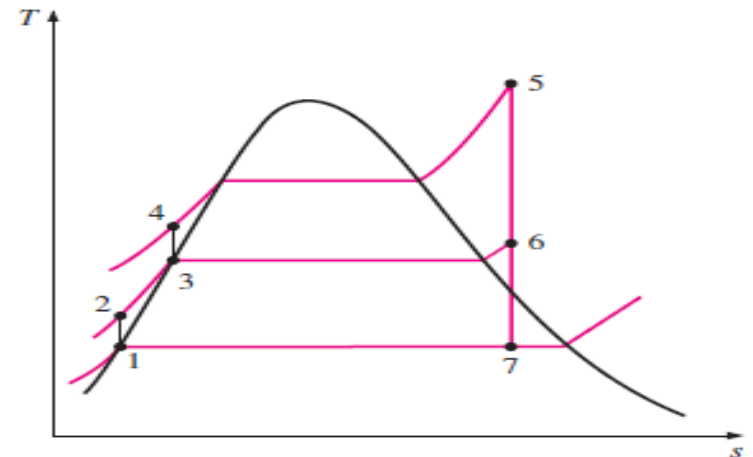
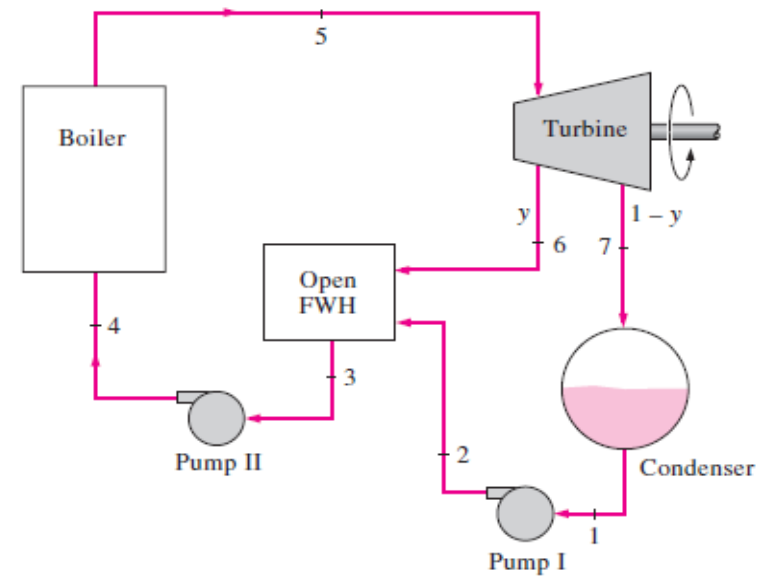
Regeneration

- A *practical regeneration process* in steam power plants is accomplished by extracting, or “*bleeding*,” steam 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*.
- 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*).

Types of Feed-water Heaters

Open Feedwater Heaters

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



- In the analysis of steam power plants, it is more convenient to work with quantities expressed per unit mass of the steam flowing through the boiler.
- For each 1 kg of steam leaving the boiler, y kg expands partially in the turbine and is extracted at state 6.
- The remaining $(1-y)$ kg expands completely to the condenser pressure.
- Therefore, the mass flow rate through the boiler is m , for example, it will be $(1-y)m$ through the condenser.
- The heat and work interactions of a regenerative Rankine cycle with one feedwater heater can be expressed per unit mass of steam flowing through the boiler as follows:

