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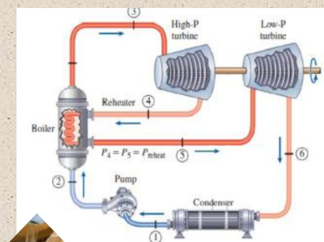
عنوان المحاضرة: دورات القدرة البخارية 1

Thermodynamic II

LECTURE 4

Vapor Power Cycles I



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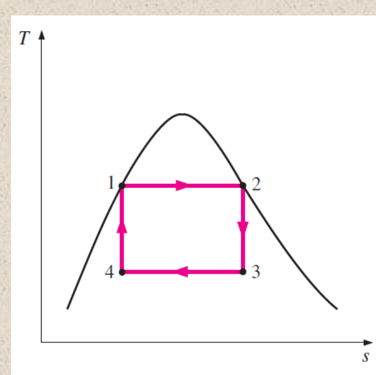
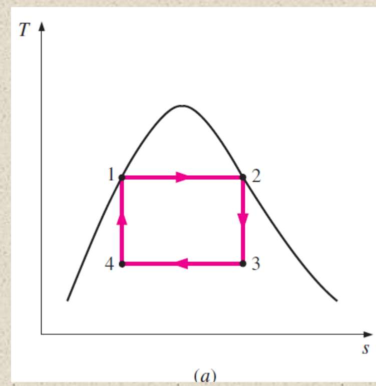
THE CARNOT VAPOR CYCLE

- We have mentioned repeatedly that the Carnot cycle is the most efficient cycle operating between two specified temperature limits.
- Thus it is natural to look at the Carnot cycle first as a prospective ideal cycle for vapor power plants. If we could, we would certainly adopt it as the ideal cycle.
- As explained below, however, the Carnot cycle is not a suitable model for power cycles.
- Throughout the discussions, we assume *steam* to be the working fluid since it is the working fluid predominantly used in vapor power cycles.



T-s diagram of Carnot vapor cycles.

- Consider a steady-flow *Carnot cycle* executed within the saturation dome of a pure substance, as shown in Fig.
- 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).

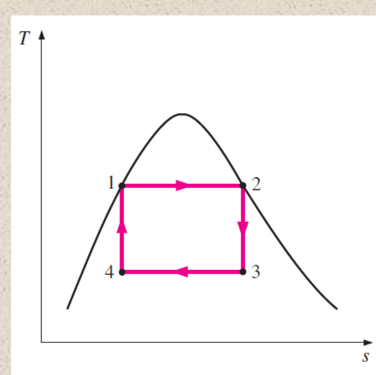


T-s diagram of Carnot vapor cycles.

- Several impracticalities are associated with this cycle:

1. Isothermal heat transfer to or from a two-phase system is not difficult to achieve in practice since maintaining a constant pressure in the device automatically fixes the temperature at the saturation value. Therefore, processes 1-2 and 3-4 can be approached closely in actual boilers and condensers.

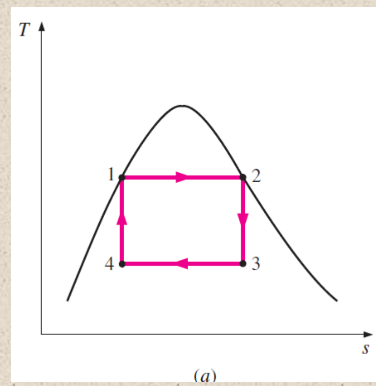
- Limiting the heat transfer processes to two-phase systems, however, severely limits the maximum temperature that can be used in the cycle (it has to remain under the critical-point value, which is 374°C for water).
- Limiting the maximum temperature in the cycle also limits the thermal efficiency.



T-s diagram of Carnot vapor cycles.

2. The isentropic expansion process (process 2-3) can be approximated closely by a well-designed turbine. However, the quality of the steam decreases during this process, as shown on the *T-s* diagram in Fig.

- Thus the turbine has to handle steam with low quality, that is, steam with a high moisture content. The impingement of liquid droplets on the turbine blades causes erosion and is a major source of wear.
- Thus steam with qualities less than about 90 percent cannot be tolerated in the operation of power plants.



