ESO201A

Lecture#37

(Class Lecture)

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By

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Vapour Power Cycles

We have learned that the Carnot cycle is the most efficient cycle operating between T_H and T_L. Thus it is natural to examine the possibility of using the Carnot cycle for vapour power plants. We will take water as the working fluid in our discussions as it is the most used substance in vapour power cycles.

Consider a steady-flow Carnot cycle executed within the liquid-vapour mixture dome of a pure substance, as shown in Fig. 1(a).

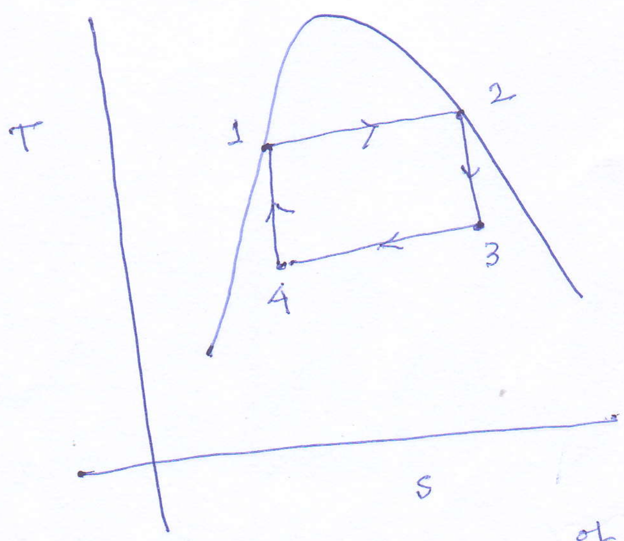

Fig. 1(a) T-s diagram of a Carnot vapour cycle

Process 1-2 : Reversible and isothermal heating of the fluid in a boiler.

Process 2-3 : Isentropic (reversible adiabatic) expansion in a turbine.

Process 3-4 : Reversible and isothermal condensation in a ~~turbi~~ condenser.

Process 4-1 : Isentropic compression by a compressor to the inlet state.

Several impracalities are associated with this cycle. They are as follows.

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 automaticalley 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 (example: T_{max} must be less than T_{cr} whose value for water is 374°c). Limiting the maximum temperature in the cycle also limits the thermal efficiency.

2. The isentropic expansion process (2-3) can be approximated closely by a well-designed turbine. However, the quality of the steam decreases during the process, as shown on the T-S diagram in Fig. 1(a). Thus the turbine has to handle a liquid-vapour mixture. The impingement of liquid droplets on the turbine blades causes erosion and is a major source of wear. Thus steam with $x < 0.9$ cannot be tolerated in the operation of power plants.

3. The isentropic compression process (Process 4-1) involves the compression of a liquid-vapour 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.

Some of these problems could be eliminated by executing the carnot cycle in a different way, as shown in Fig. 1 (b).

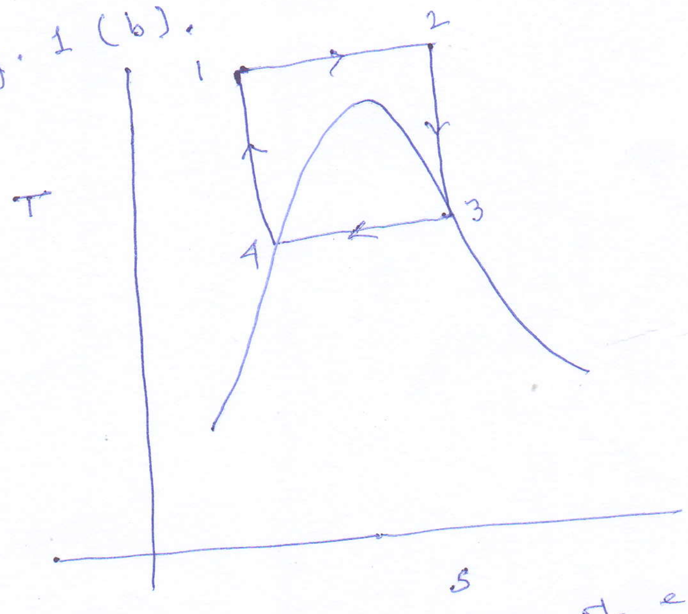

Fig. 1 (b) Alternative way of executing carnot cycle for vapour power plants

This cycle, however, presents other problems such as isentropic compression to extremely high pressures and isothermal heat transfer at variable pressures. Thus we conclude that the carnot cycle cannot be approximated in actual devices and is not a realistic model for vapour power cycles.

Rankine cycle : The Ideal Cycle for Vapour 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. A schematic diagram of a steam power plant is shown in Fig. 2(a) and the T-s diagram of the cycle is depicted in Fig. 2(b).