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

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

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

$$w_{Pump,in} = (1 - y)w_{PumpI,in} + w_{PumpII,in}$$

where

$$y = \dot{m}_6 / \dot{m}_5 \quad (\text{fraction of steam extracted})$$

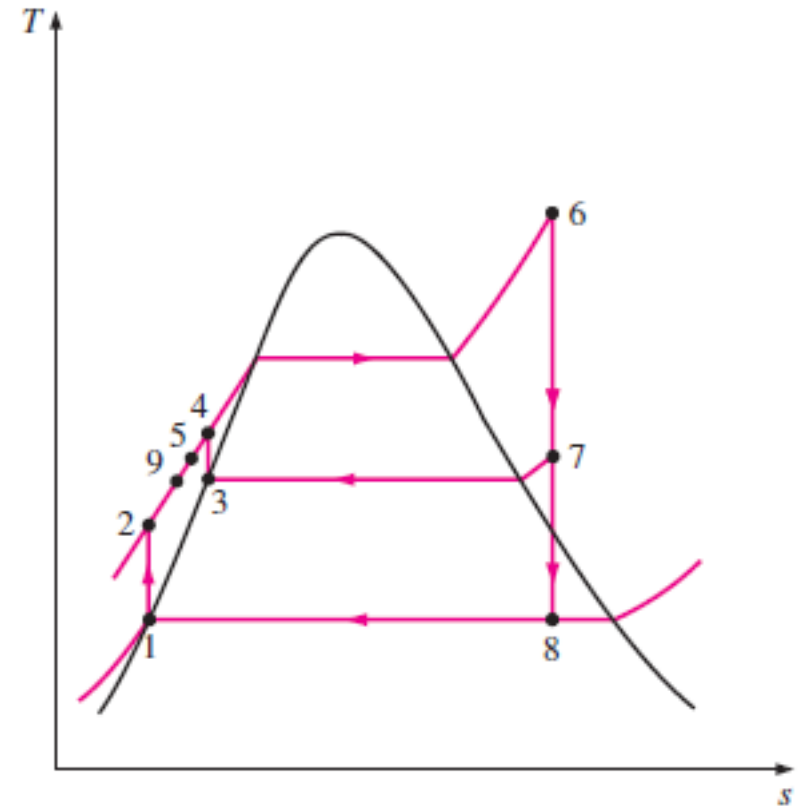
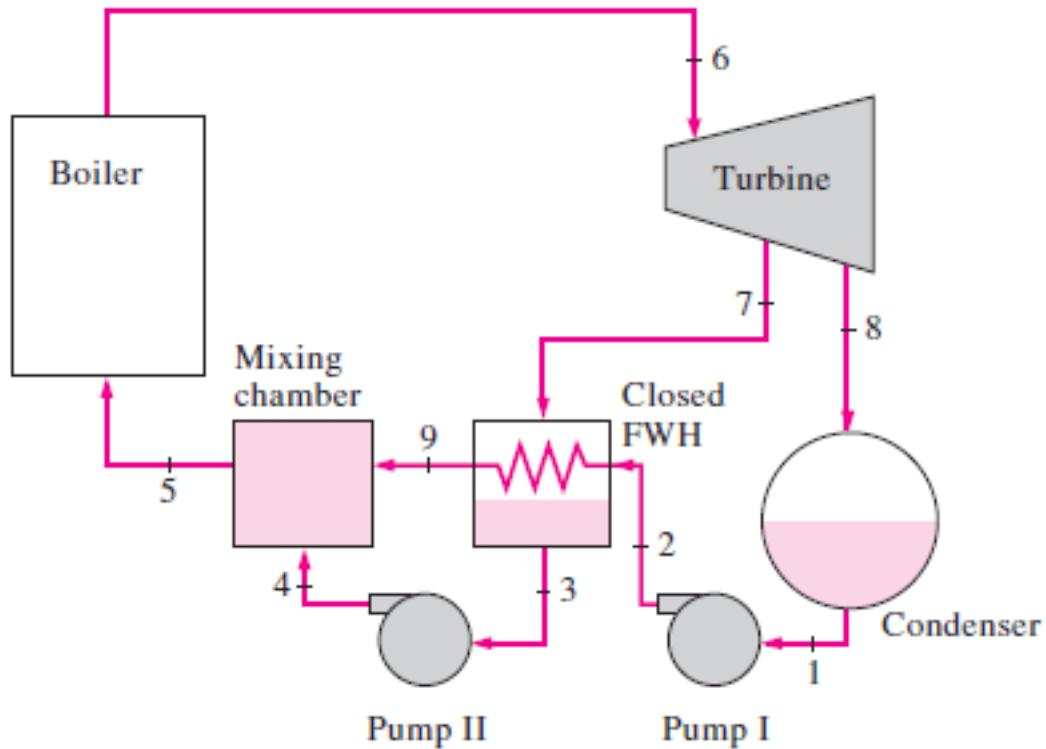
$$w_{PumpI,in} = v_1 (P_2 - P_1)$$