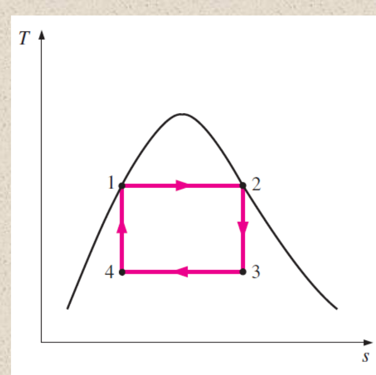
T-s diagram of Carnot vapor cycles.

3. The isentropic compression process (process 4-1) involves the compression of a liquid–vapor mixture to a saturated liquid.

There are two difficulties associated with this process.

- First, it is not easy to control the condensation process so precisely as to end up with the desired quality at state 4.
- Second, it is not practical to design a compressor that handles two phases.

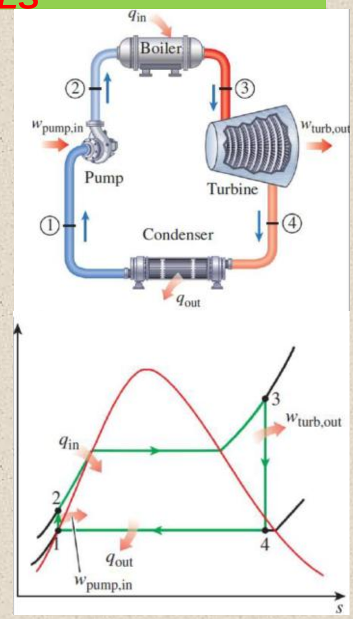
Thus we conclude that the Carnot cycle cannot be approximated in actual devices and is not a realistic model for vapor power cycles.



T-s diagram of Carnot vapor cycles.

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, as shown schematically on a *T-s* diagram in Fig.



The simple ideal Rankine cycle.

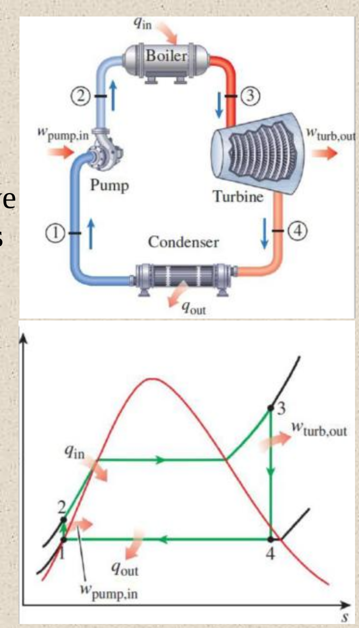
- The cycle that results is the **Rankine cycle**, which is the ideal cycle for vapor power plants.
- The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:

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



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

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

or,

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

where

$h_1 = h_{f@P1}$ and $v \cong v_1 = v_{f@P1}$

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

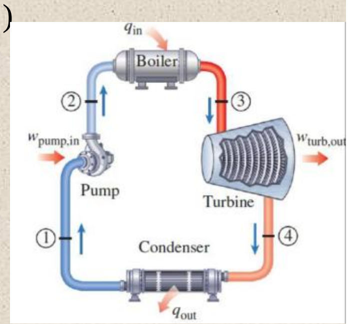
Turbine ($q = 0$): $w_{turb,out} = h_3 - h_4$

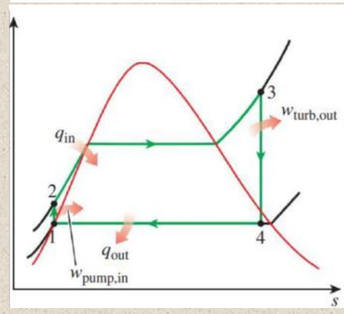
Condenser ($w = 0$): $q_{out} = h_4 - h_1$

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

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

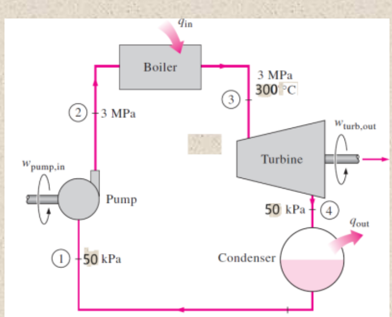
where $w_{net} = q_{in} - q_{out} = w_{turb,out} - w_{pump,in}$