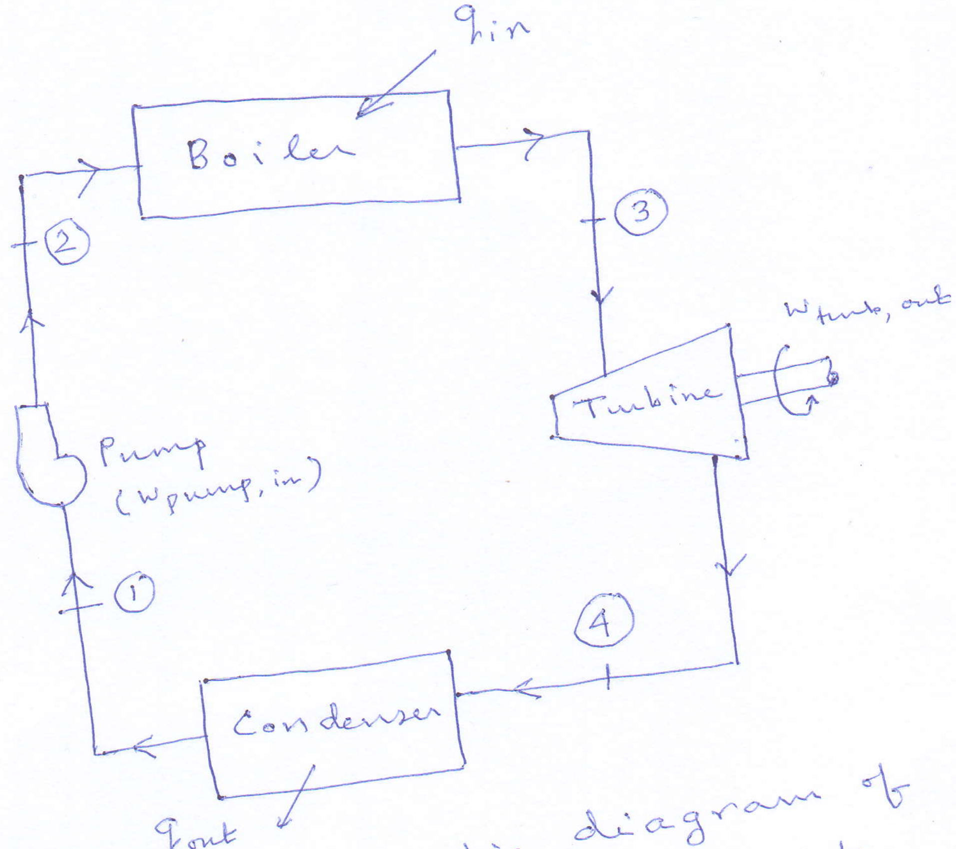

Fig. 2(a) Schematic diagram of a steam power plant

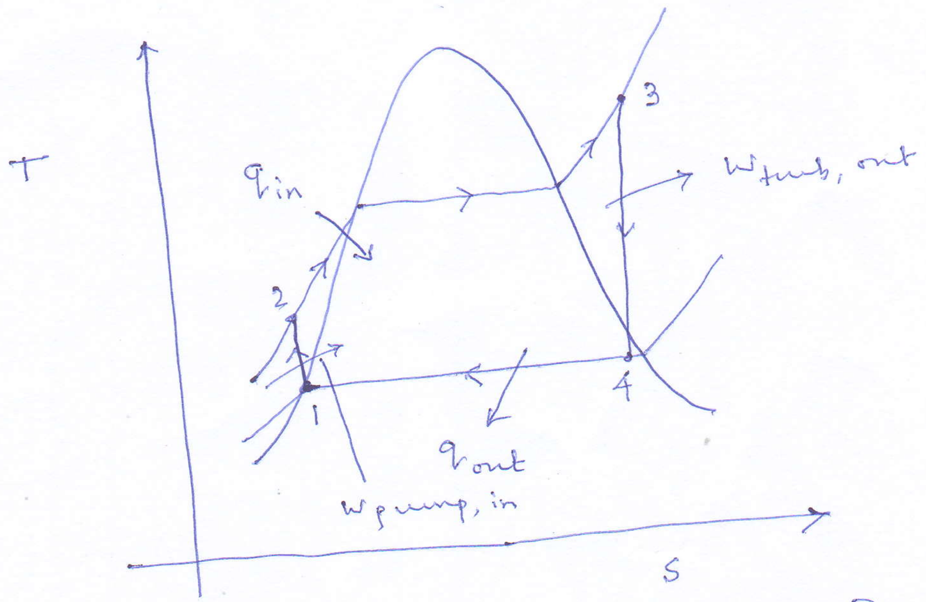

Fig. 2(b) The simple ideal Rankine cycle

All processes of the ideal Rankine cycle are internally reversible. The cycle consists of the following four processes.

1-2 : Isentropic compression in a pump

2-3 : Constant pressure heat addition to a boiler

3-4 : Isentropic expansion in a turbine

4-1 : Constant pressure heat rejection from a condenser

Water enters the pump at state 1 as saturated liquid and is compressed isentropically to the operating pressure of the boiler. The water temperature increases a little during this isentropic compression process due to a slight decrease in the specific volume of water. Note that if water was truly incompressible there would have been no temperature change at all during this process.

Water enters the boiler as a compressed liquid at state 2 and leaves as a superheated vapour at state 3.

