

Thermodynamics

(1)

Introduction :-

Thermo (Heat)
(Greek word)

- Thermodynamics is branch of science which deals with the relations among heat, work and properties of system which are in equilibrium.
- Thermodynamics basically describes four laws, Zeroth law, first law, second law and third law.
- The Zeroth law deals with thermal equilibrium and establishes a concept of temperature.
- The second law indicates the limit of converting heat into work and introduces the principle of increase of entropy.
- Third law defines absolute zero of entropy.

These laws are based on experimental observations and have no mathematical proof. These laws are based on logical reasoning.

Thermodynamic Systems :-

system :-

A system is a finite quantity of matter or prescribed region of space.

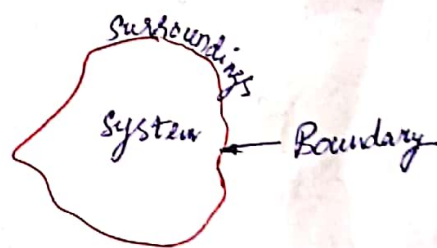


Fig → System

Boundary :-

- The actual or hypothetical envelope enclosing the system is the boundary of the system.
- The boundary may be fixed or it may move, as and when a system containing a gas is compressed or expanded.

→ The boundary may be real or imaginary.

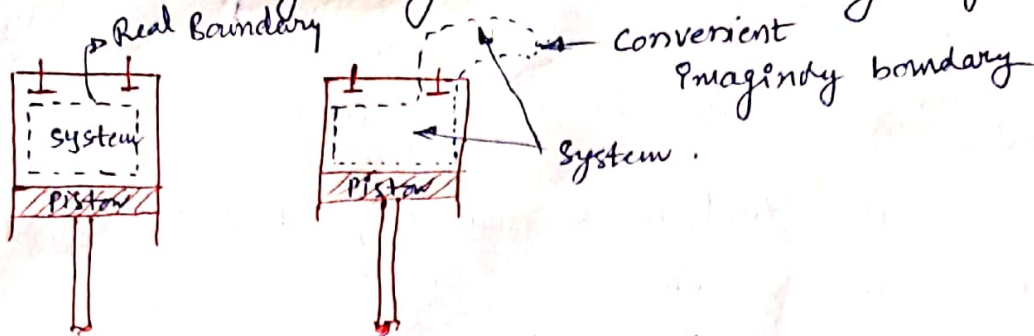


Fig → The Real and Imaginary Boundary.

Surroundings:-

- Anything outside of the system is called surroundings.
- Boundary separates system and surroundings.

$$\text{System} + \text{Surroundings} = \text{Universe.}$$

Closed System :- Energy can cross.
Mass can't cross.

- If the boundary of the system is impervious to the flow of matter, it is called closed system.

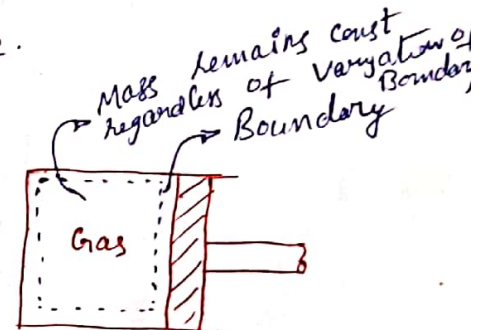


Fig → Closed System.

- In this system energy can enter or leaves the system.
- An example of this system is mass of gas or vapour contained in an engine cylinder, the boundary of which is drawn by cylinder walls, the cylinder head and piston crown.

→ Hence the boundary is continuous and no matter may enter or leave.

Open System :-

An open system is one in which both energy and matter flows into or out of the system. Most of the engineering systems are open.

ex:- Air compressor/ Turbines, boilers - - etc.

Isolated System :-

An isolated system is that system which exchanges neither energy nor matter with any other system or environment.

ex:- Thermos flask.

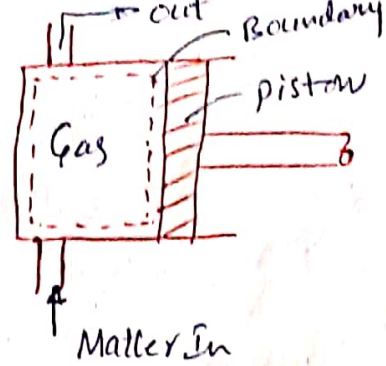


Fig → open system.

Adiabatic system :-

mass can cross (work)
Energy can't cross (heat)

→ An adiabatic system is one which is thermally insulated. It can however exchange work with its surroundings.

→ If adiabatic system does not exchange work it will become isolated system. Ex:- piston working inside a insulated cyl.

Phase :-

A phase is a quantity of matter which is homogeneous throughout in chemical composition and physical structure.

Based on phase, systems are of 2 types.

1) Homogeneous system :-

A system which consists of single phase is known as homogeneous system.

Ex:- Mixture of air and water vapour, water and milk, octane & butane

2) Heterogeneous system :-

A system which consists of two or more phases is known as heterogeneous system.

Ex:- Water and steam, ice and water, water and oil.

Macroscopic and Microscopic point of view:-

Thermodynamics studies are undertaken by following 2 different approaches.

1. Macroscopic approach - Macro means big and total
2. Microscopic approach - Micro means small.

Macroscopic - Engg. approach

→ In this certain quantity of matter is considered without taking into account of the events occurring at molecular level.

Another words this approach to thermodynamics is concerned with gross or overall behaviour.

This is known as classical T.D.

→ This approach requires simple mathematical formulas.

→ The values of properties of the system are their average value. For ex consider a gas in a closed container. The pr of gas is avg value of pr. exerted by millions of individual molecules. // by Temp.

Like pr, Temp

These properties can be measured very easily and changes in properties felt by our senses.

→ Few properties are required to describe a syst. So it is simple.

Microscopic - Scie. approach

→ The approach considers that system is made up of a large no. of discrete molecules. These molecules have diff velocities and energies.

The values of energies are constantly changing with time. This approach to thermodynamics which is concerned directly with the structure of matter is known as statistical T.D.

→ This approach requires advanced statistical and mathematical methods.

→ The properties like vel, momentum, impulse, force of impact - etc which describe the molecules cannot be easily measured with instruments. Our senses cannot feel them.

→ Large no of variables are needed to describe a system. So it is complicated.

Pure Substance:-

(3)

A pure substance is one which has a homogeneous and invariable chemical composition even though there is a change of phase.

ex:- (liquid, water) mixture of ice and water.

Thermodynamic Equilibrium:-

→ A body is said to be in thermal equilibrium if it obeys three types of equilibrium namely thermal, mechanical and chemical equilibrium.

1. Thermal Equilibrium:-

The temperature of the system doesnot change with the time and has ^{same} ~~all~~ value at all points of the system.

2. Mechanical Equilibrium:-

There are no unbalanced forces with in the system or between the surroundings. The pressure in the system is same at all the points and doesnot change with respect to time.

3. Chemical equilibrium:-

No chemical reaction takes place in the system and chemical composition which is same throughout the system.

Properties of System:-

1. Intensive properties:-

These properties doesnot depend on mass of the system.

Ex:- Pressure, Temperature

2. Extensive Properties :-

These properties depend on the mass of the system.

Ex:- Volume, weight, density - - etc.

Extensive properties are often divided by mass associated with them to obtain the intensive properties.

Ex:- $\frac{V}{m} = v$ = specific volume, which is independent of mass.

State :-

→ State is the condition of the system at an instant of time.
or

Each unique condition of the system is called a state.

→ Each property has a single value at each state.

→ A variable is a property, if and only if, it has a single value at each equilibrium state.

Process :-

→ A process occurs when the system undergoes a change in a state.

→ A process may be non-flow in which fixed mass within the defined boundary is undergoing a change of state.

Ex:- A substance which is being heated in a closed cylinder undergoes a non-flow process. All closed systems undergoes non-flow process.

→ A process may be a flow process in which mass is entering and leaving through the boundary of a system.

Ex:- All open systems undergoes flow process.

Quasi-static process :-

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→ Quasi means almost. A quasi static process is also called reversible process.

→ This process is a succession of equilibrium states and infinite slowness is its characteristic feature.

Cycle :-

→ Any process or series of processes whose ^{start} end states are identical is called as cycle.

→ The process through which the system has passed can show on state diagram.

Fig shows a cycle a system commencing at condition 1, changes in Pressure and volume through a path 123 and returns to initial condition 1.

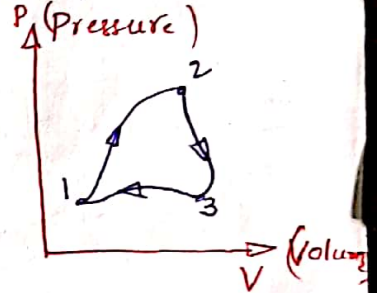


Fig → Cycle of operations.

Point function :-

There are certain properties which can be located on a graph by a point. Those properties are called as point function.

EX:- Pressure, Temperature, Volume - - - etc.

$$\int_1^2 dV = V_2 - V_1 \rightarrow \text{An exact differential.}$$

Path function :-

There are certain properties which cannot be located on a graph by a point, but are given by area. In that case the area on the graph pertaining to particular process is a function of path of the process. Those properties are called as path-functions.

EX:- Heat and work. These are inexact differentials.

$$\int_1^2 \delta Q \neq Q_2 - Q_1, \quad \int_1^2 \delta W \neq W_2 - W_1 \quad \text{in exact differential.}$$

Temperature :-

- Temperature is a thermal state of a body which distinguishes a hot body from cold body.
- Instruments for measuring ordinary temperatures are known as thermometers and those for measuring high temperatures are known as pyrometers.
- It has been found that gas will not occupy any volume at certain temperature. This temperature is known as absolute zero temperature.
- The point of absolute temperature is found to occur at 273.15°C below the freezing point of water.
 $\therefore \text{Absolute Temperature} = \text{Thermometer reading in } ^{\circ}\text{C} + 273.15$
- Absolute temperature can be represented by $\times$ ^{degree} Kelvin (K)

Zeroth Law of T.D :- If 2 thermodynamic systems are each in thermal equilibrium with a third, then they are in thermal equilibrium with each other.