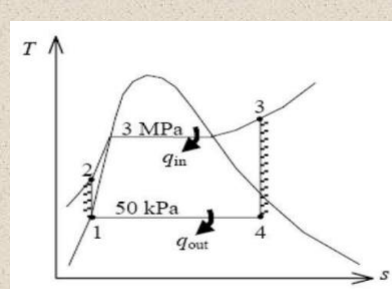




Example 1

A steam power plant operates on a simple ideal Rankine cycle between the pressure limits of 3 MPa and 50 kPa. The temperature of the steam at the turbine inlet is 300°C, and the mass flow rate of steam through the cycle is 35 kg/s. Show the cycle on a (T-S) diagram with respect to the saturation lines, and determine: (a) the thermal efficiency of the cycle (b) the net power output of the power plant.





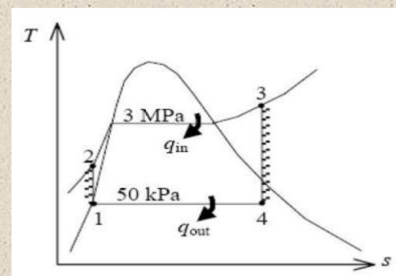
Solution:

$h_1 = h_f$ = 340.54 kJ/kg at P_1 = 50 kPa
$v_1 = v_f$ = 0.00103 m^3 /kg at P_1 = 50 kPa

$w_{pump} = v(P_2 - P_1)$ = 0.00103 × (3000 − 50)
= 3.04 kJ/kg

$w_{pump} = h_2 - h_1 \rightarrow h_2 = h_1 + w_{pump}$
= 340.54 + 3.04 = 343.58 kJ/kg

At P_3 = 3 Mpa and T_3 = 300°C → h_3 = 2994.3 kJ/kg
and s_3 = 6.5412 kJ/kg. K



$$\text{At } P_4 = 50 \text{ kPa and } s_4 = s_3 \rightarrow x_4 = \frac{s_4 - sf}{s_{fg}}$$

$$= \frac{6.5412 - 1.0912}{6.5019}$$

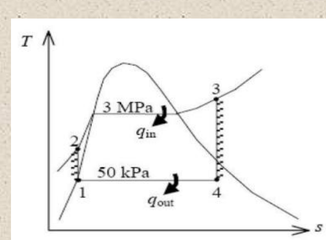
$$= 0.8382$$

$$h_4 = h_f + x_4 . h_{fg} = 340.54 + 0.8382 \times 2304.7 = 2272 \text{ kJ/kg}$$

$$q_{in} = h_3 - h_2 = 2994.3 - 343.58 = 2650.6 \text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = 2272.3 - 340.54 = 1931.8 \text{ kJ/kg}$$

$$w_{net} = q_{in} - q_{out} = 2650.6 - 1931.8 = 718.9 \text{ kJ/kg}$$



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

$$= 1 - \frac{1931.8}{2650.7}$$

$\boldsymbol{\eta th} = \mathbf{27.1\%}$ **Ans.**

$$Power = \dot{m} \times w_{net} = 35 \times 718.9$$

$\boldsymbol{Power} =$ 25 200 kW $=$ **25.2 MW** **Ans.**

Actual Rankine Cycle

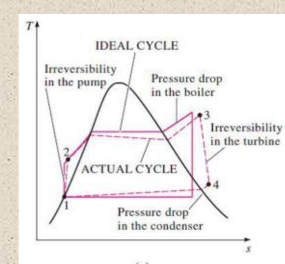
- The actual cycle differs from the ideal Rankine cycle as a result of irreversibilities in various components.
- Fluid friction and heat loss to the surroundings are the main sources of irreversibilities.
- Fluid friction causes pressure drops in the boiler, condenser and the piping between various components. As a result, steam leaves the boiler at a somewhat lower pressure. Also, 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 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.