The superheated vapour at state 3 enters the turbine, where it expands isentropically and produces work by rotating the shaft connected to an electric generator. The pressure and the temperature of steam drop during this process to the values at state 4, where the steam enters the condenser. At this state, steam is usually a saturated liquid-vapour mixture with a high quality.

Steam is condensed at constant pressure in the condenser, by rejecting heat to a cooling medium such as cooling water drawn from a lake or a river. Steam leaves the condenser as saturated liquid and enters the pump, completing the cycle.

The area under the process line 2-3 represents the magnitude of the heat transferred to the water in the boiler and the area under the process line 4-1 represents the magnitude of the heat rejected to the condenser (Fig. 2 (b)). The difference between the two (the area enclosed by the cycle) is the net work produced by the cycle.

Energy Analysis of the Ideal Rankine Cycle

All four components associated with the Rankine cycle (the pump, boiler, turbine and compressor) are steady-flow devices, and thus all four processes that constitute the Rankine cycle can be analyzed as steady-flow processes. The kinetic and potential energy changes are usually small as compared to the change in enthalpy across each component, and are therefore usually neglected.

Then the steady-flow energy equation per unit mass of steam reduces to

$$(q_{in} - q_{out}) - (w_{out} - w_{in}) = h_e - h_i \quad (1)$$

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

Pump ($q = 0$)

$$\underbrace{(q_{in} - \cancel{q_{out}}^{0})}_{q} - \underbrace{(\cancel{w_{out}}^{0} - w_{in})}_{w} = h_e - h_i$$

$$\Rightarrow w_{in} = h_2 - h_1 \quad (2)$$

$$\Rightarrow \boxed{w_{pump, in} = h_2 - h_1}$$

where $h_1 = h_{f@P_1}$

Recall $dh = \cancel{du}^{0} + v\,dp + \cancel{p\,dv}^{0}$.
Since $\Delta T = 0$ for pumping process, $du = 0$. Hence, $dh = v\,dp$.

$$\text{or} \quad \boxed{w_{pump,in} = v(P_2 - P_1)} \qquad (3)$$

where $v \equiv v_1 = v_{f@P_1}$

Boiler ($w = 0$)

$$\underbrace{(q_{in} - \cancel{q_{out}}^{0})}_{q} - \underbrace{(\cancel{w_{out} - w_{in}}^{0})}_{w}$$

$$= h_e - h_i$$

$$(4)$$

$$\Rightarrow \boxed{q_{in} = h_3 - h_2}$$

Turbine ($q = 0$)

$$(\cancel{q_{in} - q_{out}}^{0}) - (w_{out} - \cancel{w_{in}}^{0})$$

$$= h_e - h_i$$

$$\Rightarrow w_{out} = h_i - h_e \qquad (5)$$

$$\Rightarrow \boxed{w_{turb,out} = h_3 - h_4}$$

Condenser ($w = 0$)

$$(\cancel{q_{in}}^{0} - q_{out}) - (\cancel{w_{out} - w_{in}})^{0} = h_e - h_i$$

$$\Rightarrow \quad q_{out} = h_i - h_e \qquad (6)$$

$$\Rightarrow \quad \boxed{q_{out} = h_4 - h_1}$$

The thermal efficiency of the Rankine cycle is:

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

$$\Rightarrow \quad \boxed{\eta_{th} = 1 - \frac{q_{out}}{q_{in}}} \qquad (7)$$

where $w_{net} = q_{in} - q_{out}$

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

The thermal efficiency can also 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 Vapour Power Cycles from Idealized Ones

The actual vapour power cycle differs from the ideal Rankine cycle, as illustrated in Fig. 3 (a), as a result of irreversibilities in various components. Fluid friction and heat loss to the surroundings are the two 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. Also, the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to pressure drops 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 high pressure than the ideal cycle calls for. This requires a larger pump and larger work input to the pump.