⇒ A temp. scale of certain thermometer is given by the relation $t = a \log p + b$ where a and b are const and p is the thermometric property of the fluid in thermometer. If at ice point and steam point thermometric property are found to be 1.5 and 7.5 resly. what will be the temp corresponding to the thermometric property of 3.5 on Celsius scale

Sol $t = a \log p + b$

on Celsius scale

$$\text{Ice point} = 0^\circ\text{C}, \quad \text{Steam point} = 100^\circ\text{C}$$

from given condition

$$0 = a \log 1.5 + b$$

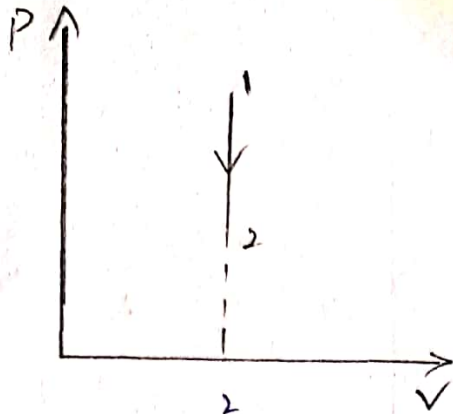
$$100 = a \log 7.5 + b$$

$$\Rightarrow a = 143.06 \quad b = -25.30$$

when $p = 3.5$

$$t = \underline{\underline{52.66^\circ\text{C}}}$$

1. Constant volume process:-

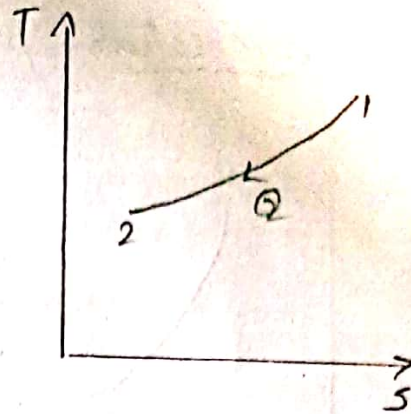


$$\oint W_{1-2} = P dV$$

$$\delta W = 0$$

$$\delta Q = dU$$

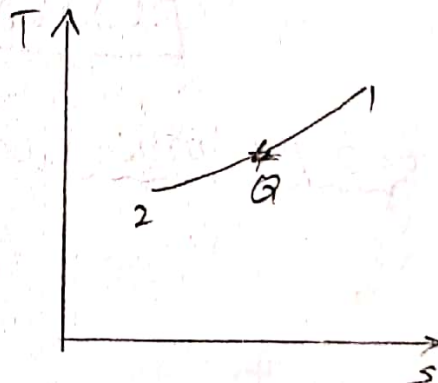
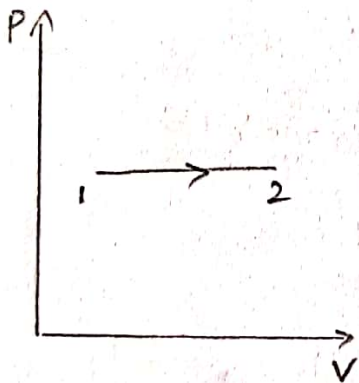
$$\therefore \delta Q = m C_V dT$$



$$\partial Q - \partial W = dU$$

$$\partial Q = \partial W + dU$$

2. constant pressure process:-



$$PV \propto T$$

$$\frac{V}{T} = \text{const}$$

$$\partial Q - \partial W = dU$$

$$\partial W = \int_1^2 P dV = P(V_2 - V_1)$$

$$\partial Q = P(V_2 - V_1) + (U_2 - U_1)$$

$$= (P_2 V_2 + U_2) - (P_1 V_1 + U_1)$$

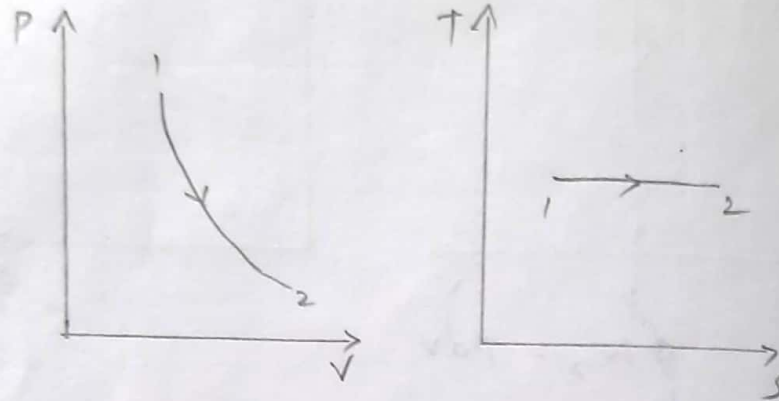
$$\Delta Q = H_2 - H_1$$

$$H = mc_p dT$$

$$C_p = \frac{dh}{dT}$$

$$C_v = \frac{du}{dT}$$

3) constant temperature process:-



$$pVdT$$

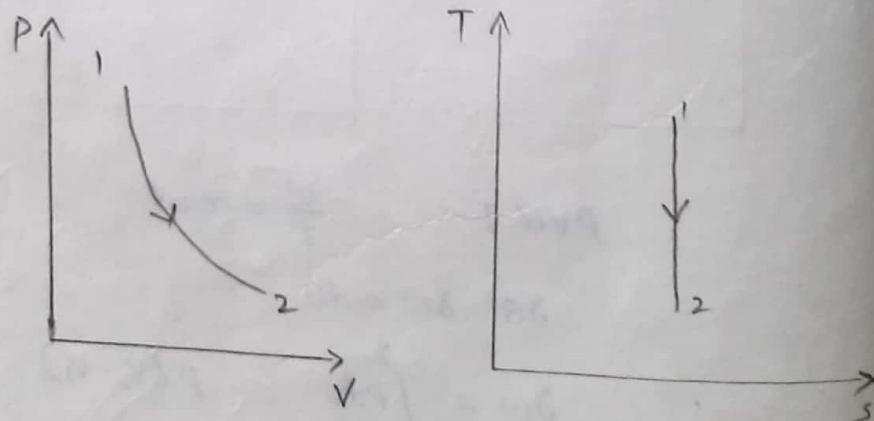
$$pV = C \Rightarrow p = \frac{C}{V}$$

$$W = \int p \cdot dv = C \int \frac{1}{V} dv = C \ln\left(\frac{V_2}{V_1}\right)$$

$$du = -mC_v(T_2 - T_1) = 0$$

$$\therefore \Delta Q = \Delta W$$

4) constant entropy process (Adiabatic):-



$$PV^\gamma = \text{const}$$

$$P = \frac{C}{V^\gamma} = C \cdot V^{-\gamma}$$

$$W = C \int_1^2 V^{-\gamma} dV$$

$$= C \left[\frac{V^{-\gamma+1}}{-\gamma+1} \right]_1^2$$

$$= PV^\gamma \left[\frac{V_2^{-\gamma+1} - V_1^{-\gamma+1}}{-\gamma+1} \right]$$

$$= \left[\frac{P_2 V_2^\gamma V_2^{-\gamma+1} - P_1 V_1^\gamma V_1^{-\gamma+1}}{-\gamma+1} \right]$$

$$\Rightarrow \frac{P_2 V_2 - P_1 V_1}{-\gamma+1}$$

$$\boxed{W = \frac{P_1 V_1 - P_2 V_2}{\gamma-1}}$$

$$\partial \theta = 0$$

$$\boxed{\therefore \partial W = dU}$$

UNIT-1 Reciprocating Air Compressors

Introduction:-

- An air compressor is a machine to compress the air and to raise its pressure.
- It sucks air from atmosphere, compresses it and then delivers the same under a high pressure to storage vessel.
- From storage vessel it may be conveyed by pipeline to a place where the supply of compressed air is required.
- Since the compression of air requires some work to be done on it, therefore a compressor must be driven by some prime mover.

Applications of Compressed air:-

- It is used for many purposes such as for operating pneumatic drills, riveters, road drills, paint spraying, in starting and supercharging of internal combustion engines, in gas turbine plants, jet engines -- etc.
- It is also utilised in the operation of lifts, rams, pumps -- etc.
- In industry compressed air is used for producing blast of air in blast furnaces.

Classification of Air Compressors:-

The air compressors may be classified in many ways, but the following are important from subject point of view.

1. According to working:-

- a) Reciprocating air compressor
- b) Rotary air compressor.

2. According to action:-

- a) Single acting air compressor
- b) Double acting air compressor

3. According to number of stages:-

- a) Single stage air compressor
- b) Multi stage air compressor.

Terminology in Air Compressors:-

1. Inlet pressure:- (Suction pressure)

It is the absolute pressure of air at the inlet of a compressor.

2. Discharge pressure:-

It is the absolute pressure of air at the outlet of a compressor.

pg-②

3. Compression Ratio (Pressure Ratio):-

→ It is the ratio of discharge pressure to inlet pressure.

→ The ratio is always greater than unity because the delivery pressure is greater than suction pressure.

4. Compressor Capacity:-

It is the volume of air delivered by the compressor, and is expressed in m^3/min or m^3/sec .

5. Free Air Delivery:-

→ It is the actual volume delivered by a compressor when reduced to the normal temperature and pressure condition.

→ The capacity of compressor generally given in terms of free air delivery.

6. Swept Volume:- (Displacement)

It is the volume of air sucked by compressor during its suction stroke.

Mathematically

$$\text{Swept volume, } V_s = \frac{\pi}{4} d^2 L \quad (\text{for single acting})$$

where d = Dia of cylinder bore

L = Length of piston stroke.

7. Mean effective pressure:-

In air compressor the pressure acting on the piston keeps on changing with the movement of piston in the cylinder.

→ The mean effective pressure of the cylinder compressor is found mathematically by dividing the workdone per cycle to the swept volume.

$$P_{\text{mean}} = \frac{\text{WORK done}}{\text{Swept Volume.}}$$

* Working of Single Stage Reciprocating Air Compressor :-

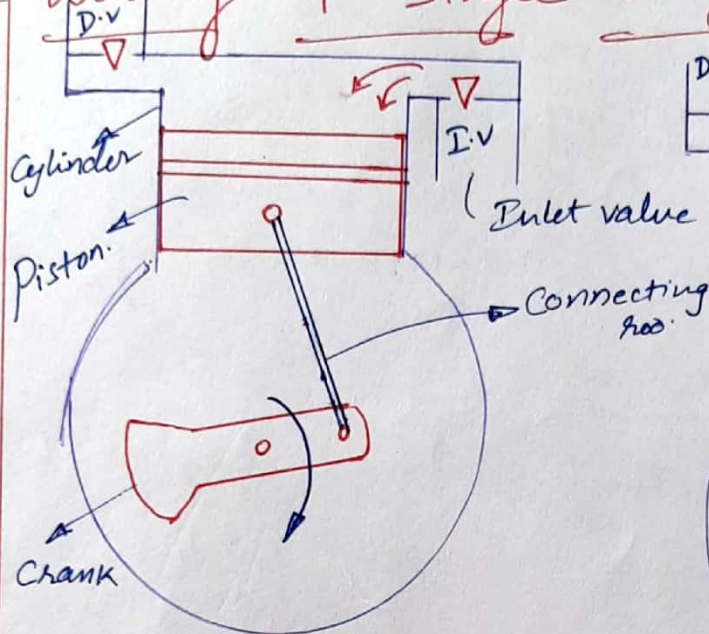


Fig:- Suction Stroke

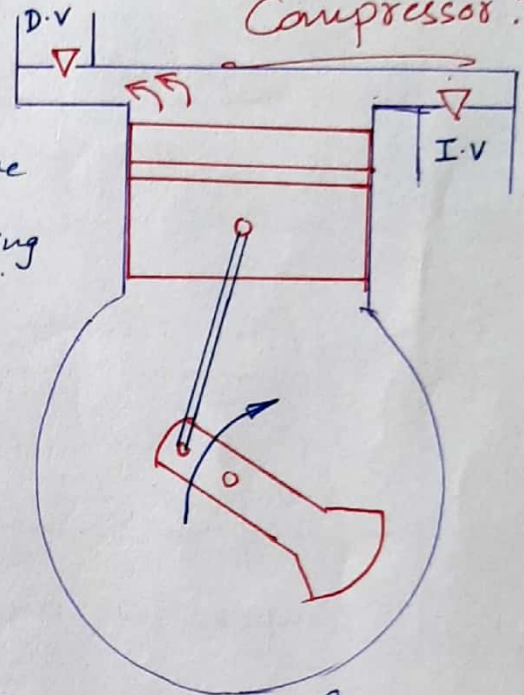


Fig:- Delivery Stroke.