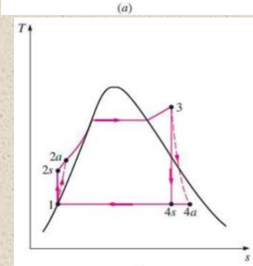
- Of particular importance are the irreversibilities occurring within the **pump** and the **turbine**.
- A pump requires a greater work input, and a turbine produces a smaller work output as a result of irreversibilities.
- Under ideal conditions, the flow through these devices is isentropic.
- The deviation of actual pumps and turbines from the isentropic ones can be accounted for by utilizing isentropic efficiencies, defined as:

$$\eta_P = \frac{w_{Ps}}{w_{Pa}} = \frac{h_{2s} - h_1}{h_{2a} - h_1} \qquad \text{For pump}$$

$$\eta_T = \frac{w_{Ta}}{w_{Ts}} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \qquad \text{For turbine}$$

Where states **2a** and **4a** are the actual exit states of the pump and the turbine respectively, and **2s** and **4s** are the corresponding states for the isentropic case.

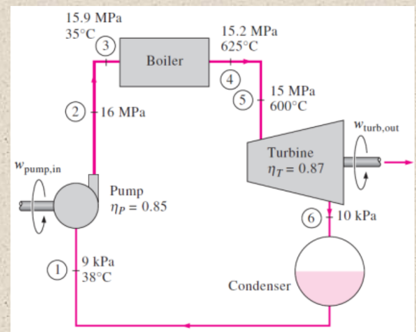


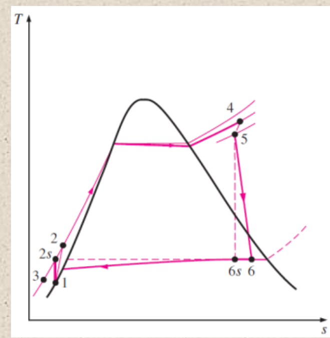


(*a*) Deviation of actual vapor power cycle from the ideal Rankine cycle. (*b*) The effect of pump and turbine irreversibilities on the ideal Rankine cycle.

Example 2

A steam power plant operates on the cycle shown in the figure below. 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:

The thermal efficiency of a cycle is the ratio of the net work output to the heat input.

$$\eta_p = \frac{W_{ps}}{W_{pa}} = \frac{v_1(P_2 - P_1)}{W_{pa}} \rightarrow W_{pa} = \frac{v_1(P_2 - P_1)}{\eta_p}$$

$$W_{pa} = \frac{(0.001009)(16 \times 10^3 - 9)}{0.85} = 18.98 \quad kj/kg$$

$$\eta_T = \frac{W_{Ta}}{W_{Ts}} \rightarrow W_{Ta} = \eta_T \times W_{Ts} = \eta_T \times (h_5 - h_{6s})$$

$$W_{Ta} = 0.87 \times (3583.1 - 2115.3) = 1277 \quad kj/kg$$

Heat input to the boiler is:

$$Q_{add} = (h_4 - h_3) = (3647.6 - 160.1) = 3487.5 \quad kj/kg$$

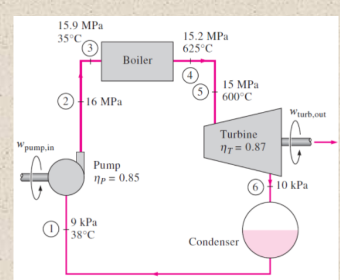
Net work output is:

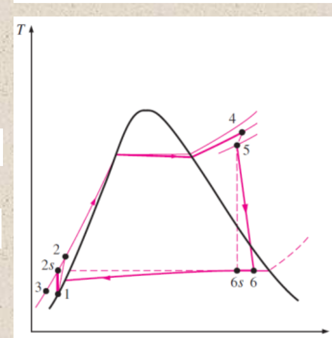
$$W_{net} = W_{Ta} - W_{pa} = 1277 - 18.98 = 1258.02 \ kj/kg$$

$$\eta_{th} = \frac{W_{net}}{Q_{add}} = \frac{1258.02}{3487.5} = 36.1 \ \%$$

The power output from the plant is:

$$power = m^{\circ} \times W_{net} = 15 \times 1258.02 = 18.9 \ MW$$





HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

- Steam power plants are responsible for the production of most electric power in the world, and even small increases in thermal efficiency can mean large savings from the fuel requirements. Therefore, every effort is made to improve the efficiency of the cycle on which steam power plants operate.