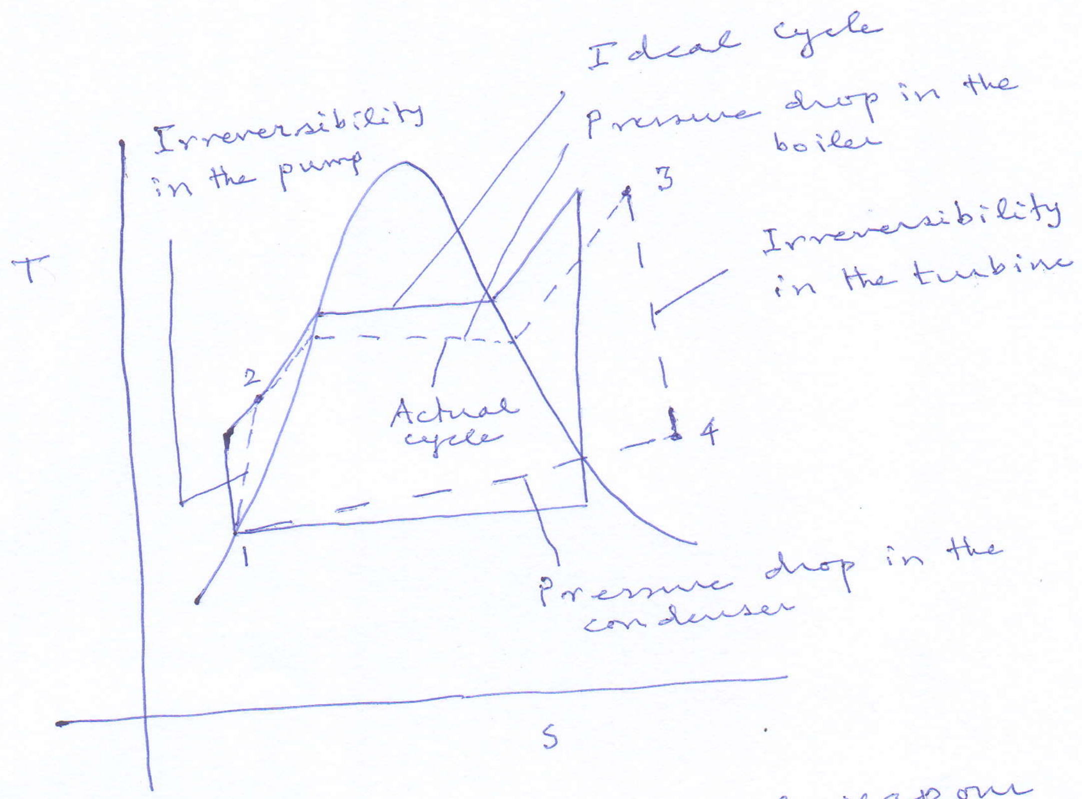

Fig. 3 (a) Deviation of actual vapour power cycle from the ideal Rankine cycle

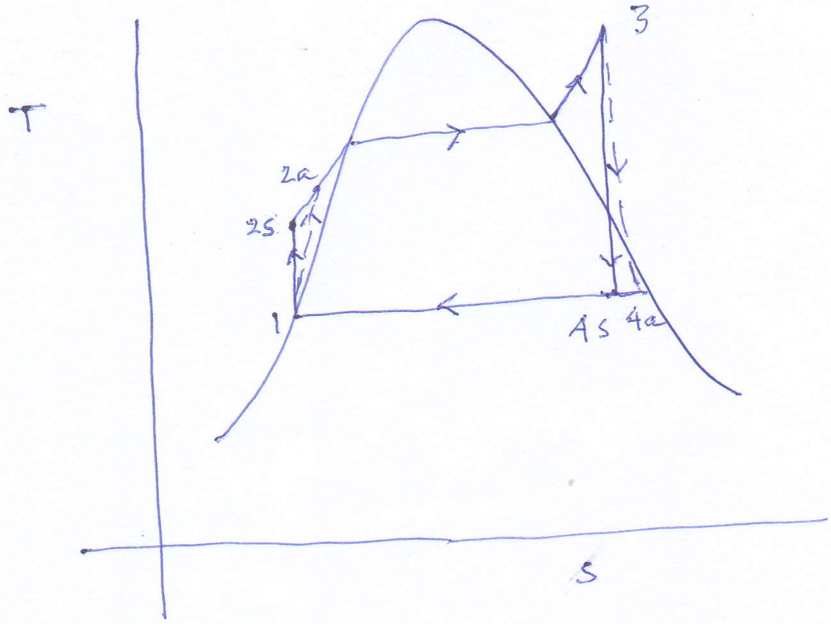

Fig. 3 (b) The effect of pump and turbine irreversibilities on the ideal Rankine cycles

The other major sources 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 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 actual condition is irreversible adiabatic. See Fig. 3(b) on p. 14. Thus, the isentropic efficiency of a pump is defined as

$$\eta_p = \frac{W_s}{W_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1} \qquad (8)$$

where 'a' stands for irreversible adiabatic and 's' stands for reversible adiabatic (isentropic)

The isentropic efficiency of a turbine is defined as

$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \qquad (9)$$

The isentropic efficiency of pumps and turbines lie between 0.7 to 0.88, with larger machines having higher efficiency.

How can we increase the efficiency of the Rankine cycle?

The basic idea behind increasing the thermal efficiency of a power cycle is the same:

Increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser.

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

The exit of the turbine is the inlet to the condenser. Steam exists as a saturated liquid-vapor mixture at the turbine exit. Thus lowering the turbine exit pressure (that is, lowering the condenser inlet pressure) automatically lowers the temperature of steam and thus the temperature at which heat is rejected. This is illustrated in Fig. 4.

