5.0 VAPOUR POWER CYCLES

One method of producing mechanical power is the transfer of heat to a working fluid, which goes through a thermodynamic cycle, converts part of the heat to work and rejects heat to a sink. If the working fluid undergoes a phase change, the cycle is called a Vapour Power Cycle.

5.1 Characteristics of a vapour power cycle

a) The working fluid is a condensable vapour, which is in the liquid phase during part of the cycle.
b) The cycle consists of steady flow processes with each process carried out in a separate component specially designed for the purpose.

Each component constitutes an open system and all the components are connected in series so that each fluid element passes through mechanical and thermodynamic states. In each component, matter flows into and out of region of space as well as heat and work. All the processes are considered to be flow processes.

5.2 Decision making in design

- Capacity: Total power output required
- Energy Source: Chemical sources (coal, oil, natural gas, fuel cell and biomass); Nuclear energy; solar radiation etc.
- Energy Sink: From the 2nd Law, we know that all heat engines must reject some heat. Also, the efficiency is higher if the sink temperature is low. Available natural heat sinks are atmosphere, rivers, lakes, and oceans. At the moment about up to approximately 60 % of heat absorbed is rejected. This is a serious ecological problem because of thermal pollution.
- Working Fluid: Water because it is cheap and chemically stable.
- Component selection: Design of hardware. Decision on component arrangement etc. and also work out the economic and technical feasibility of the project.

5.3 Criteria for Comparing cycles

The choice of a power plant is determined largely by:

- Capital cost: This is determined by size and complexity of the plant
- Operating cost: This is determined by the overall efficiency of the plant. In general, the efficiency can be improved by increasing the complexity of the plant. A compromise between low operating and capital costs is therefore required.

5.4 Efficiency

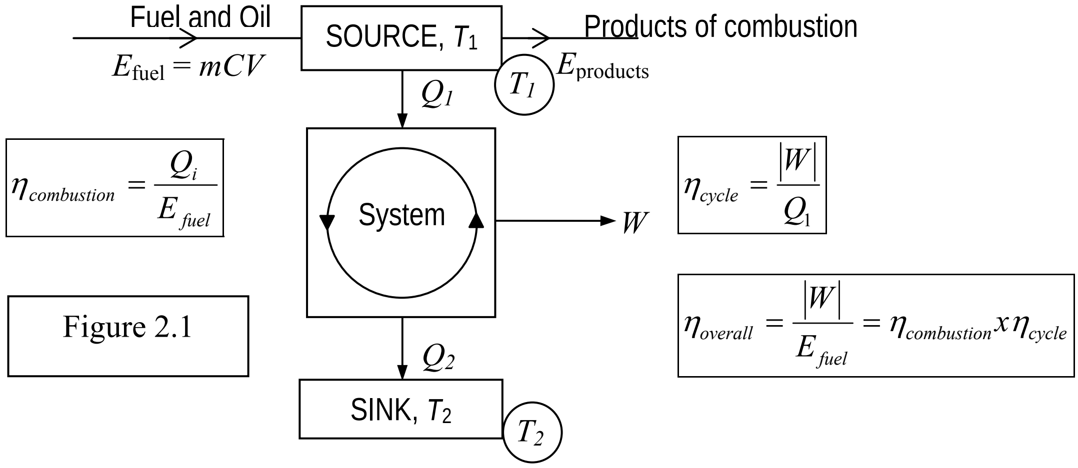


Figure 2.1

The essential features of a vapour power cycle are as illustrated in Fig. 5.1. The overall efficiency of a vapour power plant is suitably measured by the proportion of latent heat in the fuel, which is converted into useful mechanical work. The overall thermal efficiency can be expressed as a product of two efficiencies:

i) **Combustion efficiency**: This expresses the proportion of the internal energy or latent energy in the fuel transferred as heat to the working fluid.

ii) **Cycle efficiency**: This expresses the proportion of heat energy transferred to the working fluid and which is subsequently converted into mechanical work.

The cycle efficiency is **not unity** for the following reasons:

a) The 2nd law expresses the fact that even in the best power cycle (Carnot cycle), some form of heat must be rejected. The best power cycle is that in which all heat supplied is transferred while the working fluid is at a constant upper temperature T_1 and all the heat rejected leaves while the working fluid is at a constant lower temperature T_2, and all the processes are reversible. The efficiency of such a cycle is given by the expression $\frac{T_1 - T_2}{T_1}$ irrespective of the working fluid.

b) From the practical point of view, all real processes are irreversible and irreversibility in cycles reduces the overall efficiency. The first law is still obeyed, i.e. but the cycle efficiency $|W|/|Q_1|$ is reduced.