→ A single stage reciprocating air compressor consists of a cylinder, piston, crank, connecting rod, I.V, D.V as shown in figure.

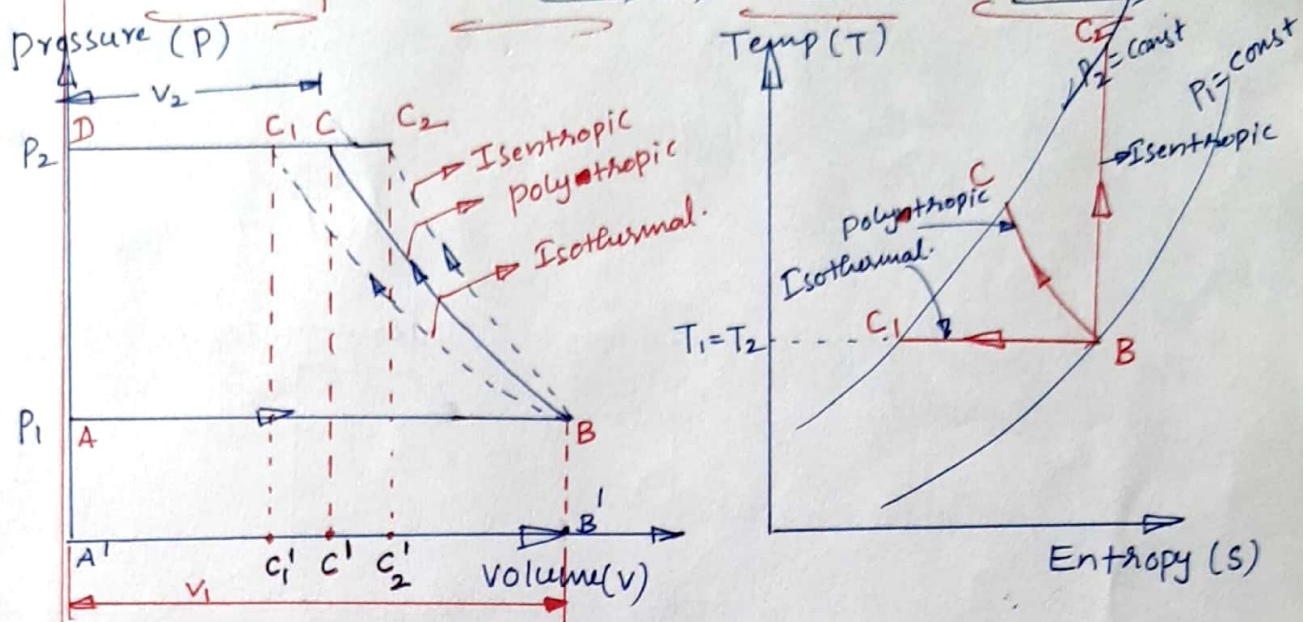
→ When piston moves downwards, the pressure inside the cylinder falls below the atmospheric pr. Due to this pressure difference, inlet valve (I.V) gets opened and air is sucked in to the cylinder at inlet pressure.

- pg-③ → Now when the piston moves upwards, the pressure inside the cylinder goes on increasing till it reaches discharge pressure.
- At this stage, the discharge valve gets opened and air is delivered to the container.
- At the end of delivery stroke, a small quantity of air at high pressure is left in clearance space.
- As the piston starts its suction, the air contained in clearance space expands till its pressure falls below atmosphere.
- At this stage, the inlet valve (I.V) gets opened as a result of which fresh air is sucked into the cylinder and the cycle is repeated.

NOTE → ① In single acting reciprocating ~~pump~~ compressor the suction, compression and delivery of air takes place in 2-strokes of piston or one revolution of crank shaft.

② In double acting, suction, compression and delivery takes place on both sides of piston. So it supplies double the volume of air than single acting reciprocating compressor.

* WORK done by a single stage Reciprocating Air Compressor without clearance volume :-



A $\rightarrow$ B = Suction
 B $\rightarrow$ C = Compression
 C $\rightarrow$ D = Discharge

Consider a single acting reciprocating air compressor without clearance volume.

Let P_1 = Initial pressure of air (Before compression)

V_1 = Initial Volume of air (" ")

T_1 = Initial Temp of air (" ")

P_2, V_2, T_2 are corresponding values of air at delivery

r = pressure ratio i.e. $\frac{P_2}{P_1}$

P-V & T-S diagram of air compressor are shown in the fig. During return stroke of the piston, the air is compressed at const temp.

The compression continues till the pressure (P_2) in the cylinder is sufficient to force open the delivery valve at C. After that no more compression takes place.

pg No. 41

- Now during remaining stroke of compression, the compressed air is delivered till the piston head reaches the cylinder end.
- After that the air is sucked from atmosphere during suction stroke AB at pressure P_1 .
- The compression of air maybe isothermal, polytropic or isentropic.

1. Workdone during Isothermal compression :- ($PV = \text{const}$)

Isothermal compression & ^{delivery} of air is shown by BC, & CD. CD represents the volume of air delivered.

Workdone by air compressor per cycle $W = \text{Area under the curve ABCD}$.

$$\begin{aligned}
 W &= \text{Area ABCD} \\
 &= \text{Area A'DC'C'} + \text{Area CBB'C'} - \text{Area A'ABB'} \\
 &= P_2 V_2 + 2.3 P_2 V_2 \log \left(\frac{V_1}{V_2} \right) - P_1 V_1 \\
 &= 2.3 P_1 V_1 \log \left(\frac{V_1}{V_2} \right) = 2.3 P_1 V_1 \log \left(\frac{P_2}{P_1} \right) \\
 &\quad (\because P_1 V_1 = P_2 V_2) \\
 &= 2.3 P_1 V_1 \log (\gamma) = \underline{\underline{2.3 m R T_1 \log \gamma}}
 \end{aligned}$$

2. Workdone during polytropic compression :- ($PV^n = \text{const}$)

polytropic compression is shown by line BC.

Workdone by air compressor per cycle $W = \text{Area under the curve ABCD}$.

$$\begin{aligned}
 W &= \text{Area ABCD} \\
 &= \text{Area A'DC'C'} + \text{Area CBB'C'} - \text{Area A'ABB'}
 \end{aligned}$$

pg 4

$$W = P_2 V_2 + \frac{P_2 V_2 - P_1 V_1}{\eta - 1} - P_1 V_1$$

$$= P_2 V_2 - P_1 V_1 \left[1 + \frac{1}{\eta - 1} \right]$$

$$W = \frac{\eta}{\eta - 1} (P_2 V_2 - P_1 V_1)$$

$$= \frac{\eta}{\eta - 1} P_1 V_1 \left[\frac{P_2 V_2}{P_1 V_1} - 1 \right]$$

for polytropic process, $P_1 V_1^\eta = P_2 V_2^\eta$

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2} \right)^{1/\eta}$$

$$\therefore W = \frac{\eta}{\eta - 1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right) \left(\frac{P_1}{P_2} \right)^{\frac{1}{\eta}} - 1 \right]$$

$$= \frac{\eta}{\eta - 1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\eta - 1}{\eta}} - 1 \right]$$

$$W = \frac{\eta}{\eta - 1} m R T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\eta - 1}{\eta}} - 1 \right] = \frac{\eta}{\eta - 1} m R (T_2 - T_1)$$

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

$$\Rightarrow \frac{V_1}{V_2} = \frac{P_2}{P_1} \cdot \frac{T_1}{T_2} \rightarrow (1)$$

$$P_1 V_1^\eta = P_2 V_2^\eta$$

$$\Rightarrow \frac{V_1}{V_2} = \left(\frac{P_2}{P_1} \right)^{1/\eta} \rightarrow (2)$$

$$(1) = (2)$$

$$\frac{P_2}{P_1} \cdot \frac{T_1}{T_2} = \left(\frac{P_2}{P_1} \right)^{1/\eta}$$

$$\frac{T_1}{T_2} = \left(\frac{P_2}{P_1} \right)^{\frac{1 - \eta}{\eta}}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\eta - 1}{\eta}}$$

3. Workdone during isentropic compression ($PV^\gamma = \text{const}$):-

→ The isentropic compression is shown by curve BC_2 .

→ Workdone on air per cycle during isentropic compression may be calculated in the same way as polytropic Compⁿ.

$$\text{Workdone on air per cycle } W = \frac{\gamma}{\gamma - 1} m R T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]$$

$$W = \frac{\gamma}{\gamma - 1} m R (T_2 - T_1)$$

We know that $C_p - C_v = R$ & $\frac{C_p}{C_v} = \gamma$

$$R = C_p \left(1 - \frac{C_v}{C_p} \right)$$

$$R = C_p \left(1 - \frac{1}{\gamma} \right) = C_p \left(\frac{\gamma - 1}{\gamma} \right)$$

$$\therefore W/d = \frac{\gamma}{\gamma - 1} m C_p \left(\frac{\gamma - 1}{\gamma} \right) (T_2 - T_1) = m C_p (T_2 - T_1) = W$$

→ Workdone on air is minimum when the compⁿ is isothermal and it is maximum when the compⁿ is isentropic.

→ In order to perform isothermal process, the compression should be very slow so that the temperature is maintained const. which is not possible in actual practice.

→ However isothermal compression is achieved if

- ① the cooling (air/water) is done during compⁿ.
- ② By injecting the cold water during compⁿ.
- ③ By adding intercoolers in multistage compⁿ.

*

Power required to drive single stage Air compressor:

$$\text{Power} = \frac{\text{Workdone} \times \text{Number of working strokes/min}}{60} \quad \text{watts}$$

$$= \frac{W \times N_w}{60} \quad \text{watts}$$

$N_w = N$ for single acting

$= 2N$ for double acting

where N = Speed of compressor in r.p.m.

$$\therefore \text{Isothermal power} = \frac{W_{\text{isothermal}} \times N_w}{60} \quad \text{watts}$$

$$\text{Isentropic power} = \frac{W_{\text{isentropic}} \times N_w}{60} \quad \text{watts}$$

$$\text{Indicated power} = \frac{W_{\text{polytropic}} \times N_w}{60} \quad \text{watts}$$

→ Indicated power is also known as Power of Compressor.