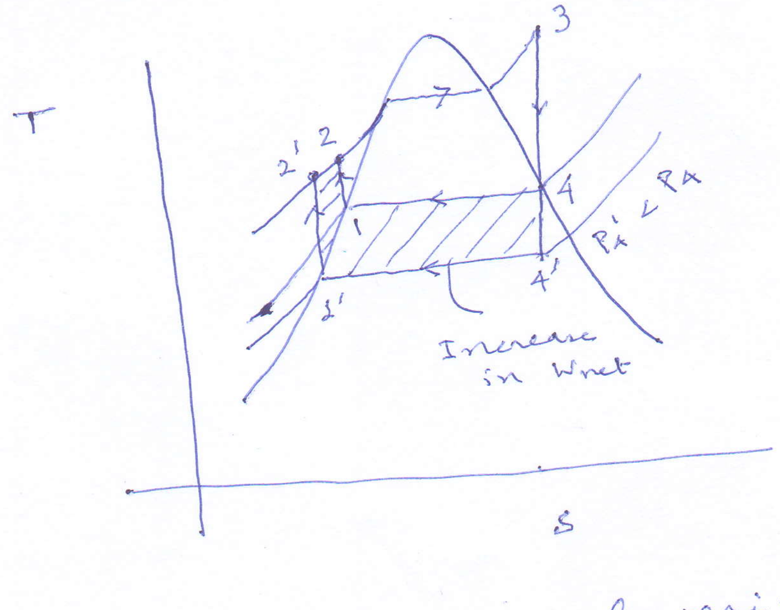

Fig. 4 The effect of lowering the condenser pressure on the ideal Rankine cycle

The hatched 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 (represented by the area under curve $2'-2$), but this increase is very small. Hence, the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle.

<u>Superheating the steam to high temperatures</u> (<u>Increases $T_{high,\,avg}$</u>)

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. This is shown in Fig. 5.

The hatched area on this diagram represents the increase in the net work.

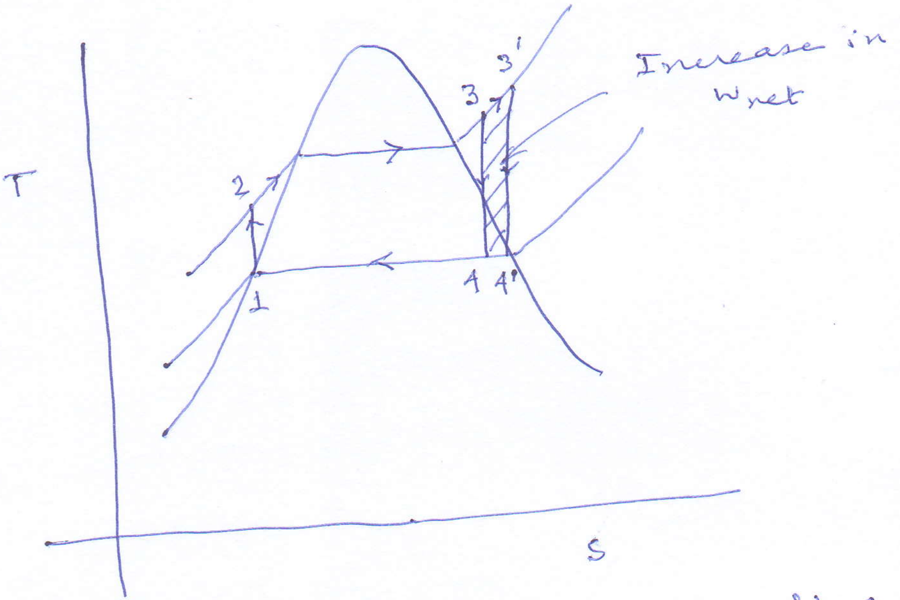

Fig. 5 The effect of superheating the steam on the ideal Rankine cycle

The total area under 3 - 3' represents the increase of 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, since the average temperature at which heat is added increases.

Increasing the boiler pressure (Increases $T_{high,avg}$)

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 transferred to the steam and thus raises the thermal efficiency of the cycle.

The effect of increasing the boiler pressure on the performance of vapour power cycles is shown in Fig. 6. Notice that for a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This undesirable side effect can be corrected, however, by reheating the steam, as discussed next.