Ideal and actual cycle efficiencies and efficiency ratio:

If all the processes of a power cycle are assumed to be reversible, then the efficiency calculated is known as the ideal cycle efficiency. The ratio of the actual cycle efficiency to the ideal cycle efficiency is called the efficiency ratio.

$$\eta_{ratio} = \frac{\eta_{actual}}{\eta_{ideal}}$$

5.5 Work ratio

	Ideal		**Actual $\eta_c = \eta_T = 0.90$**	
	Cycle 1	**Cycle 2**	**Cycle 1**	**Cycle 2**
Q_{in}	120	120	120	120
W_T	100	40	90	36
W_c	61	1	67.9	1.1
W_{NET}	39	39	22.2	34.9
r_w	0.329	0.975	0.249	0.969
η	0.325	0.325	0.185	0.291

Some cycles are more sensitive to Irreversibilities than others and so high ideal cycle efficiency only is not by itself a good indicator of whether or not the cycle will provide a power plant of high overall efficiency.

The work ratio, r_w, gives an indication of how sensitive a cycle is to any irreversibility introduced into it. It defined as the ratio of the *net work output* to the gross *work output.* That is,

$$\boxed{r_w = \frac{\text{Net work output}}{\text{Gross work output}}}$$

Irreversibilities have the effect of decreasing the work outputs and increasing the work inputs and hence there is a decrease in the network output. Summarising, we may say that a *high ideal cycle efficiency together with a high work ratio provide a reliable indication that the actual power plant will have a good overall efficiency.* A work ratio of unity means that the components producing work will be of least possible size for a given net power output. Work ratio is in itself not very informative.

5.6 Specific steam consumption

A more direct indication of the relative sizes of steam power plant is provided by the specific steam consumption (SSC). It is the mass flow of steam required per unit net power output. It is usually expresses in kg/kWh and if $|W|$ is the magnitude of the network output per unit mass of steam in kJ/kg,

Then, $$\frac{1}{|W|} = \frac{3600}{W} kg/kWh, \text{ if } |W| \text{ is in kJ/kg}$$

5.7 Process efficiencies

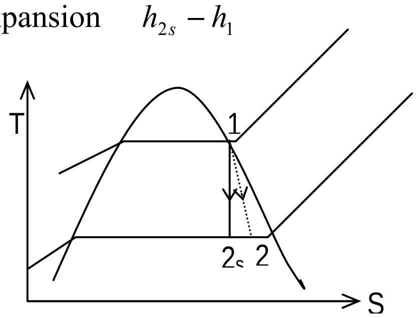
It is a measure of irreversibility and can be defined for a steady flow process as the ratio of isentropic to actual work or vice versa, depending on whether the process is a work producing process, or a work requiring process,

Turbine process:

$$\eta_{turbinr,isentropic} = \frac{\text{work done during actual expansion}}{\text{work done during isentropic expansion}} = \frac{h_2 - h_1}{h_{2s} - h_1}$$

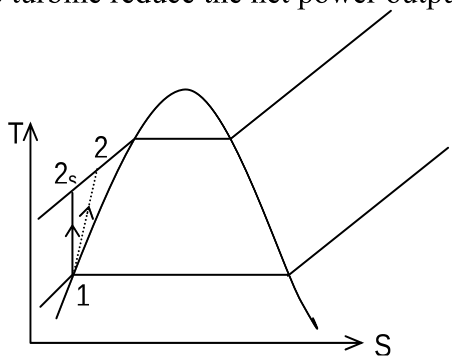
$$\text{Work product} = \frac{\text{Actual}}{\text{Isentropic}}$$

$$\text{Work requiring} = \frac{\text{Isentropic}}{\text{Actual}}$$



Irreversibility within the turbine reduce the net power output of the plant

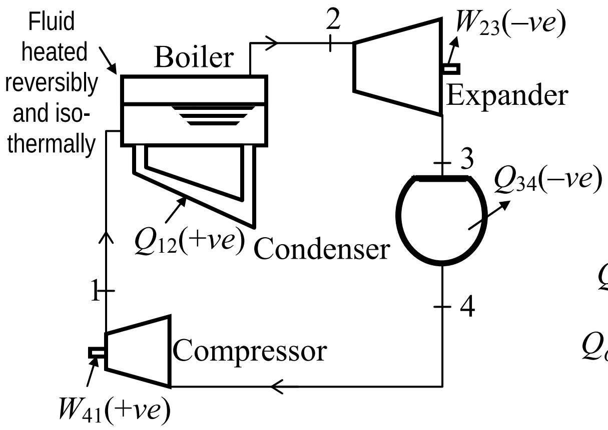
Compressor or pump process



$$\eta_{compressor,isentropic} = \frac{\text{work required for isentropic compression}}{\text{work required for actual compression}} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