- 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.*

- That is, the average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection.

Next we discuss three ways of accomplishing this for the simple ideal Rankine cycle.

LOWERING THE CONDENSER PRESSURE (LOWERS $T_{LOW,AVG}$)

Steam exists as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser.

Therefore, lowering the operating pressure of the condenser automatically lowers the temperature of the steam, and thus the temperature at which heat is rejected.

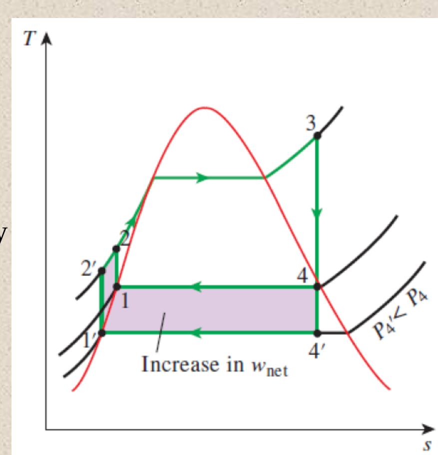


FIGURE 6
The effect of lowering the condenser pressure on the ideal Rankine cycle.

SUPERHEATING THE STEAM TO HIGH TEMPERATURES (INCREASES $T_{HIGH,AVG}$)

- The average temperature at which heat is transferred to steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of vapor power cycles is illustrated on a *T-s* diagram in Fig. 7.

- The colored area on this diagram represents the increase in the network. The total area under the process curve 3-3′ represents the increase in the heat input. Thus both the network and heat input increase as a result of superheating the steam to a higher temperature.
- The overall effect is an increase in thermal efficiency, however, since the average temperature at which heat is added increases.

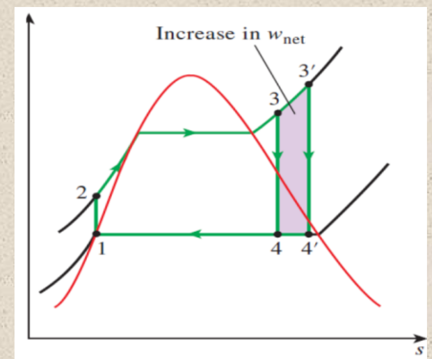
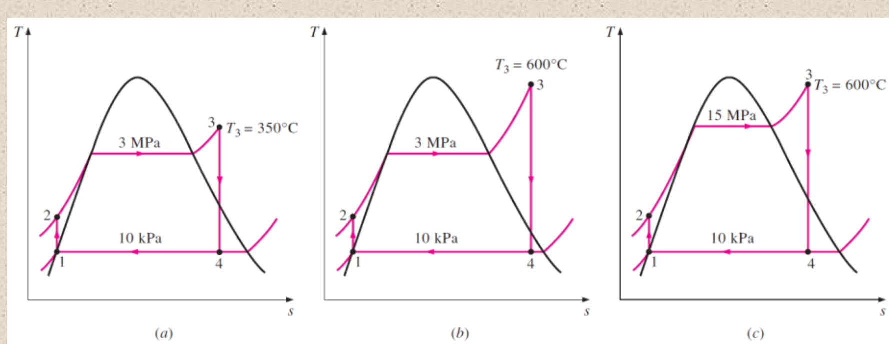


FIGURE 7
The effect of superheating the steam to higher temperatures on the ideal Rankine cycle.

Example 2

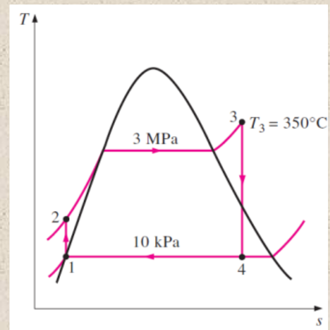
Consider a steam power plant operating on an ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350°C and is condensed in the condenser at a pressure of 10 kPa. Determine: (a) the thermal efficiency of this power plant (b) the thermal efficiency if steam is superheated to 600°C instead of 350°C (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. From saturated steam tables:

State 1: at $P_1 = 10$ kPa $\rightarrow h_1 = h_f = 191.81$ kJ/kg,
$v_1 = v_f = 0.00101$ m^3/kg