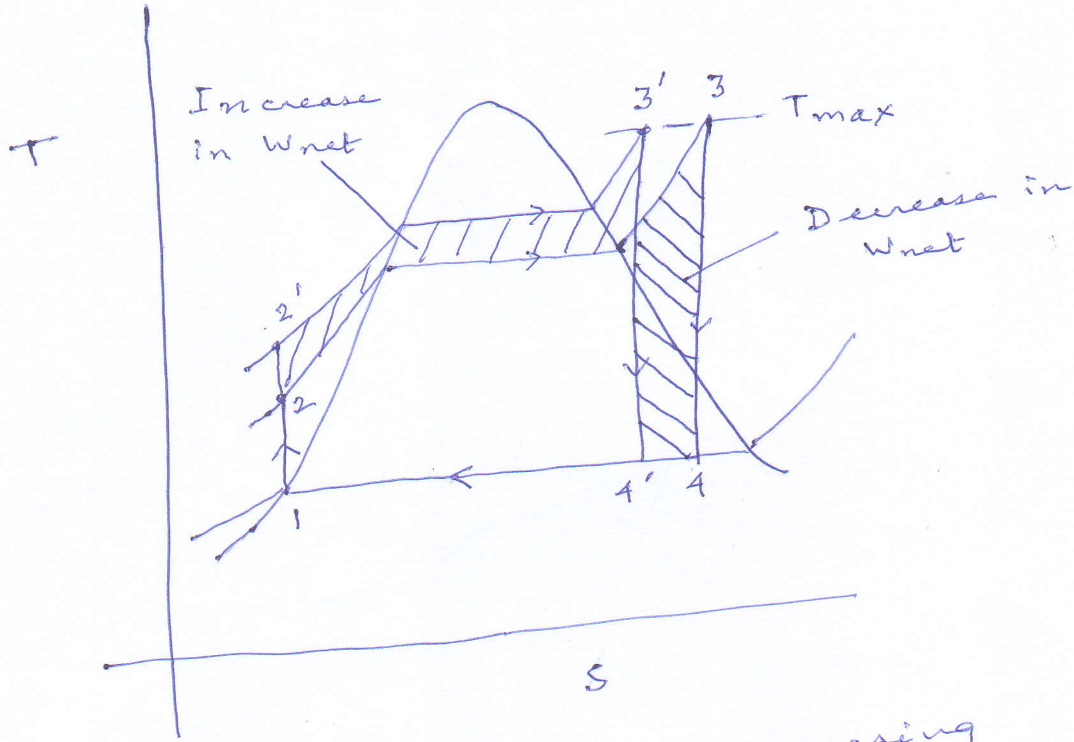

Fig. 6 The effect of increasing the boiler pressure on the ideal Rankine cycle

The Ideal Reheat Rankine Cycle

In the last section it has been shown that 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. This disadvantage can be alleviated by expanding the steam in turbine in two stages, and reheat it in between. The T-s diagram of the reheat cycle is shown in Fig. 7.

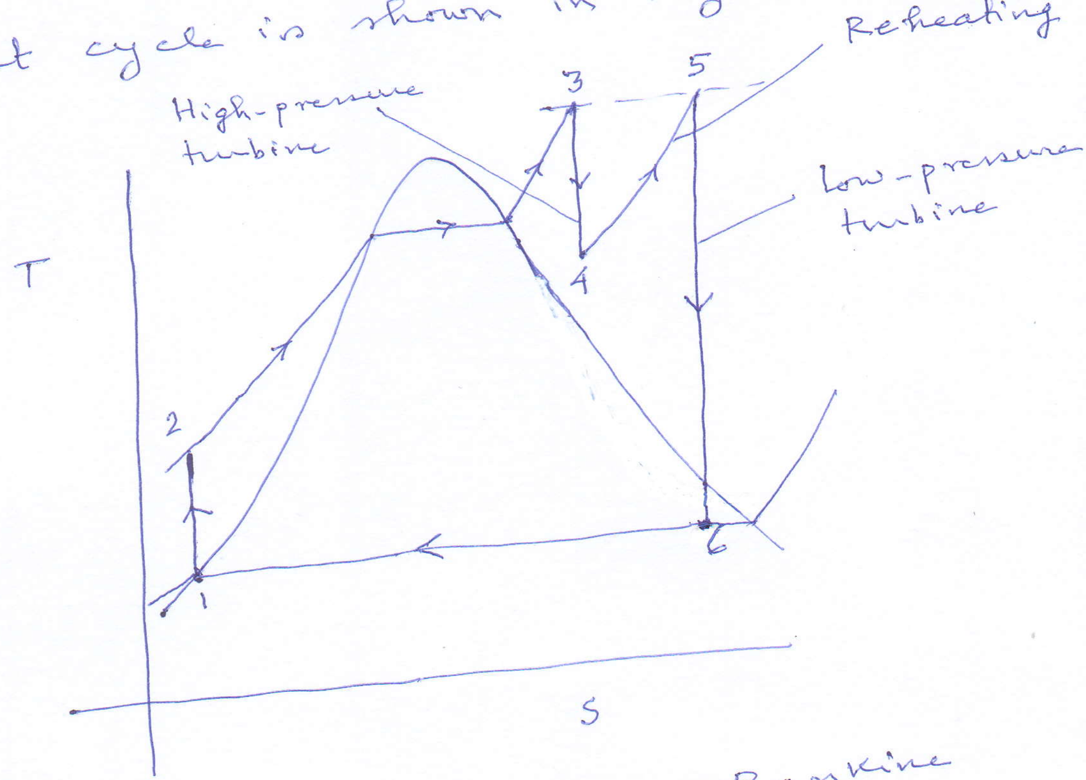

Fig. 7 The ideal reheat Rankine cycle

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. Thus the total heat input and the total turbine work output for a reheat cycle become

$$q_{in} = q_{primary} + q_{reheat}$$
$$= (h_3 - h_2) + (h_5 - h_4) \qquad (10)$$

and

$$w_{turb,out} = w_{turb,I} + w_{turb,II}$$
$$= (h_3 - h_4) + (h_5 - h_6) \qquad (11)$$

The incorpation of 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 Vapour-Compression Refrigeration cycle

Reversed carnot cycle

We know that a reversed carnot cycle is a refrigeration cycle. Figure 1 shows such a cycle.