5.8 CARNOT CYCLE

It consists of two reversible isothermal processes at T_1 and T_2 respectively, connected by two reversible adiabatic (isentropic) processes. When the working fluid is a condensable vapour the isothermal processes are obtained by heating and cooling at constant pressure while the fluid is a wet vapour.



$$Q_{in} = h_{fg}eT_1$$

$$Q_{OUT} = \frac{T_2}{T_1}Q_{IN}$$

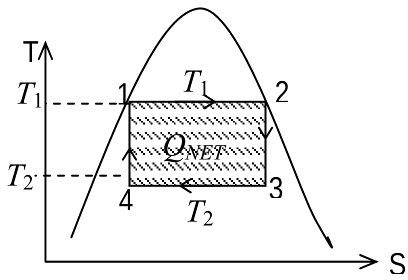


Fig. 5.2(a) Flow diagram of the Carnot Cycle *Fig. 5.2(b) T -s diagram of Carnot Cycle*

Notes For a Cycle, from 1st Law

$$Q_{NET} + W_{NET} = 0$$
$$Q_{IN} + Q_{12} = T_1(S_2 - S_1)$$
$$Q_{OUT} + Q_{34} = T_2(S_3 - S_4)$$
$$Q_{NET} = (T_1 - T_2)(S_3 - S_4)$$

$$\boxed{\eta_{cycle} = |W_{NET}| \big/ Q_{IN}}$$

1-2: Saturated liquid in state 1 is evaporated in a boiler at constant pressure to form saturated steam in state 2.
Using the steady-state open flow energy equation
$Q_{12} + W_{12} = h_2 - h_1 + \Delta(KE) + \Delta(PE)$, but $\Delta(KE) = 0$ and $\Delta(PE) = 0$

Hence,

$$\boxed{Q_{12} = h_2 - h_1}$$

$h_2 - h_g$ at the pressure of the boiler and
$h_1 - h_f$ at the pressure of the boiler

2-3: Steam is expanded isentropically to state 3 while doing work in a turbine ($s_2 = s_3$) Note s_2=s_g and it evaluated at the pressure of the boiler. Since the entropy at state 3 lies between the s_f and s_g values corresponding to the boiler pressure, s_3 is evaluated as follows:

$$s_3 = s_{f,3} + x_3 s_{fg,3}$$

where, x_3 is the dryness fraction of the working fluid at state 3. $s_{f,3}$ and $s_{f,g,3}$ values are read at the pressure corresponding to the condenser pressure at state point 3.

Using the steady-state open flow energy equation, we obtain

$$\boxed{W_{23} = h_3 - h_2}$$

Quality of steam decreases during this process thus, turbine handles steam with low quality (i.e. steam with high moisture content). The impingement of liquid droplets on the turbine blades causes erosion and is a major source of wear. x ≥ 90% is desirable.

$$x = \frac{m_g}{m_t}: \qquad m_t = m_f + m_g \qquad V = V_f + V_g = m_f v_f + m_g v_g$$

$$m_t v_{av} = m_f v_f + m_g v_g \qquad \boxed{v_{or} = (1-x)v_f + xv_g}$$

3-4: After expansion the wet steam is then partially condensed at constant reassure and constant temperature while heat is rejected.

$$Q_{3,4} = h_4 - h_3 \qquad s_4 = s_{f4} + x_4 s_{fg,4}$$

Note: $s_1 = s_4$ and s_{f4}, $s_{fg,4}$ are read at the pressure corresponding to the condenser at state point 4. Condensation is stopped at state 4.

4-1: Wet steam is compressed isentropically in a rotary or reciprocating compressor to state 1 that is the boiler pressure and temperature, the work required being

$$W_{41} = h_1 - h_4$$

$s_1 = s_f$ and is read at the pressure corresponding to that of the boiler pressure.

Thermal efficiency: $\eta_{th,c} = -\frac{W_{NET}}{Q_{IN}} = -\frac{W_{23} + W_{41}}{Q_{12}} = \frac{Q_{12} + Q_{34}}{Q_{12}} = 1 - \frac{T_2}{T_1}$

Note: The Carnot Cycle is **not** a realistic model for steam power plants because

1. Limiting the heat transfer processes to two-phase systems to maintain isothermal conditions severely limits the maximum temperature that can be used in the cycle.
2. The turbine would have to handle steam with a high moisture content which causes erosion of turbine blades and
3. It is not practical to design a compressor that will handle two-phase fluid.