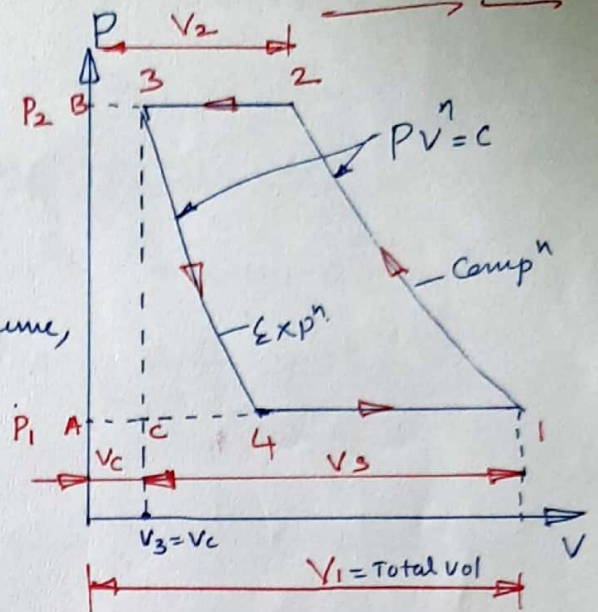
* Workdone by Reciprocating Air Compressor with Clearance Volume:-

Consider a reciprocating air compressor with clearance volume

Let

P_1, V_1, T_1 are pressure, volume, Temp of air before compression.

P_2, V_2, T_2 are corresponding values at delivery.



$$r = \text{pressure ratio} = \frac{P_2}{P_1}$$

$$V_c = \text{Clearance volume} \neq V_1 - V_3$$

$$V_s = \text{Stroke/swept volume} = V_1 - V_c$$

$$\eta = \text{polytropic index for } \text{Comp}^n \text{ \& exp}^n.$$

$$V_1 - V_4 = \text{Effective swept vol.}$$

$$V_1 - V_3 = \text{Swept vol.}$$

$$V_3 = V_c = \text{Clearance vol.}$$

→ During return stroke the air is compressed by its major part i.e. compression stroke 1-2. This stroke continues till the pressure reaches P_2 .

→ This pressure P_2 , force the delivery valve at 2.

→ During remaining stroke of compression stroke, compressed air is delivered till the piston reaches 3.

→ At this stage there will be some air (equal to clearance volume) left in the clearance space will expand during outward stroke of piston i.e. exp^n stroke 3-4.

→ This exp^n continues till the pressure P_1 in the cylinder is sufficient to force open the I.V at 4. After this the air is sucked from atmosphere during suction stroke 4-1 at pressure P_1 .

Let assume the compression and expansion may be polytropic.

Work done by the compressor per cycle = $W = \text{Area } 1-2-3-4$.

$$= \text{Area } A12B - \text{Area } A43B.$$

$$= \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} P_2 V_4 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$W = \frac{n}{n-1} P_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] (V_1 - V_4)$$

Note →

V_4 is called expanded clearance volume.

$(V_1 - V_4)$ is called effective swept volume.

Need for Multistage Compression:-

In single stage compression, if the requirement of pressure of compressed air is more than that case, we employ a large pressure ratio in a single cylinder. (8 to 10 bar)

If we employ a single stage compression for producing higher pressure ratio, it suffers from following drawbacks.

- ① The size of cylinder is too large.
- ② Due to compression, there is rise in temp of air. It is difficult to reject heat from the air in small time available for compression.
- ③ Due to higher air temp at the end of compⁿ, it may heat up cylinder head or burn the lubricating oil.

Advantages of Multi Stage Compression :-

1. The work done per kg of air is reduced in multi stage compression with intercooler as compared to single stage compression for the same air delivery.
2. It improves volumetric efficiency for same p_r ratio.
3. The size of 2 cylinders (H.P & L.P) may be adjusted to suit the volume and pressure of air.
4. It reduces leakage loss.
5. It produces more uniform torque and hence smaller size fly wheel is required.
6. It produces effective lubrication because of lower temperature range.
7. It reduces the cost of compressor.

pg no. ⑦

2-stage Reciprocating Air Compressor with Intercooler

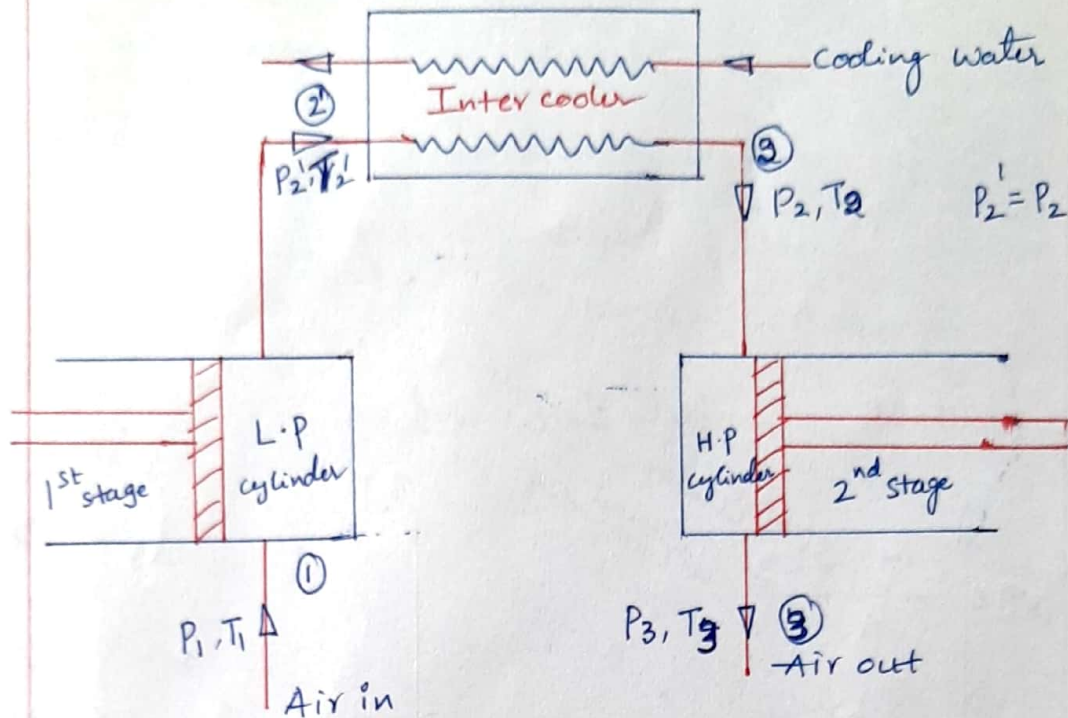


Fig → 2 stage Reciprocating Air Compressor with Intercooler.

- Fresh air is sucked from atmosphere in low pressure cylinder during its suction stroke at pressure P_1 , temperature T_1 .
- The air after compression in L.P. cylinder from 1 → 2', is delivered to the intercooler at pressure P_2' , temp T_2' .
- Now the air is cooled in intercooler from 2' to 2 at const pressure P_2 and from temperature T_2' to T_2 .
- After that the air is sucked in the high pressure cylinder during its suction stroke.
- Finally the air is further compressed in H.P. cylinder from 2 → 3 and its pressure is P_3 , temperature T_3 .

Assumptions:-

1. The effect of clearance is neglected.
2. There is no pressure drop in intercooler
3. The compression in both cylinders is polytropic
4. The suction and delivery of air takes place at constant pressure.

1. Perfect / Complete Intercooling:-

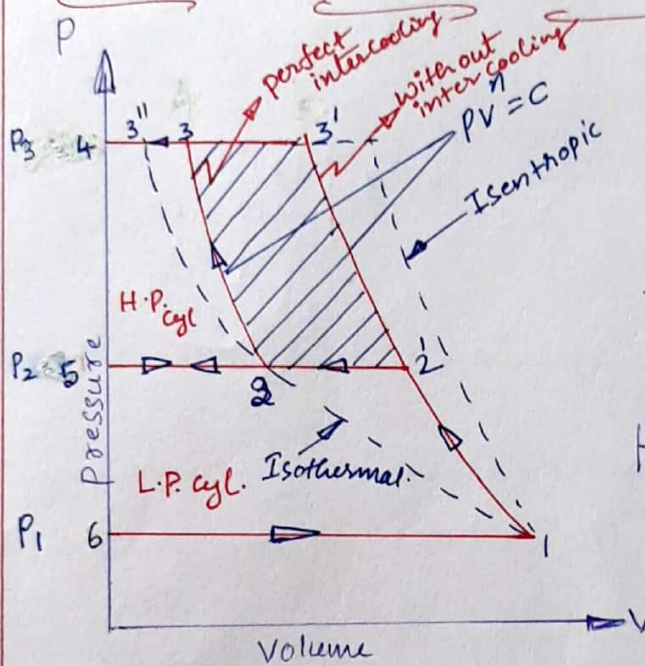


Fig → P-V diagram.

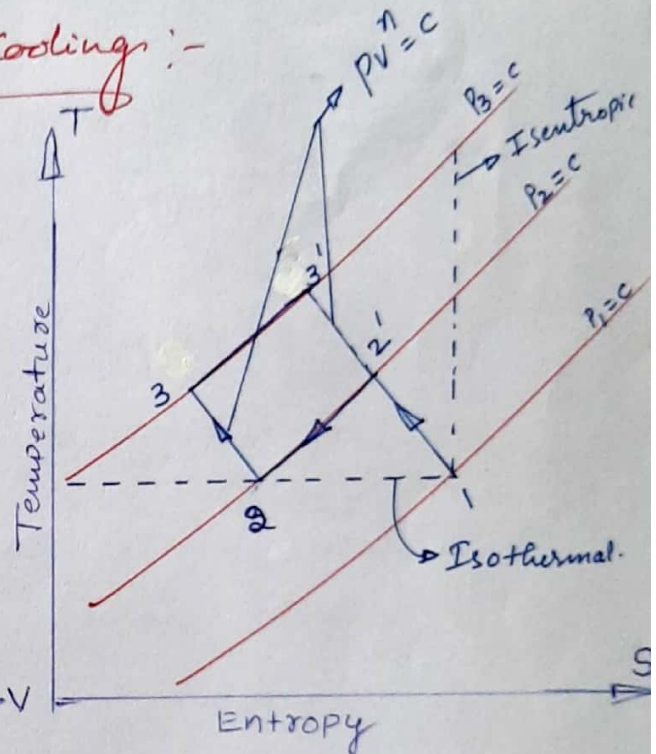


Fig → T-S diagram

- When the temperature of air leaving the intercooler T_3 , is equal to original atmospheric air temp T_1 , then the intercooling is known as perfect Intercooling.
- Cycle 613'4 is that of single stage compressor.
- cycles 612'5 & 52'3'4 are that of 2-stage compressor without inter cooling
- cycles 612'5 & 5234 are that of 2-stage compressor with perfect intercooling.