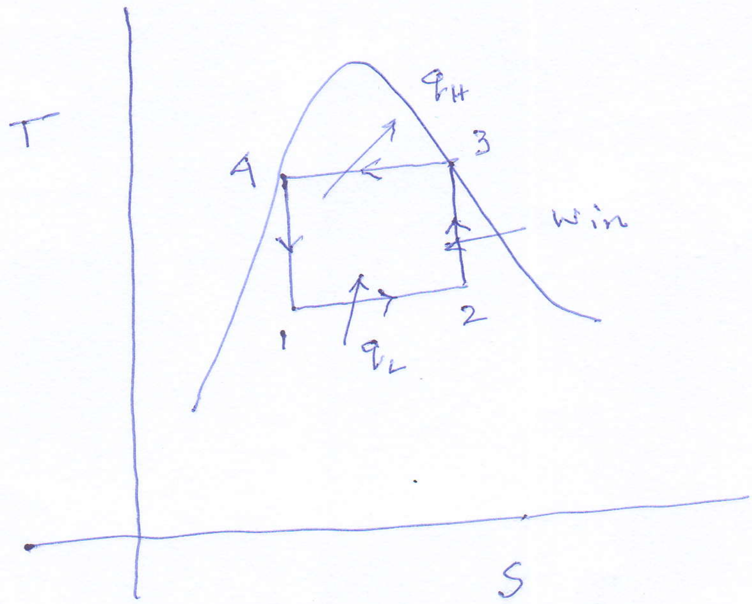

Fig. 1 T-S diagram of a Carnot refrigeration cycle

Figure 2 shows a schematic diagram of a Carnot refrigerator.

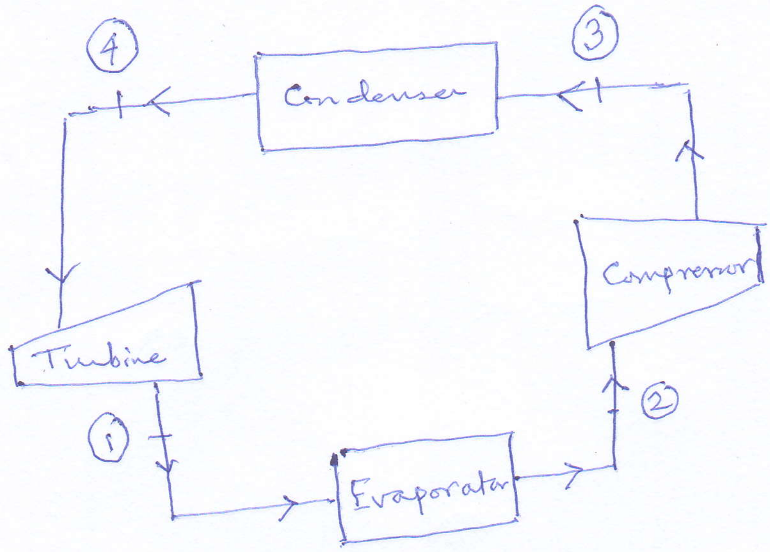

Fig. 2 Schematic diagram of a carnot refrigerator

The two isothermal heat transfer processes are not difficult to achieve in practice since maintaining a constant pressure automatically fixes the temperature of a two-phase mixture at the saturation value. Therefore, processes 1-2 and 3-4 can be approached closely in actual evaporators and condensors.

However, processes 2-3 and 4-1 cannot be approximated closely in practice. This is because process 2-3 involves the compression of a liquid-vapour mixture. It is virtually impossible to compress, at a reasonable rate, a mixture such as that represented by state 2 and still maintain equilibrium between liquid and vapour. In the isentropic expansion process 4-1 the substance will be mostly liquid. As a consequence, there will be very little work output from this process, so it is not worth the cost of including this piece of equipment in the system.

The Ideal Vapour-Compression Refrigeration Cycle

The impracticalities associated with the reversed Carnot cycle can be eliminated by vaporizing the refrigerant completely before it is compressed and by replacing the turbine with a throttling device, such as an expansion valve or a capillary tube.

Figure 3 shows a T-S diagram of such a refrigerator.

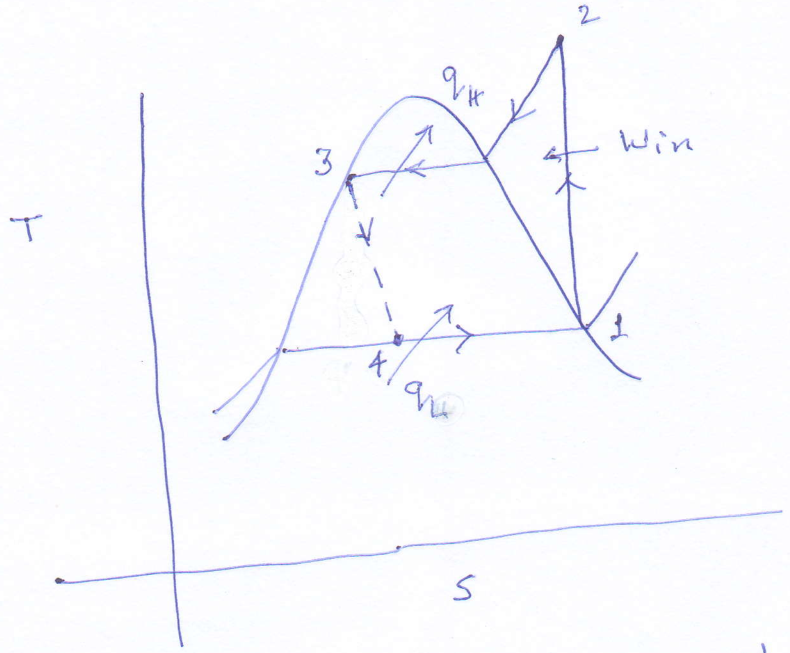

Fig. 3 T-S diagram of an ideal vapour-compression refrigeration cycle

Note that throttling is an irreversible process. A schematic diagram of an ideal vapour-compression refrigeration cycle is shown in Fig. 4.