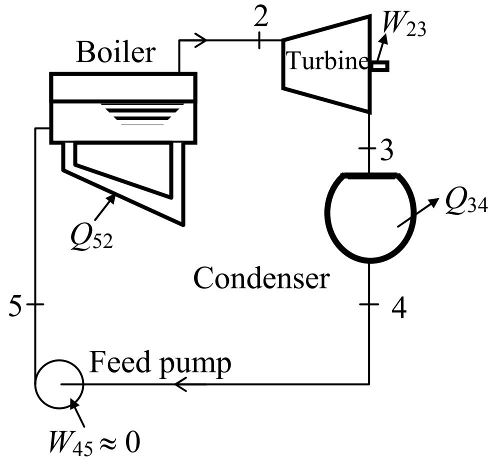
5.9 The Rankine cycle [Basic Power Cycle]

Although the Carnot cycle is the most efficient cycle, its work ratio is low. Further, there are practical difficulties in following it. Consider the Carnot cycle of Fig. 5.2: at state 4 the steam is wet at T_2 but it is difficult to stop the condensation process at the point 4 and then compress it just to state 1. It is difficult to compress wet mixtures since the liquid tends to separate out from the vapour and the compressor would have to deal with a non-homogeneous mixture. It is convenient to allow the condensation process to proceed to completion, as in Fig. 5.3. The working fluid is water at the new state point 4 in Fig. 5.3, and this can be conveniently pumped to boiler pressure as shown at state point 5. The pump has much smaller dimensions than it would have if it had to pump a wet vapour, the compression process is carried out more efficiently, and the equipment required is simpler and less expensive.

With the new cycle, we realise that at state point 5; the water is not at the saturation temperature corresponding to the boiler pressure. Thus, heat must be supplied to change the state from water at 5 to saturated water at 1; this is constant pressure process, but it is not at constant temperature. If we let T_{eff} be the effective that temperature at which heat is added or supplied at some constant pressure and temperature, then its value can be evaluated from the relation $Q_{52} = T_{eff}\,\Delta S$. The effective temperature at which is added is lower than the corresponding Carnot temperature. Hence, the efficiency of the modified cycle is **not** as high as that of the Carnot cycle. But the net work output in the modified cycle is greater than that of the Carnot cycle. It follows that the *SSC* is less and the work ratio is greater.

$$r_w = \left(\frac{W_{net}}{W_{turbine}} \right)$$

This resulting ideal cycle, which is more suitable as a criterion for steam cycles rather than the Carnot cycle, is called the *Rankine cycle.*



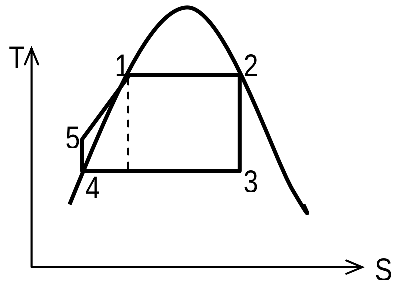


Fig 5.3 The Rankine cycle without superheat *Fig. 5.4 T-s diagram of the Rankine cycle*

<u>Feed pump work:</u>

For a reversible process: $dQ = du - dW$ (1[st] Law) (1)

But $dW = -pdv$ and $h = u + pv$ (2)

$dh = du + pdv + vdp$ (3)

Substituting equations (2) and (3) into (1) we arrive at

$$dQ = dh - vdp$$

For isentropic process $dQ = 0$, hence $dh = vdp$

Thus $\boxed{W_{45} = h_5 - h_4 = v_{f4}(P_5 - P_4)}$ (4)

where v_{f4} can be taken from tables for water at the pressure P_4=P_3 (condenser pressure)

In general for a steady state, for process 1-2, $\dot{m}(s_2 - s_1) - \frac{d\dot{Q}}{T} \geq 0$. But for an adiabatic process $dQ = 0$, hence $s_2 \geq s_1$. Thus, for an irreversible process $s_2 > s_1$ and the irreversibility in process in process 1-2 is accounted for by introducing the process efficiency. The actual expansion and compression processes are irreversible and are indicated by lines 1-2 and 3-4 in Fig. 2-5.