Case-I. Single Stage Compressor :-

As earlier stated cycle 613'4 is that of single stage compressor.

$$\text{For this cycle work done } W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_3}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$\text{Delivery temp } T_3 = T_1 \left(\frac{P_3}{P_1} \right)^{\frac{n-1}{n}}$$

Case-II (a) 2-stage Compressor without Inter cooler :-

612'5 - Low pressure cycle

52'3'4 - High pressure cycle

For this cycle

$$\text{work done } W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2'}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} P_2 V_2 \left[\left(\frac{P_3'}{P_2'} \right)^{\frac{n-1}{n}} - 1 \right]$$

$P_2' = P_2, P_3' = P_3.$

This workdone is also equal to workdone in case I.
Because of no intercooling

(b) 2-stage compressor with perfect Intercooling :-

612'5 - Low pressure cycle

5234 - High pressure cycle

For this cycle

$$\text{workdone } W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2'}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} P_2 V_2 \left[\left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 1 \right]$$

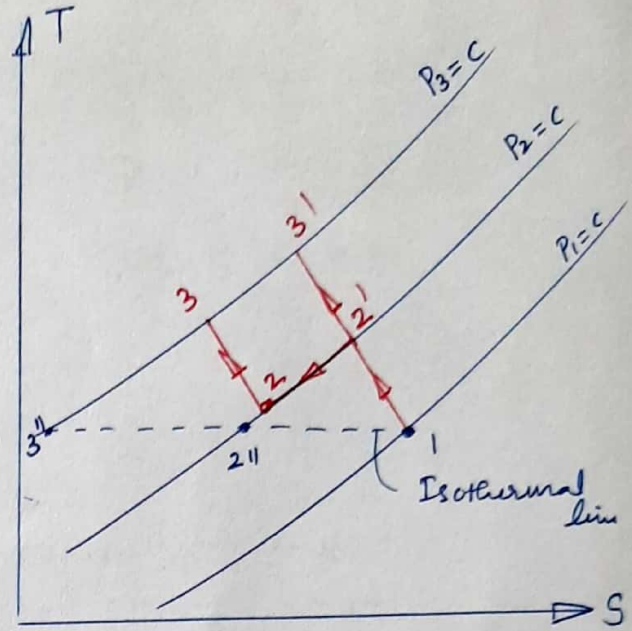
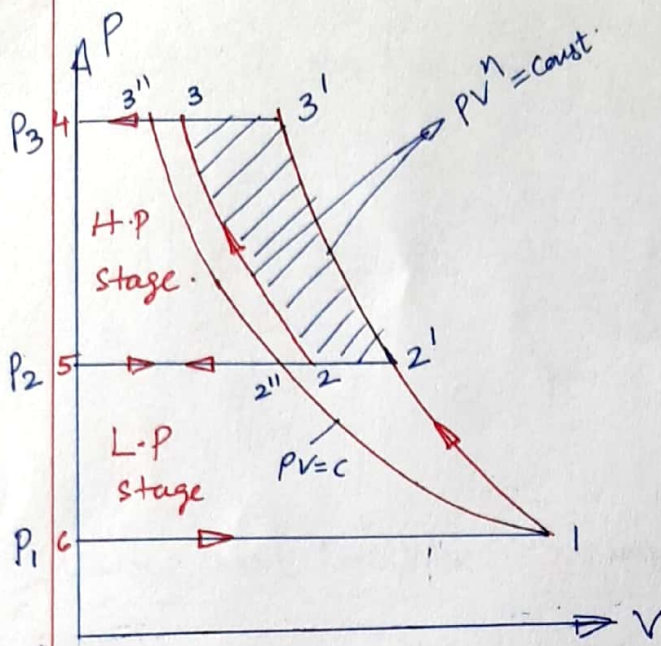
But Point 2' & 1' are on isothermal line

$$P_1 V_1 = P_2 V_2$$

$$\therefore W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 2 \right]$$

→ The area shaded 2'233' which is the work saving which occurs as a result of using intercooler.

2-stage Air Compressor with Imperfect Cooling:-



$$\text{Work done} = \frac{\eta}{\eta-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\eta-1}{\eta}} - 1 \right] + \frac{\eta}{\eta-1} P_2 V_2 \left[\left(\frac{P_3}{P_2} \right)^{\frac{\eta-1}{\eta}} - 1 \right]$$

$$W = \frac{\eta}{\eta-1} \left[P_1 V_1 \left(\left(\frac{P_2}{P_1} \right)^{\frac{\eta-1}{\eta}} - 1 \right) + P_2 V_2 \left(\left(\frac{P_3}{P_2} \right)^{\frac{\eta-1}{\eta}} - 1 \right) \right] \because P_2' = P_2$$

Minimum work required for 2-Stage Reciprocating Air Compressor:-

We know that for 2-stage reciprocating air compressor,

With perfect intercooling

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 2 \right]$$

If the intake pressure P_1 and delivery pressure P_3 are fixed, then least value of intermediate pressure P_2 may be obtained by differentiating the above equation

wrt P_2 and equate to zero. At this pressure P_2 , work required is minimum.

$$\frac{dW}{dP_2} = 0 \Rightarrow \frac{d}{dP_2} \left[\frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 2 \right] \right] = 0$$

$$\text{Let } \frac{n-1}{n} = a$$

$$\frac{d}{dP_2} \left[\left(\frac{P_2}{P_1} \right)^a + \left(\frac{P_3}{P_2} \right)^a - 2 \right] = 0$$

$$P_1^{-a} \cdot a P_2^{a-1} + P_3^a \cdot -a P_2^{-a-1} - 0 = 0$$

$$\cancel{P_1^{-a}} P_2^{a-1} = + \cancel{P_3^a} P_2^{-a-1}$$

$$\frac{P_2^{a-1}}{P_2^{-a-1}} = \frac{P_3^a}{P_1^{-a}}$$

$$P_2^{a-1+a+1} = (P_3 P_1)^a$$

$$P_2^{2a} = (P_3 P_1)^a \Rightarrow P_2^2 = P_1 P_3 \Rightarrow P_2 = \sqrt{P_1 P_3}$$

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \left(\frac{P_3}{P_1} \right)^{\frac{1}{2}}$$

Substitute $\frac{P_3}{P_2} = \frac{P_2}{P_1}$ in workdone to get $W_{\min}$

$$W_{\min} = \frac{\eta}{\eta-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\eta-1}{\eta}} + \left(\frac{P_2}{P_1} \right)^{\frac{\eta-1}{\eta}} - 2 \right]$$

$$= \frac{2\eta}{\eta-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\eta-1}{\eta}} - 1 \right]$$

= 2x Work required for each stage.

Substitute $\frac{P_2}{P_1} = \left(\frac{P_3}{P_1} \right)^{\frac{1}{2}}$ in the above eqn.

$$W = 2 \times \frac{\eta}{\eta-1} P_1 V_1 \left[\left(\frac{P_3}{P_1} \right)^{\frac{\eta-1}{2\eta}} - 1 \right]$$

Similarly for 3-s compressor

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = \left(\frac{P_4}{P_1} \right)^{\frac{1}{3}} \quad \&$$

Minimum work for 3-stage compressor

$$W = 3 \times \frac{\eta}{\eta-1} P_1 V_1 \left[\left(\frac{P_4}{P_1} \right)^{\frac{\eta-1}{3\eta}} - 1 \right]$$

NOTE:-

In general

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = \dots = \frac{P_{q+1}}{P_q} = \left(\frac{P_{q+1}}{P_1} \right)^{\frac{1}{q}}$$

$$W = q \times \frac{\eta}{\eta-1} P_1 V_1 \left[\left(\frac{P_{q+1}}{P_1} \right)^{\frac{\eta-1}{q\eta}} - 1 \right]$$

Heat Rejected in a Reciprocating Air Compressor:-

The total heat rejected in a reciprocating air compressor is the sum of heat rejected during polytropic compression per kg and heat rejected in the intercooler per kg of air.

W.K.T

Heat rejected during polytropic compression per kg

$$\begin{aligned} \text{of air } q_1 &= \frac{\gamma - n}{\gamma - 1} \times \text{work done} \\ &= \frac{\gamma - n}{\gamma - 1} \times \frac{R(T_2' - T_1)}{(n - 1)} \end{aligned}$$

$$C_p - C_v = R \quad C_p / C_v = \gamma$$

$$\frac{C_p - C_v}{C_v} = \frac{R}{C_v}$$

$$\gamma - 1 = R / C_v \Rightarrow R = C_v(\gamma - 1)$$

$$q_1 = \frac{\gamma - n}{\gamma - 1} \cdot C_v(\gamma - 1) \cdot \frac{(T_2' - T_1)}{n - 1}$$

$$q_1 = C_v \cdot \frac{\gamma - n}{n - 1} (T_2' - T_1)$$

Heat rejected in intercooler per kg of air,

$$q_2 = C_p(T_2' - T_2)$$

$$\therefore \text{Total heat rejected } q = q_1 + q_2$$

$$q = C_v \cdot \frac{\gamma - n}{n - 1} (T_2' - T_1) + C_p(T_2' - T_2)$$

for perfect intercooling $T_2 = T_1$

$$q = (T_2' - T_1) \left[\left(\frac{\gamma - n}{n - 1} \right) C_v + C_p \right]$$

Free Air Delivery :-

- It is the amount of atmospheric air that can be sucked by the compressor during suction at 1 atm pressure, 20°C, 100% dry air when the compressor motor running at 100% of rated R.P.M.
- FAD is an important purchasing parameter and it measures the capacity of compressor in terms of air-flow it handle.

Efficiencies of Compressor :-

① Isothermal efficiency :-

In a compressor the work is minimum when the compression follows isothermal process. The ratio of isothermal work to actual work is called isothermal efficiency.

$$\eta_{iso} = \frac{\text{Isothermal work}}{\text{Actual work}}$$

$$\textcircled{2} \quad \eta_{vol} = \frac{\text{Effective swept volume}}{\text{Swept volume}} = \frac{V_1 - V_4}{V_1 - V_3}$$

$$\textcircled{3} \quad \eta_{mech} = \frac{\text{Indicated power of a compressor}}{\text{Shaft power}}$$

pg-11

After cooling in Reciprocating Air Compressor:-

→ The Compressed air discharged from air compressor is hot.

Compressed air at this temperature contains large quantity of water in the vapour form.

→ Due to after cooler water vapour can be condensed into water.

Advantages:-

- 1) To reduce final temperature of air thus tank size can be reduced & greater amount of air can be stored for the same size
- 2) It increases volumetric efficiency
- 3) Reduces risk of fire.
- 4) protect the equipment from excess heat.

Volumetric Efficiency :-

The volumetric efficiency of a compressor is defined as ratio of free air delivery to the displacement of the compressor. or

It is the ratio of effective swept volume to the swept volume.

$$\eta_{vol} = \frac{\text{Effective swept vol}}{\text{Swept vol}} = \frac{V_1 - V_4}{V_1 - V_3}$$

→ Clearance ratio is defined as clearance vol to the swept volume.

$$\text{clearance ratio} = K = \frac{V_c}{V_1 - V_3} = \frac{V_c}{V_s}$$

$$\eta_{vol} = \frac{V_1 - V_4}{V_1 - V_3} = \frac{(V_1 - V_3) + (V_3 - V_4)}{V_1 - V_3} = 1 + \frac{V_3 - V_4}{V_1 - V_3}$$

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3} = 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3} \left(\frac{V_3}{V_3} \right)$$

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_3}{V_1 - V_3} \left(\frac{V_4}{V_3} \right)$$

$$P_1 V_1^\eta = P_2 V_2^\eta$$

$$\left(\frac{P_2}{P_1} \right)^{1/\eta} = \frac{V_1}{V_2}$$

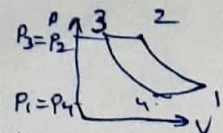
$$\eta_{vol} = 1 + K - K \left(\frac{V_4}{V_3} \right)$$

$$\eta_{vol} = 1 + K - K \left(\frac{P_3}{P_4} \right)^{1/\eta}$$

$$= 1 + K - K \left(\frac{P_3}{P_1} \right)^{1/\eta} = 1 + K - K \left(\frac{V_1}{V_2} \right)$$

$$P_3 V_3^\eta = P_4 V_4^\eta$$

$$\left(\frac{P_3}{P_4} \right)^{1/\eta} = \frac{V_4}{V_3}$$



$$\therefore \eta_{vol} = 1 + K - K \left(\frac{V_1}{V_2} \right)$$

→ In actual practice air that is sucked in during suction stroke gets heated up while passing through the hot valves and coming in contact with hot cylinder walls.