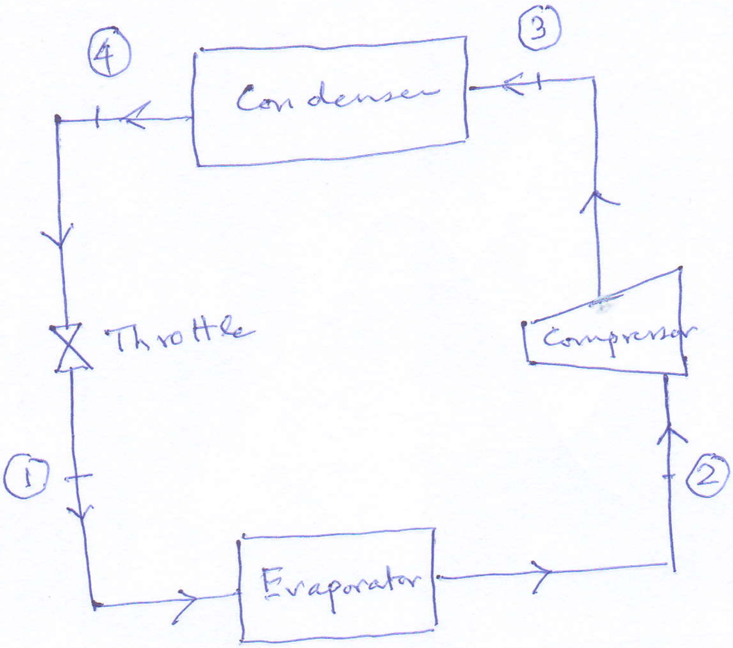

Fig. 4 Schematic diagram of an ideal vapour-compression refrigeration system

All four components associated with the vapour-compression refrigeration cycle are steady-flow devices. Neglecting Δke and Δpe as compared to Δh across each component, the energy equation on a unit mass basis is:

$$(q_{in} - q_{out}) - (w_{out} - w_{in}) = h_e - h_i \qquad (1)$$

The condenser and the evaporator do not involve any work, and the compressor can be approximated as adiabatic. The P-h diagram is shown in Fig. 5.

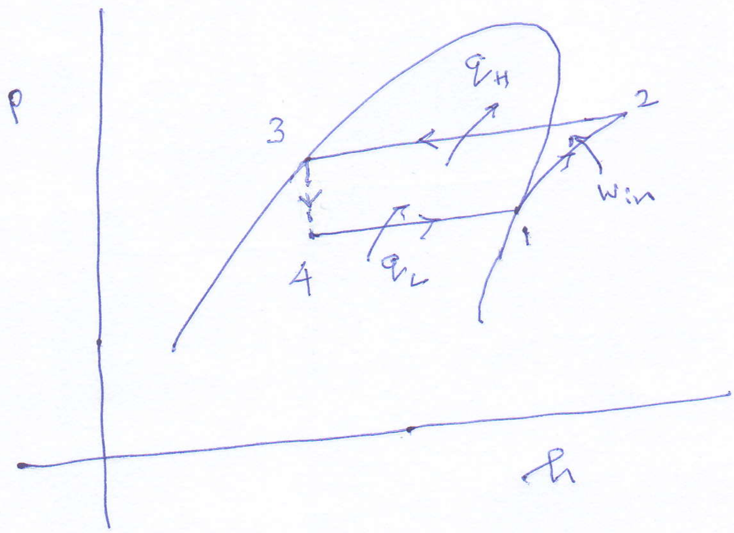

Fig. 5 The P-h diagram of an ideal vapour-compression refrigeration cycle

Energy balance

Evaporator ($w = 0$)

$$q_L = h_1 - h_4 \quad (2)$$

Condenser ($w = 0$)

$$q_H = h_2 - h_3 \quad (3)$$

Compressor ($q=0$)

$$w_{in} = h_2 - h_1 \tag{4}$$

Throttle ($w=0,\ q=0$)

$$h_3 = h_4 \tag{5}$$

Then the COPs of refrigerators and heat pumps operating on the vapour-compression refrigeration cycle can be expressed as

$$COP_R = \frac{q_L}{w_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{6}$$

$$COP_{HP} = \frac{q_H}{w_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1} \tag{7}$$

(8)

where $h_1 = h_{g@P_1}$ (9)

and $h_3 = h_{f@P_3}$.