Turbine isentropic efficiency $= \frac{h_2 - h_1}{h_{2s} - h_1}$ and

Compression isentropic efficiency $= \dfrac{h_{4S} - h_3}{h_4 - h_3}$

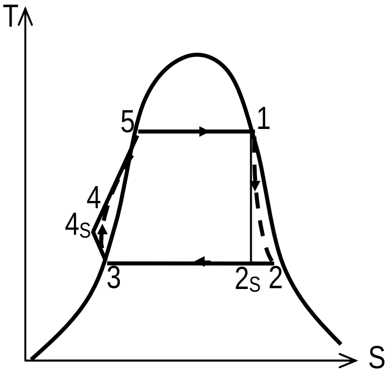


Fig. 5.5 Rankine cycle showing real Processes on a *T* -*s* diagram

Condenser heat load:

The rate of heat removal from the condenser, per unit power output is given by the product of *SSC* and the heat removed in the condenser by the cooling water per unit mass of steam.

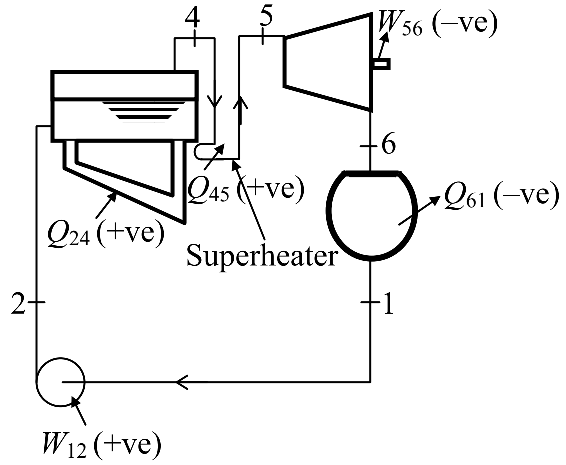
Condenser heat load per kW of power output = $SSC\ x\ \left(h_2 - h_3\right)$

5.10 The Rankine cycle with superheat (The Simple Rankine Cycle)

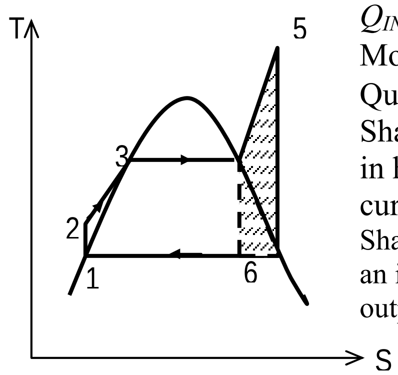
a) The metallurgical limit of the boiler materials is **not** approached when the steam leaves the boiler in a saturated condition ($T_c \approx 374$°C but $T_{metallurgical} \approx 620$ °C). The quality of steam at turbine exit in the simple Rankine cycle is too low. The isentropic efficiency of turbines is affected by the wetness of steam. (Note that wet steam corrodes the turbine blades). In practice, we require a minimum quality of 0.9 at the turbine exit.

But by placing in the combustion gases a separate batik of tubes (the superheater) leading saturated steam away from the boiler, it is possible to raise the steam temperature without at the time raising the boiler pressure. The resulting cycle with this modification is a *Rankine cycle with superheat,* as in Fig. 5.7.

The Ideal Rankine Cycle (Cycle employed in practical vapour power cycles)



Thus both net work output and heat input increases. Overall effect is increase in η cycle.



$Q_{IN}\uparrow$, $Q_{NET}\uparrow \Rightarrow |W_{NET}|\uparrow$
Moisture content $\downarrow$ ie.
Quality$\uparrow$ $\eta\uparrow$ $W_T\uparrow$ $W_P \rightarrow$
Shaded represent increase in heat input under process curve 4-5
Shaded area also represents an increase in net work output

Fig. 5.7a The Rankine cycle with superheat *Fig. 5.7b T-s diagram of Rankine cycle*

The superheater is a heat exchanger in which heat is transferred to the saturated steam to increase its temperature. It may have its own source of heat or share with the boiler. If it is to share with the boiler the bank is situated such that hot gases from the furnace heat it until the steam reaches the required temperature. Normally, they have smaller bores than the actual boiler tubes.

Comments:

i) The effective temperature at which heat is added externally is increased. Hence, the efficiency of cycle increases. Unlike the efficiency of a simple Rankine cycle, the efficiency of the Rankine cycle with superheat increases continuously with pressure.

ii) The work ratio does **not** change since the work ratio in unsuperheated Rankine cycle is very near unity.

iii) The specific steam consumption is markedly reduced, the net work per unit mass of steam being much greater, so that the added complexity of the superheater is compensated by a reduction in the size of the other components.

iv) The dryness of the steam at the last stages of the steam turbine is increased, however the required quality of steam at the turbine exit is not attained.