$$w_{PumpII,in} = v_3 (P_4 - P_3)$$

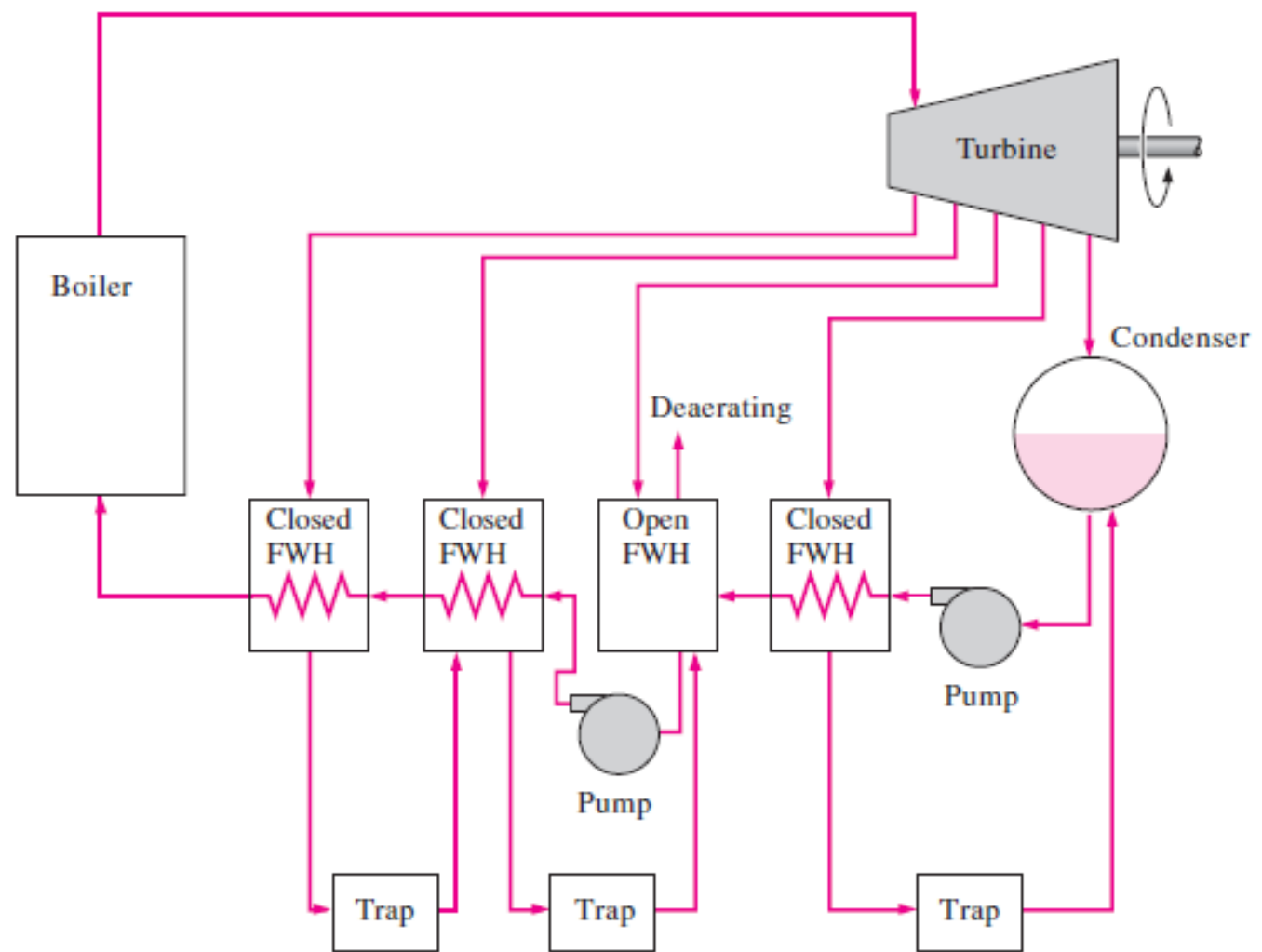
Closed Feedwater Heaters

- Closed feedwater heater- heat is transferred from the extracted steam to the feedwater without any mixing taking place.
- The two streams can be at different pressures, since they do not mix.
- In an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.
- In actual power plants, the feedwater leaves the heater below the exit temperature of the extracted steam because a temperature difference of at least a few degrees is required for any effective heat transfer to take place.