Thus ambient conditions are different from conditions obtained at state 1.

P_{amb} = Pr of ambient air

T_{amb} = Temp of ambient air

$$\frac{P_{amb} \cdot V_{amb}}{T_{amb}} = \frac{P_1 (V_1 - V_4)}{T_1}$$

$$\Rightarrow V_{amb} = \frac{P_1}{T_1} \cdot \frac{T_{amb}}{P_{amb}} \cdot (V_1 - V_4)$$

Volumetric efficiency refer to ambient conditions may be

written as $\eta_{vol(amb)} = \frac{V_{amb}}{V_1 - V_3} = \frac{P_1}{T_1} \cdot \frac{T_{amb}}{P_{amb}} \left(\frac{V_1 - V_4}{V_1 - V_3} \right)$

$$\eta_{vol(amb)} = \frac{P_1}{T_1} \cdot \frac{T_{amb}}{P_{amb}} \left(1 + K - K \left(\frac{V_1}{V_2} \right) \right)$$

or

$$\eta_{vol(amb)} = \frac{P_1}{T_1} \cdot \frac{T_{amb}}{P_{amb}} \left(1 + K - K \left(\frac{P_2}{P_1} \right)^{\frac{1}{\gamma}} \right)$$

④ A single stage double acting air compressor is required to deliver 14 m^3 air per min measured at 1.013 bar and 15°C . The delivery pressure is 7 bar and speed is 300 rpm take clearance volume as 5% of swept volume and index n is 1.3 . calculate (i) swept volume of cylinder (ii) delivery temperature (iii) Indicated power.

so $V_1 - V_4 = 14 \text{ m}^3/\text{min} = \frac{14}{600} \text{ m}^3 = 0.023 \text{ m}^3$

$P_1 = 1.013 \text{ bar} = 1.013 \times 10^5 \text{ N/m}^2$

$T_1 = 15^\circ\text{C} = 288 \text{ K}$

$P_2 = 7 \text{ bar} = 7 \times 10^5 \text{ N/m}^2$

$N = 300 \text{ rpm}$

$V_c = \frac{5}{100} V_s = 0.05 V_s$

$n = 1.3$

$V_s = V_1 - V_3 = V_1 - V_c$

$V_s = V_1 - 0.05 V_s$

$V_1 = 1.05 V_s \rightarrow \textcircled{1}$

$P_3 V_3^n = P_4 V_4^n$

$P_4 = P_1, P_3 = P_2$

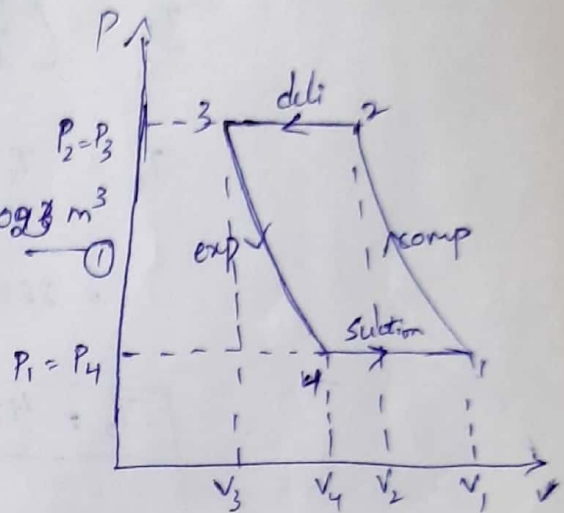
$V_3 = V_c$

$\left(\frac{V_4}{V_c}\right)^n = \left(\frac{P_2}{P_1}\right)^{1/n}$

$V_4 = V_c \left(\frac{P_2}{P_1}\right)^{1/n}$

$= 0.05 V_s \left(\frac{7}{1.013}\right)^{1/1.3}$

$V_4 = 0.22 V_s \rightarrow \textcircled{2}$



② & ③ in ①

$$1.05V_3 - 0.22V_3 = 0.023$$

$$V_3(1.05 - 0.22) = 0.023$$

$$\therefore V_3 = 0.027 \text{ m}^3$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$
$$= 288 \left(\frac{7}{1.013} \right)^{\frac{1.3-1}{1.3}}$$

$$\therefore T_2 = 450 \text{ K}$$

$$\text{work done} = \frac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.3}{1.3-1} \times 1.013 \times 10^5 (0.023) \left[\left(\frac{7}{1.013} \right)^{\frac{1.3-1}{1.3}} - 1 \right]$$

$$\therefore W = 5675.79 \text{ N-m}$$

$$P = \frac{W \times N_w}{60} \quad \text{--- } 2N$$

$$= \frac{5675.79 \times 2 \times 300}{60}$$

$$\therefore P = 56.75 \text{ kW}$$

⑤ A single stage double acting air compressor has a Free Air Delivery (F.A.D) of $14 \text{ m}^3/\text{min}$ measured at 1.013 bar and 15°C . The pressure and temperature in cylinder during induction are 0.95 bar and 32°C . The delivery pressure is 7 bar and index of compression and expansion $n = 1.3$. The clearance is 5% of swept volume. calculate (i) Indicated power required (ii) volumetric efficiency.

$$\underline{\text{Sol}} \quad V_1 - V_4 = 14 \text{ m}^3/\text{min} = \frac{14}{60} \text{ m}^3 = 0.233 \text{ m}^3$$

$$P_1 = 0.95 \text{ bar} = 0.95 \times 10^5 \text{ N/m}^2$$

$$T_1 = 32^\circ\text{C} = 305 \text{ K}$$

$$P_2 = 7 \text{ bar} = 7 \times 10^5 \text{ N/m}^2$$

$$n = 1.3$$

$$V_c = 0.05 V_s \Rightarrow \frac{V_c}{V_s} = 0.05 = K$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

$$T_2 = 305 \left(\frac{7}{0.95} \right)^{\frac{1.3-1}{1.3}}$$

$$\boxed{\therefore T_2 = 483.57 \text{ K}}$$

$$W = \frac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.3}{1.3-1} \times 0.95 \times 10^5 \times \left(\frac{14}{60} \right) \left[\left(\frac{7}{0.95} \right)^{\frac{1.3-1}{1.3}} - 1 \right]$$

$$W = 33.74 \times 10^5 \text{ N-m/min}$$

$$\text{power} = \frac{33.74 \times 10^5}{60} = 56.23 \text{ kW}$$

$$\eta_{vol} = 1 + K - K \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \left[\frac{P_a}{P_a} \cdot \frac{T_a}{T_1} \right]$$

$$= 1 + 0.05 - 0.05 \left(\frac{7}{0.95} \right)^{\frac{1}{1.3}} \left[\frac{0.95 \times 10^5}{1.013 \times 10^5} \times \frac{288}{305} \right]$$

$$= 0.844$$

$$\therefore \eta_{vol} = 84.4\%$$

- ⑥ A single stage single acting air compressor delivers 0.6 Kg of air per min at 6 bar. The temperature and pressure at end of suction stroke are 30°C and 1 bar. The bore and stroke of compressor are 100mm and 150mm respectively. The clearance volume is 3% of swept volume and $n = 1.3$. find
- volumetric efficiency of compressor
 - power required to drive the compressor if mechanical efficiency is 85%
 - speed of compressor in rpm.

sol.

$$m = 0.6 \text{ Kg/min}$$

$$P_2 = 6 \text{ bar}$$

$$T_1 = 30^\circ\text{C}$$

$$P_1 = 1 \text{ bar}$$

$$d = 100 \text{ mm}, L = 150 \text{ mm}$$

$$V_c = 0.03 V_s \Rightarrow K = \frac{V_c}{V_s} = 0.03$$

$$n = 1.3$$

$$\eta_{vol} = 1 + K - K \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}}$$

$$= 1 + 0.03 - 0.03 \left(\frac{6}{1} \right)^{\frac{1}{1.3}}$$

$$= 0.910$$

$$\eta_{vol} = 91\%$$

ii) work done = $\frac{n}{n-1} P_1 (V_1 - V_4) \left(\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right)$

$$= \frac{n}{n-1} m R T_1 \left(\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right)$$

$$= \frac{1.3}{1.3-1} \times 0.6 \times 287 \times 303 \left(\left(\frac{6}{1} \right)^{\frac{1.3-1}{1.3}} - 1 \right)$$

$$W. = 1.01 \times 10^5 \text{ N-m/min}$$

$$I.P = \frac{1.01 \times 10^5}{60} = 1929.6 \text{ W} = 1.92 \text{ kW}$$

$$\eta_{mech} = \frac{I.P}{\text{shaft power}}$$

$$S.P = \frac{1929.6}{0.85} = 2270 \text{ W}$$

$$= 2.27 \text{ kW}$$

iii) speed of compressor

$$\eta_{vol} = \frac{FAD}{V_s}$$

$$F.A.D = P_1 (V_1 - V_4) = m R T_1$$

$$FAD = \frac{m R T_1}{P_1}$$

$$= \frac{0.6 \times 287 \times 303}{1 \times 10^5} = 0.52 \text{ m}^3/\text{min}$$

$$P = \frac{W \times N}{60}$$

$$\frac{60P}{W}$$

$$\frac{60 \times 1929.6}{1.01 \times 10^5}$$

$$\eta_{vol} = \frac{0.52}{\frac{\pi d^2}{4} \times N}$$

$$0.91 = \frac{0.52}{\frac{\pi}{4} \times 100^2 \times 150 \times N}$$

N

⑦ A single stage single acting compressor delivers $14 \text{ m}^3/\text{min}$ from 1 bar to 7 bar. The speed of compressor is 310 rpm. $n = 1.35$ and clearance volume is 5% of swept volume. Stroke length is 1.5 times the diameter. Assume the temperature and pressure of air at suction are same as atmospheric air. Determine the bore and stroke of compressor.