5.11 How do 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.*

Three ways of increasing the efficiency of the Rankine Cycle above are discussed below:

(a) Lowering the condenser Pressure (Lowers $T_{low,\ av}$)

Steam exists as saturated mixture in the condenser at the saturation pressure corresponding to the pressure inside the condenser. Therefore, lowering the condenser pressure lowers the temperature of the steam, and thus temperature at which heat is rejected.

The effect of lowering the condenser pressure is illustrated in Figure 2.8. For purposes of comparison the inlet state temperature is maintained the same. The net work increases. The heat input requirement increases (represented by the area under $2\text{-}2^{'}$), but this is small compared to the increase in the net work output (i.e. area $1\text{-}4\text{-}4^{/}\text{-}1^{/}$). Thus, *the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle*.

The condensers of steam power plants operate well below the atmospheric pressure. But there is a limit on the condenser pressure that can be used. It cannot be lower than the saturation pressure corresponding to the temperature of the cooling medium. For example, for a condenser that is cooled by a nearby water at say 15 ^{0}C allowing for difference of 10 ^{0}C for effective heat transfer, the steam temperature in the condenser must be above 25 ^{0}C. From steam tables, the condenser pressure must be above 3.2 kPa (i.e. the corresponding pressure at 25 ^{0}C).

The side effects of lowering the condenser pressure option are:

- It creates the possibility of air leakage into the condenser
- It increases the moisture content of steam at the final stages of the turbine, which is undesirable in turbines because it decreases the turbine efficiency and erodes the turbine blades.

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

The average temperature at which heat is added to the steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of a vapour power cycle is illustrated in Figure 2.9.

We recognise an increase in the net work. The area under curve 3-3$^{/}$ represents the increase in heat input. Thus superheating the steam increases both the net work and the heat input. The overall effect is an increase in cycle efficiency since the average temperature at which heat is added is increased.

Other advantages of superheating:

- It decreases the moisture content of the steam at the turbine exit

Limitation of superheating:

- The temperature to which steam is superheated is limited by metallurgical limit of material used. T = 620 ^{0}C is about the present optimum.

(c) Increasing Boiler Pressure (Increases $T_{high,\ av}$)

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

The effect of increasing the boiler pressure is illustrated in Figure 2.10. We note that for a fixed turbine inlet temperature, the moisture content of steam at the turbine exit increases. This undesirable effect is corrected with reheating.

Today modern power plants operate at supercritical pressures (P > 22.1 MPa) and have thermal efficiencies of 40% for fossil-fuel plants and 34 % for nuclear plants. The lower thermal efficiency values for nuclear plants are due to the fact that lower maximum temperatures values are used in those plants for safety reasons.

5.12 Reheat Cycle (Ideal Reheat Cycle)

It is desirable to increase the average temperature at which heat is supplied to the steam, and also to keep the steam as dry as possible in the lower pressure stages of the turbine. The wetness of steam at the turbine exhaust should be no greater than 10 %.

Higher boiler pressures are required for high efficiency, but expansion in one stage can result in the exhaust steam, which is wet. This condition is somehow improved by superheating but it is further improved by re-heating the steam, the expansion being carried out in two stages. With the reheat cycle, the expansion takes place in two turbines. The steam expands in the high-pressure turbine to some intermediate pressure, and is then passed back to yet another bank of tubes in the boiler where it is reheated at constant pressure, usually to the original superheat temperature. It then expands in the low pressure turbine to the condenser pressure. The Reheat cycle appears as in Fig. 2.11.

<u>Two options to increase quality of steam at the turbine exit:</u>

1) Superheat steam to very high temperature is appropriate but **<u>not</u>** a viable solution since it will require raising the steam to metallurgical unsafe limits.
2) Expands steam in two stages, and reheat in between. This is practical solution to reducing the wetness of steam of turbine exit.

The larger the number of reheats the higher the average temperature of heat addition process. But reheating more than twice is not desirable and practical. Double reheat is only used on supercritical (P>22.1 MPa) power plants.