Closed Feedwater Heaters



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

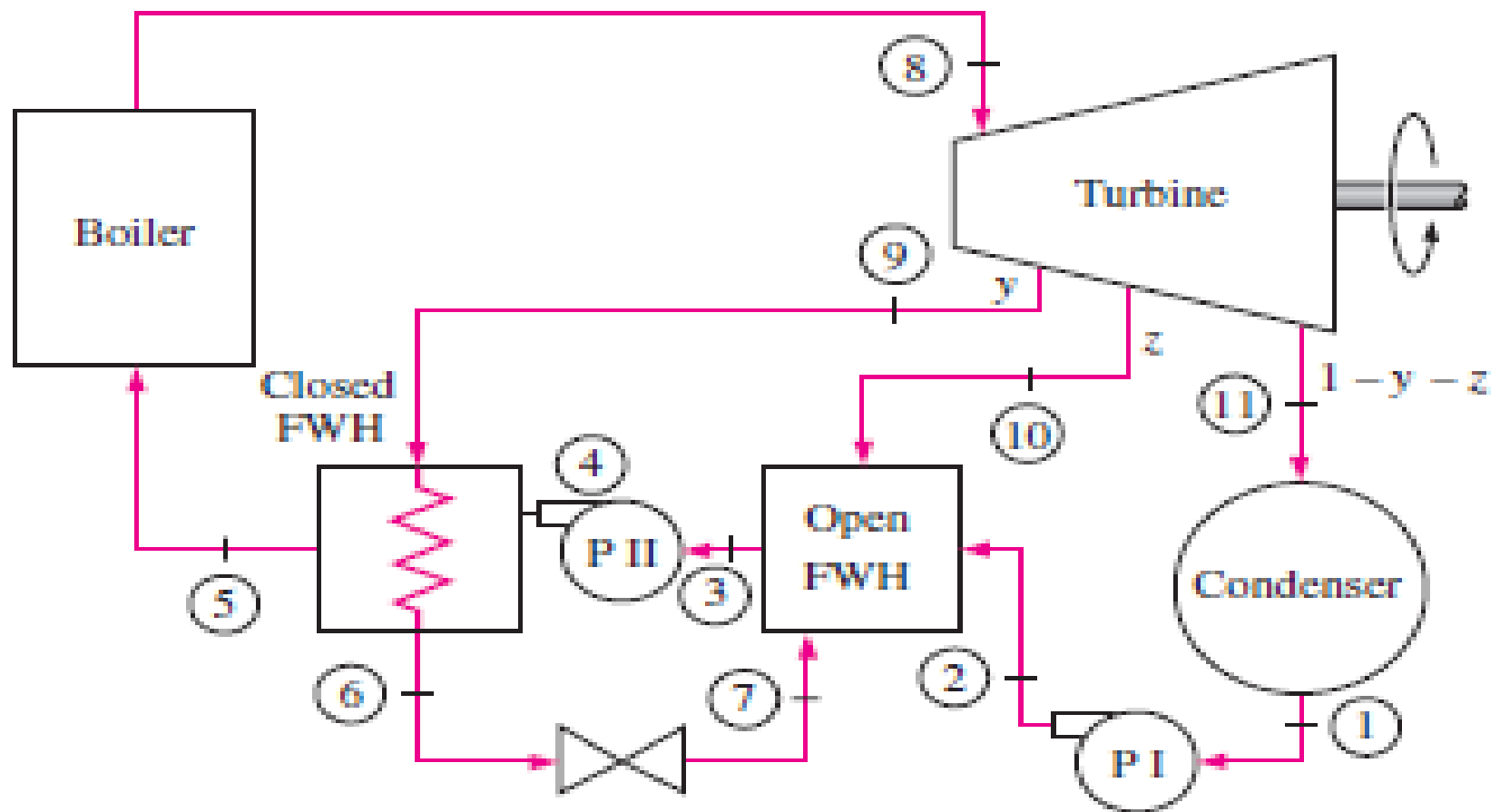


Example 1.5

❖ Consider an ideal steam regenerative Rankine cycle with two feed water heaters, one closed and one open. Steam enters the turbine at 12.5 MPa and 550 °C and exhausts to the condenser at 10 kPa. Steam is extracted from the turbine at 0.8 MPa for the closed feed water heater and at 0.3 MPa for the open one. The feed water heater is heated to the condensation temperature of the extracted steam in the closed feed water heater. The extracted steam leaves the closed feed water heater as a saturated liquid, which is subsequently throttled to the open feed water heater.

Example (Cont...)

- (a) Show the cycle on a T-s diagram with respect to saturation lines, and determine
- (b) the mass flow rate of steam through the boiler for a net power output of 250 MW, and
- (c) the thermal efficiency of the cycle.



The End !!!!!