(a)

State 2: $P_2 = 3$ MPa

$$w_{pump} = v_1(P_2 - P_1) = 0.00101 \times (3 \times 10^3 - 10)$$
$$= 3.02 \text{ kJ/kg}$$

$$w_{pump} = h_2 - h_1 \rightarrow h_2 = h_1 + w_{pump} = 191.81 + 3.02$$
$$= 194.83 \text{ kJ/kg}$$

State 3: at $P_3 = 3$ MPa and $T_3 = 350$°C from superheated steam tables:

$h_3 = 3116.1$ kJ/kg and $s_3 = 6.745$ kJ/kg. K

State 4: at $P_4 = 10$ kPa and $s_4 = s_3 = 6.745$ kJ/kg. K

From saturated steam tables: $s_f = 0.6492\ kJ/kg.K$, $s_{fg} = 7.4996$ kJ/kg. K

$h_f = 191.81$ kJ/kg, $h_{fg} = 2392.1$ kJ/kg

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.745 - 0.6492}{7.4996} = 0.8128$$

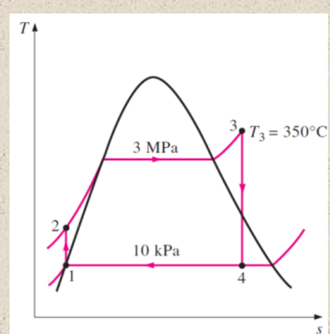
$$h_4 = h_f + x_4 . h_{fg} = 191.81 + 0.8128 \times 2392.1 = 2136.1 \text{ kJ/kg}$$

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

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

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1944.3}{2921.3}$$

$\boldsymbol{\eta th = 33.4\%}$ **Ans.**



(a)

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:

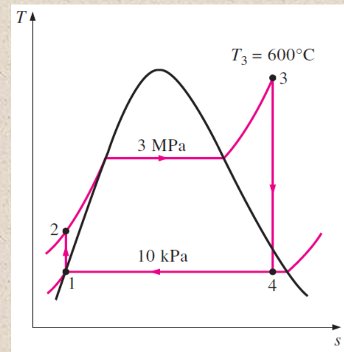
$h3 = 3682.8$ kJ/kg
$h4 = 2380.3$ kJ/kg ($x4 = 0.915$)

$qin = h_3 - h_2 = 3682.8 - 194.83 = 3488$ kJ/kg

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

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{2188.5}{3488}$$

$\boldsymbol{\eta th} = \mathbf{37.3\%}$ **Ans.**



(b)

- Therefore, the thermal efficiency increases from **33.4%** to **37.3%** as a result of superheating the steam from 350°C to 600°C.
- At the same time, the quality of the steam (dryness fraction) increases from **0.813** to **0.915**.

c. State 1 remains the same in this case, but the other states change. The enthalpies at state 2 (15 MPa and $s2 = s1$), state 3 (15 MPa and 600°C) and state 4 (10 kPa and $s4 = s3$) are determined in a similar manner to be:

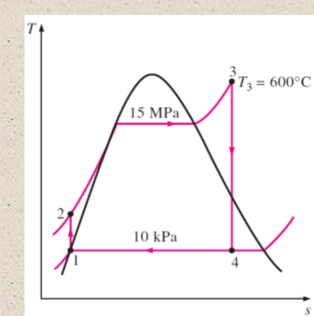
$h_2 = 206.95$ kJ/kg, $h_3 = 3583.1$ kJ/kg and
$h_4 = 2115.3$ kJ/kg ($x_4 = 0.804$)

$q_{in} = h3 - h2 = 3583.1 - 206.95 = 3376.2$ kJ/kg

$qout = h4 - h1 = 2115.3 - 191.81 = 1923.5$ kJ/kg

$$\eta th = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1923.5}{3376.2}$$

$\boldsymbol{\eta th} = \mathbf{43\%}$ **Ans.**



(c)

The thermal efficiency increases from 37.3% to 43% as a result of raising the boiler pressure from 3 MPa to 15 MPa while maintaining the turbine inlet temperature at 600°C.
At the same time, the quality of the steam decreases from 0.915 to 0.804 (in other words, the moisture content increases from 0.085 to 0.196).

Any Questions???