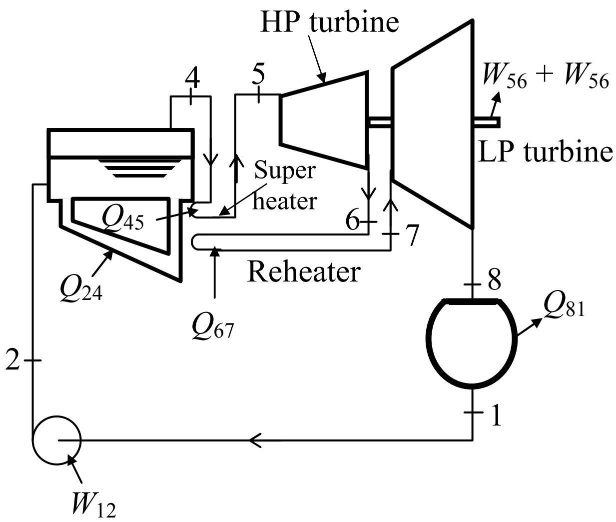


Fig 2.11a The Reheat cycle

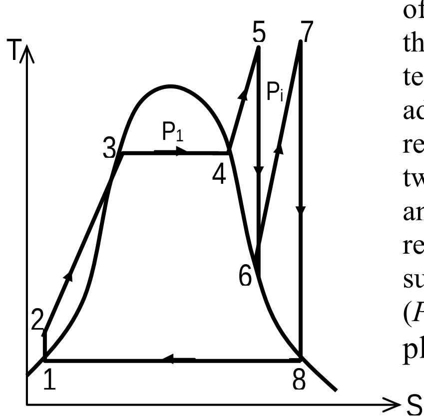


Fig. 2.11b T-s diagram of the Reheat cycle

<u>5-6</u> Isentropic expansion in the high-pressure turbine to an intermediate pressure. To determine the intermediate pressure, we find from the saturation' table the pressure at which $s_g = s_6$ To find s_6 use the superheat tables.

<u>6-7:</u> Isentropic expansion in the low pressure turbine.

Heat supplied = $Q_{245} + Q_{67}$

Work output = $W_{56} + W_{78}$, Also, $Q_{67} = h_7 - h_6$

Cycle efficiency, $$\boxed{\eta = -\frac{W_{56} + W_{78}}{Q_{245} + Q_{67}} = -\frac{h_6 - h_5 + h_8 - h_7}{h_5 - h_2 + h_7 - h_6}}$$

If we neglect the feed pump work, $h_2 \approx h_1$ and $\therefore Q_{245} = h_5 - h_1$

<u>Boiler Efficiency:</u>
The boiler efficiency is the heat supplied to the steam in the boiler expressed as a percentage of the chemical energy of the fuel, which is available on combustion. That is;

$$\text{Boiler efficiency} = \frac{\dot{m}_s x\left[h_5 - (\text{enthalpy of the feed water})\right]}{\dot{m}_f x(\text{GCV or NCV})}$$

where h_5 is the enthalpy of the steam entering the turbine and $\dot{m}_f$ the mass of fuel burned per unit time and $\dot{m}_s$ the mass flow rate of steam. The GCV and NCV are the higher and lower calorific values of the fuel. The size of a boiler, or its capacity, is quoted as the rate in kilogram per hour at which the steam is generated.

Comments:

Reheating reduces the steam consumption appreciably, because the area of the cycle on the T-S diagram, which equals the network done per unit mass of steam, increases. In high-pressure cycles this implies a smaller boiler (expensive item in high-pressure plant). This decrease in size goes with added complexity. Note that the effective average temperature at which the heat is added may be lower than that in the Rankine cycle if the intermediate pressure is too low. The use of re-heating gives drier steam at the turbine exit thus reducing blade erosion in the latter stages of the turbine. The cost of reheating, however, adds to the capital cost of the plant.
Sole purpose of reheating is to reduce the moisture content of the turbine exit (fluid stages in the expansion process). Therefore, if we had materials that could withstand high temperatures there would be no need for reheating.

Cogeneration System

Utilization factor $\xi = \dfrac{|W_{ST}| + Q_{pa}}{Q_{IN}} \qquad \xi = 1 - \dfrac{|Q_{out}|}{Q_{in}}$

In actual cogeneration plants the utilization factor $\xi = 70\%$

In all cycles discussed so far, the sole purpose was to convert a portion of the heat transferred to the working fluid to work, which is the most valuable form of energy. The remaining portion of the heat is rejected to rivers, lakes, oceans or the atmosphere as waste heat. Wasting a large amount of heat is a price we have to pay to produce work. Many systems or devices, however, require energy input in the form of heat, called process heat. A plant that produces electricity while meeting the process-heat requirements of certain plants is called a co-generation plant.

<u>Note</u>: A plant for the productio of more than one useful form of energy from the same energy source is called a co-generation plant.