So $FAD = 14 \text{ m}^3/\text{min}$

$$P_1 = 1 \text{ bar}$$

$$P_2 = 7 \text{ bar}$$

$$N = 310 \text{ rpm}$$

$$n = 1.35$$

$$V_c = 5\% V_s \Rightarrow K = \frac{V_c}{V_s} = 0.05$$

$$L = 1.5D$$

$$\eta_{vol} = 1 + K - K \left(\frac{P_2}{P_1} \right)^{1/n}$$

$$= 1 + 0.05 - 0.05 \left(\frac{7}{1} \right)^{1/1.35}$$

$$= 0.838$$

$$\eta_{vol} = 83.8\%$$

$$\eta_{vol} = \frac{FAD}{V_s}$$

$$\eta_{vol} = \frac{FAD}{V_s}$$

$$V_s = \frac{FAD}{\eta_{vol}}$$

$$\left(\frac{\pi}{4} d^2 L\right) N = \frac{14}{0.838}$$

$$\frac{\pi}{4} \times d^2 \times 1.5d \times 310 = \frac{14}{0.838}$$

$$365.21 d^3 = 16.70$$

$$\therefore d = 0.35 \text{ m}$$

$$L = 1.5D$$

$$= 1.5 \times 0.35$$

$$\therefore L = 0.53 \text{ m}$$

⑤ Air at 103 KPa and 27°C is drawn LP cylinder of 2-stage air compressor and is isentropically compressed to 700 KPa. The air is then cooled at constant pressure to 37°C in an inter cooler and is then again compressed isentropically to 4 MPa in HP cylinder and then delivered at that pressure. Determine the power required to run the compressor if it has to deliver 30 m³ of free air per hour at inlet condition.

$$P_1 = 103 \text{ KPa}$$

$$T_1 = 27^\circ \text{C}$$

$$P_2 = 700 \text{ KPa}$$

$$T_2' = 37^\circ \text{C}$$

$$P_3 = 4 \text{ MPa}$$

$$FAD = V_1 = 30 \text{ m}^3/\text{hr}$$

Isentropically, $\gamma = 1.4$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = 300 \times \left(\frac{700}{103}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 518.7 \text{ K}$$

$$\frac{T_3}{T_2'} = \left(\frac{P_3}{P_2'}\right)^{\frac{\gamma-1}{\gamma}} = T_3 = 310 \times \left(\frac{4000}{700}\right)^{\frac{1.4-1}{1.4}}$$

$$T_3 = 510.07 \text{ K}$$

$$\begin{aligned} \text{Work done} &= \frac{\gamma}{\gamma-1} [P_2 V_2 - P_1 V_1] + \frac{\gamma}{\gamma-1} [P_3 V_3 - P_2' V_2'] \\ &= \frac{\gamma}{\gamma-1} \left\{ mR(T_2 - T_1) + \frac{\gamma}{\gamma-1} mR(T_3 - T_2') \right\} \end{aligned}$$

$$m = \frac{P_1 V_1}{RT_1}$$

$$= \frac{103 \times 10^3 \times 30}{287 \times 300}$$

$$= 9.96 \times 10^3$$

$$= \frac{\gamma}{\gamma-1} mR[(T_2 - T_1) + (T_3 - T_2')]$$

$$= \frac{1.4}{1.4-1} \times 9.96 \times 10^3 \times 287 [(518.7 - 300) + (510.07 - 310)]$$

$$W = 4.18 \text{ kJ/s}$$

⑨ A 2-stage single acting compressor takes in air at the rate of $0.2 \text{ m}^3/\text{s}$, the intake pressure and temperature are 0.1 mPa and 16°C . The air is compressed to a final pressure of 0.7 mPa . The intermediate pressure is ideal and intercooling is perfect. The compression $n = 1.25$ and $N = 600 \text{ rpm}$. neglecting the clearance. Determine 1) intermediate pressure

2) total volume of each cylinder 3) power required to drive the compressor. 4) heat rejection in intercooler.

So $V_1 = \text{FAD} = 0.2 \text{ m}^3/\text{s}$

$$P_1 = 0.1 \text{ MPa}$$

$$P_3 = 0.7 \text{ MPa}$$

$$T_1 = 16^\circ\text{C}$$

$$n = 1.25$$

$$N = 600 \text{ rpm}$$

$$C_p = 1.005 \text{ kJ/kg K}$$

$$R = 0.287 \text{ kJ/kg K} \quad \text{for perfect intercooling}$$

$$T_2 = T_1, \quad (1) \quad P_2 = \sqrt{P_1 P_3}$$

$$\therefore P_2 = \sqrt{0.1 \times 0.7} = 0.26 \text{ MPa}$$

(2) $V_1 = \text{Vol of L.P cyl}$

$V_2 = \text{Vol of HP cyl.}$

$$V_1 = \frac{0.2 \text{ m}^3/\text{s}}{\frac{600}{60}} = 0.02 \text{ m}^3$$

$$T_1 = T_2 \text{ (100\% I.C)}$$

$$P_1 V_1 = P_2 V_2$$

$$V_2 = \frac{P_1 V_1}{P_2} = \frac{0.1 \times 0.02}{0.26}$$

$$\therefore V_2 = 0.0076 \text{ m}^3$$

(3) $W = \frac{2n}{n-1} P_1 V_1 \left[\left(\frac{P_3}{P_1} \right)^{\frac{n-1}{2n}} - 1 \right]$

$$= \frac{2 \times 1.25}{1.25-1} \times 100 \times 0.02 \left[\left(\frac{0.7}{0.1} \right)^{\frac{1.25-1}{2 \times 1.25}} - 1 \right]$$

$$W = 4296.28 \text{ N-m} \Rightarrow 42.9 \times 10^3 \text{ N-m/s}$$

⑥ Heat rejected

$$\frac{T_2'}{T_1} = \left(\frac{P_2'}{P_1}\right)^{\frac{n-1}{n}}$$

$$T_2' = 289 \times \left(\frac{0.26}{0.1}\right)^{\frac{1.25-1}{1.25}}$$

$$T_2' = 349.85 \text{ K}$$

$$m = \frac{P_1 V_1}{R T_1} = \frac{0.1 \times 10^6 \times 0.02}{0.287 \times 289} = \frac{24.11}{0.287 \times 289} \text{ kg/s}$$

$$\therefore Q = m c_p (T_2 - T_2')$$

$$= \frac{24.11}{0.287 \times 289} \times 0.005 (289 - 349.85)$$

$$= -0.146$$

⑩ A 2-stage air compressor with complete inter cooling delivers air to the mains at a pressure of 30 bar. The suction conditions are 1 bar and ^{27°C} 15°C. If the both the cylinders have same stroke, find the ratio of cylinder diameters, assume the index for compression is 1.3

So

$$P_3 = 30 \text{ bar}$$

$$P_1 = 1 \text{ bar}$$

$$T_1 = 15^\circ \text{C}$$

$$L_1 = L_2$$

$$n = 1.3$$

for complete intercooling

$$T_1 = T_2 = 288 \text{ K}$$

$$P_2 = \sqrt{P_1 P_3} = \sqrt{30 \times 1}$$

$$\therefore P_2 = 5.47 \text{ bar}$$

$$V_1 = \text{Vol of L.P}$$

$$V_2 = \text{Vol of H.P}$$

$$\frac{V_1}{V_2} = \frac{\frac{\pi}{4} d_1^2 L_1 N_1}{\frac{\pi}{4} d_2^2 L_2 N_2} \Rightarrow \frac{d_1}{d_2} = \sqrt{\frac{V_1}{V_2}} \quad \text{--- (1)}$$

$$P_1 V_1^n = P_2 V_2'^n$$

$$\frac{1-2'}{2'-2} \left(\frac{V_1}{V_2'} \right) = \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \quad \text{--- (2)}$$

$$\frac{V_1'}{T_2'} = \frac{V_2}{T_2} \Rightarrow \frac{V_2'}{V_2} = \frac{T_2'}{T_2} \quad \text{--- (3)}$$

$$\frac{T_2'}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

$$\frac{T_2'}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

$$\Rightarrow T_2' = 288 \times \left(\frac{5.47}{1} \right)^{\frac{1.3-1}{1.3}} = 1.249$$

$$T_2' = 426.28 \text{ K}$$

$$T_2 =$$

$$\frac{V_1}{V_2} = \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \cdot \frac{T_2'}{T_2}$$

$$\frac{V_1}{V_2} = \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \cdot \frac{T_2'}{T_2}$$

$$= \left(\frac{5.47}{1} \right)^{\frac{1}{1.3}} \times \frac{426.28}{288}$$

$$\frac{V_1}{V_2} = 5.46$$

$$\frac{d_1}{d_2} = \sqrt{5.46}$$

$$\therefore \frac{d_1}{d_2} = 2.33$$

11. In a single acting 2-stage reciprocating air compressor 4.5 kg of air per min are compressed from 1.013 bar and 15°C through a pressure ratio of 9 to 1. Both the stages have same pressure ratio and law of compression ratio $PV^{1.3} = c$. If the intercooling is complete. calculate the indicated power, cylinder swept volume, assume that the clearance volume of both the stages are 5% of their swept volumes and speed of compressor is 300 rpm.

So $m = 4.5 \text{ kg/min}$

$P_s = 1.013 \text{ bar}$

$T_s = 15^\circ\text{C}$

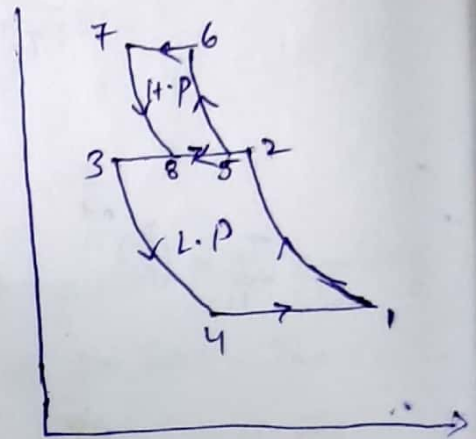
Pressure ratio $= \frac{9}{1} = \frac{P_d}{P_s}$

$n = 1.3$

$\Rightarrow P_d = 9 P_s$

$V_c = 0.05 V_s$

$N = 300 \text{ rpm}$



Pressure ratio is same for both stages $\Rightarrow \frac{P_i}{P_s} = \frac{P_d}{P_i} = 3$

Inter cooling complete $= P_i = \sqrt{P_s P_d}$

$P_i^2 = P_s \cdot P_d$

$P_d = 9 P_s$

$P_i^2 = 9 P_s^2 \Rightarrow \boxed{P_i = 3 P_s}$

$$\frac{T_i}{T_s} = \left(\frac{P_i}{P_s}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_i = T_s \left(\frac{P_i}{P_s}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_i}{T_s} = \left(\frac{P_i}{P_s}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_i = 288 \times \left(\frac{3P_s}{P_s}\right)^{\frac{1.3-1}{1.3}}$$

$$\therefore T_i = 371.10 \text{ K}$$

$$\frac{T_d}{T_s} = \left(\frac{P_d}{P_s}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_d}{T_i} = \left(\frac{P_d}{P_i}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_d}{T_i} = \left(\frac{P_d}{P_i}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_d = 371.10 \times (3)^{\frac{1.3-1}{1.3}}$$

$$\frac{9.3}{3P_s}$$

$$T_d = 478.18 \text{ K}$$

$$\text{Work done} = \frac{n}{n-1} P_s V_s \left[\left(\frac{P_i}{P_s}\right)^{\frac{\gamma-1}{\gamma}} - 1 + \left(\frac{P_d}{P_i}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$W/d = \frac{2n}{n-1} P_s V_s \left[\left(\frac{P_d}{P_s}\right)^{\frac{\gamma-1}{2n}} - 1 \right]$$

$$P_s V_s = mRT_s$$

$$= \frac{2 \times 1.3}{1.3-1} \times 1.013 \times 10^5 \times 3.67 \left[\left(\frac{9P_s}{P_s}\right)^{\frac{1.3-1}{2 \times 1.3}} - 1 \right]$$

$$1.013 \times 10^5 \times V_s = 4.5 \times 287 \times 288$$

$$V_s = \frac{4.5 \times 287 \times 288}{1.013 \times 10^5}$$

$$= 929747.22 \text{ N-m/min}$$

$$V_s = 3.67 \text{ m}^3/\text{min}$$

$$\text{power} = 929747.22/60$$

$$\therefore \text{power} = 15.49 \text{ kW}$$

$$1 + 0.05 - 0.05 \left(\frac{3P_s}{P_s}\right)^{\frac{1.3-1}{1.3}}$$

$$= 0.985$$

→

$$K = \frac{V_c}{V_s} = 0.05$$

(L.P)