5.16 Further considerations in improving the efficiency of a steam power plant

Hitherto, considerations of efficiency have been based on the heat, which is actually supplied to the steam, and not the heat, which has been produced by the combustion of the fuel in the boiler. The heat is transferred to the steam from gases, which are at a higher temperature (approx. 2000 K) than the steam, and the exhaust gases pass to the atmosphere at a high temperature.

Energy of the flue gases can be utilized to preheat the feed water before entry into the boiler. The external heat supplied is reduced thus increasing the thermal efficiency of the system.

<u>The Economiser</u>:
To utilise some of the energy in the flue gases an *economiser* can be fitted (see Fig. 2.15(a)). This is heat exchanger placed in the flue gases, which extracts useful energy in the form of heat to preheat the feed water before entry to the boiler. (*For the Carnot, ideal regenerative and complete feed heating cycles,* ***no*** *use can be made of an economiser since the feed water enters the boiler at the saturation temperature corresponding to the boiler pressure*).

<u>The Air pre-heater:</u>
To cool the flue gases even further and improve the plant efficiency, the air, which is required for the combustion of the fuel, can be pre-heated (see Fig. 2.15 (b)). For a given temperature

of the combustion gases, the higher the initial temperature of the air then the less will be the energy input required, and hence less fuel will be used.

One practical consequence of cooling the combustion gases is that there is a pressure drop in the chimney and fans may be required to achieve the necessary forced draught for the flue gases.

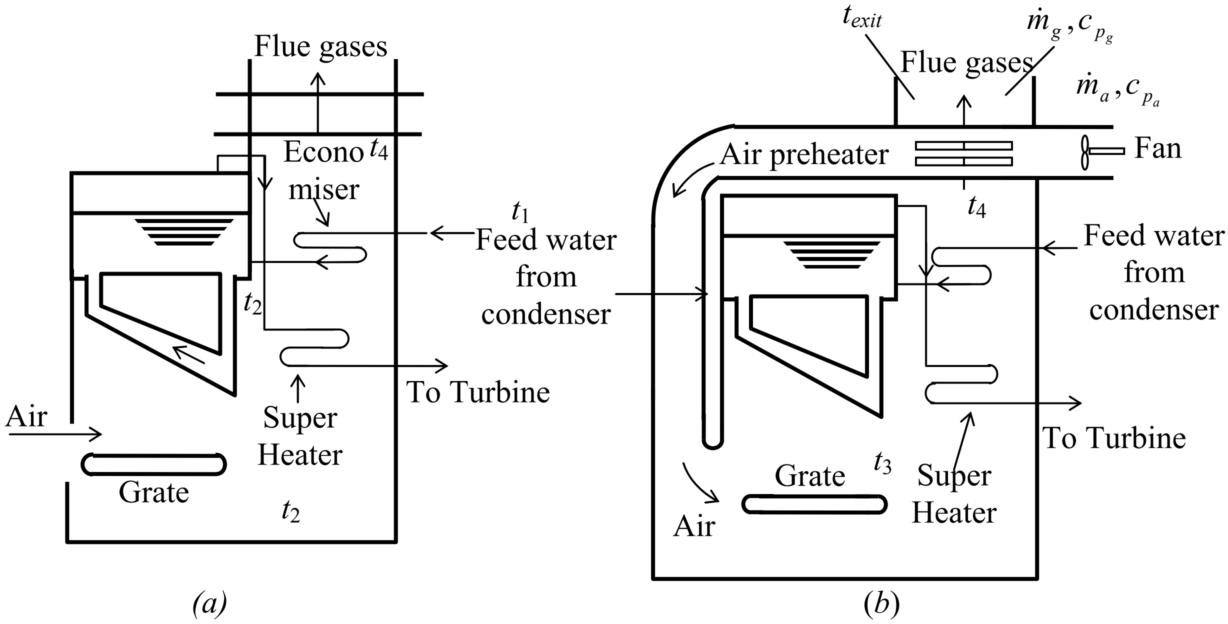


Fig. 5.15 Diagrammatic sketch of a boiler plant with (a) an economiser (b) an air preheater

Analysis of the Economiser

Let us assume that the temperature of the feed water is raised from t_1 to t_2 in the economiser and that the flue gases are also cooled from t_3 to t_4.

For air pre-heater $\dot{m}_a x c_{pa} \Delta t = c_{pg} x \dot{m}_g \left(t_4 - t_{exit}\right)$

Heat Exchanger

We have $\dot{m}_g c_{pg} \left(t_3 - t_4\right) = \dot{m}_s \left(h_2 - h_1\right) + m c_p \left(t_{exit} - t_{entry}\right)$