η_{vol}

$$= 1 + K - K \left(\frac{P_i}{P_s}\right)^{\frac{\gamma-1}{\gamma}} = 0.985 \Rightarrow 98.5\%$$

(H.P)

η_{vol}

$$= 1 + K - K \left(\frac{P_d}{P_i}\right)^{\frac{\gamma-1}{\gamma}} = 0.985 = 98.5\%$$

$$\eta_{vol} = \frac{L.P}{V_s}$$

$$0.985 = \frac{3.68}{V_s}$$

$$V_s = \frac{3.68}{0.985}$$

$$V_s = 3.73 \text{ m}^3/\text{min}$$

$$\therefore V_s = 0.0124 \text{ m}^3$$

$$V_s(HP) = ?$$

$$P_3 \overbrace{(V_1 - V_4)}^{FAD} = mRT_s$$

$$FAD = \frac{mRT_s}{P_3}$$

$$= \frac{4.5 \times 287 \times 288}{1.013 \times 10^5}$$

$$FAD = 3.68 \text{ m}^3/\text{min}$$

$$\eta_{vol} = 0.985$$

$$P_1 V = mRT_1$$

$$V = \frac{mRT_1}{\frac{P_1}{3P_3}} = \frac{4.5 \times 287 \times 288}{3 \times 1.013 \times 10^5} = 1.59 \text{ m}^3/\text{min}$$

$$\eta_{vol} = \frac{FAD}{V_s} \Rightarrow V_s = \frac{1.59}{0.985} = 1.61 \text{ m}^3/\text{min}$$

$$\therefore V_s = 0.0053 \text{ m}^3$$

(12) A single acting 2-stage compressor with complete inter cooling delivered 10.5 kg/min at 16 bar. The suction occurs at 1 bar and 27°C, n is 1.3, N is 440 rpm. calculate

- power required to drive the compressor
- isothermal efficiency
- free air delivery
- heat transferred in intercooler
- calculate swept volume and clearance volume of each cylinder. If clearance ratio's for L.P and H.P are 0.04 & 0.06 respectively.

Sp

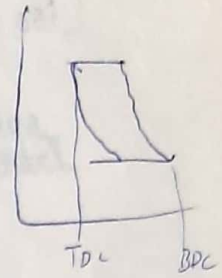
$$m = 10.5 \text{ Kg/min}$$

$$P_d = 16 \text{ bar}$$

$$P_s = 1 \text{ bar}$$

$$T_s = 27^\circ\text{C}, \quad n = 1.3, \quad N = 440 \text{ rpm},$$

$$K_{LP} = 0.04, \quad K_{HP} = 0.06.$$



$$P_i = \sqrt{P_s P_d}$$

$$P_i = 4 \text{ bar}$$

$$\frac{T_i}{T_s} = \left(\frac{P_i}{P_s} \right)^{\frac{n-1}{n}} \Rightarrow T_i = 300 \times \left(\frac{4}{1} \right)^{\frac{1.3-1}{1.3}}$$

$$T_i = 413.10 \text{ K}$$

$$\frac{T_d}{T_i} = \left(\frac{P_d}{P_i} \right)^{\frac{n-1}{n}} = T_d = 413.10 \times \left(\frac{16}{4} \right)^{\frac{1.3-1}{1.3}}$$

$$T_d = 568.84 \text{ K}$$

$$\text{Work done} = \frac{2n}{n-1} P_s V_s \left[\left(\frac{P_d}{P_s} \right)^{\frac{n-1}{2n}} - 1 \right]$$

$$= \frac{2 \times 1.3}{1.3-1} \times 1 \times 10^5 \times 9.04 \left[\left(\frac{16}{1} \right)^{\frac{1.3-1}{2 \times 1.3}} - 1 \right]$$

$$P_s V_s = m R T_s$$

$$1 \times 10^5 \times V_s = 10.5 \times 287 \times 300$$

$$V_s = 9.04 \text{ m}^3/\text{min}$$

$$W_d = 2953743.38 \text{ N-m/min}$$

$$\text{power} = \frac{2953743.38}{60}$$

$$P = 49.22 \text{ kW}$$

②

$$\eta_{iso} = \frac{\text{Isothermal work}}{\text{Indicated work}} \times 100$$

$$\frac{\text{Isothermal work}}{\text{Indicated work}} = P_3 V_3 \ln\left(\frac{P_1}{P_3}\right)$$

$$= 1 \times 10^5 \times 9.04 \ln\left(\frac{16}{1}\right)$$

$$= 2506420.20 \text{ N-m/min}$$

$$\eta_{iso} = \frac{2506420.20}{2953743.38} = 0.848$$

$$\therefore \eta_{iso} = 84.8\%$$

③ Free air delivery

$$P_3(V_1 - V_4) = mRT_3$$

$$V_1 - V_4 = \frac{mRT_3}{P_3}$$

$$= \frac{10.5 \times 287 \times 300}{1 \times 10^5}$$

$$\therefore \text{FAD} = 9.04 \text{ m}^3/\text{min}$$

④

$$\frac{T_2}{T_3} = \left(\frac{P_1}{P_3}\right)^{\frac{n-1}{n}}$$

$$T_2 = 300 \times \left(\frac{4}{1}\right)^{\frac{1.3-1}{1.3}} = 413.10 \text{ K}$$

$$Q = mC_p(T_3 - T_2)$$

$$= 10.5 \times 1005(300 - 413.10)$$

$$= -1193487.75 \text{ J/min}$$

$$= -19891 \text{ W} \Rightarrow 19.89 \text{ kW (-ve)}$$

$$\eta_{vol} = 1 + K - K \left(\frac{P_i}{P_s} \right)^{\frac{n-1}{n}}$$

$$= 1 + 0.04 - 0.04 \left(\frac{4}{1} \right)^{\frac{1.3-1}{1.3}}$$

$$= 0.923 \Rightarrow 92.3\%$$

$$V_{s(LP)} = \frac{\cancel{FAD}}{\cancel{FAD}} \frac{FAD}{\eta_{vol}} = \frac{9.04}{0.923} = 9.79 \text{ m}^3/\text{min}$$

$$\therefore V_{s(LP)} = 0.022 \text{ m}^3$$

$$\eta_{vol} = 1 + K - K \left(\frac{P_{d'}}{P_i} \right)^{\frac{1}{n}}$$

$$= 1 + 0.06 - 0.06 \left(\frac{16}{1} \right)^{\frac{1}{1.3}}$$

$$\eta_{vol} = 88.5\%$$

$$P_i \overset{FAD}{(V)} = mRT_s$$

$$V_{s(HP)} = 0.000$$

$$V = \frac{10.5 \times 287 \times 300}{4 \times 10^5}$$

$$\therefore V = 2.26 \text{ m}^3/\text{min}$$

$$V_s = \frac{2.26}{0.88} = 2.55 \text{ m}^3/\text{min}$$

$$= \frac{2.55}{400} \text{ m}^3$$

$$\therefore V_s = 0.00637 \text{ m}^3$$

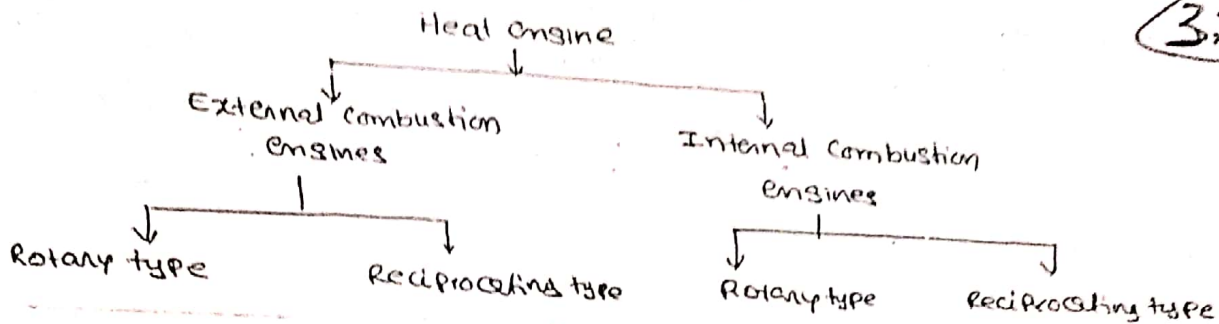
IC ENGINES

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Heat engine: It is a device which is used to convert Heat energy into mechanical energy



→ If the combustion of fuel takes place outside the cylinder, then engine is known as External combustion engine

example: steam engine, steam turbine

→ If the combustion of fuel takes place inside the cylinder, then engine is known as Internal combustion engine

example: Petrol engines, Diesel engines

Parts of an I.C engine:

<u>Component</u>	<u>Function</u>	<u>Material</u>
1. Cylinder block :	Main body of the engine It accommodates all other components.	Grey cast iron or Aluminium alloy
2. Cylinder :	It is the cylindrical barrel in which piston reciprocates	"
3. Cylinder head :	It accommodates valves & spark plug/ fuel injector	"
4. Piston :	It is the main reciprocating component in engine. It transmits gas force to the crank shaft through connecting rod	Cast iron or Aluminium alloy Cast steel

<u>Component</u>	<u>Function</u>	<u>Material</u>
5. Piston rings :	They provide a seal between Piston & cylinder. It prevents leakage of combustion gases in to crank case	Special Cast Iron alloy with chrome plating
6. connecting rod :	It converts the reciprocating motion of Piston in to rotary motion of crankshaft	Medium Carbon Steel & alloy steel
7. Gudgeon Pin/ Piston Pin/ Wrist pin	It connects the Piston & Connecting rod	steel
8. Crank pin :	It connects the Connecting rod & crankshaft	steel
9. crankshaft :	It transmits the power developed in the cylinder to driven shaft	Forged steel
10. cam shaft :	It is driven by crankshaft by means of timing gears. It operates inlet & outlet valves	Forged steel
11. Crank Case :	It serves as an enclosure for moving parts	Grey Cast Iron or Aluminium alloy
12. Valves :	To supply charge & To Exhaust combustion gases	Inlet valve - Nickel-chromium alloy steel Exhaust valve - Silicon-chrome steel

Basic Terminology

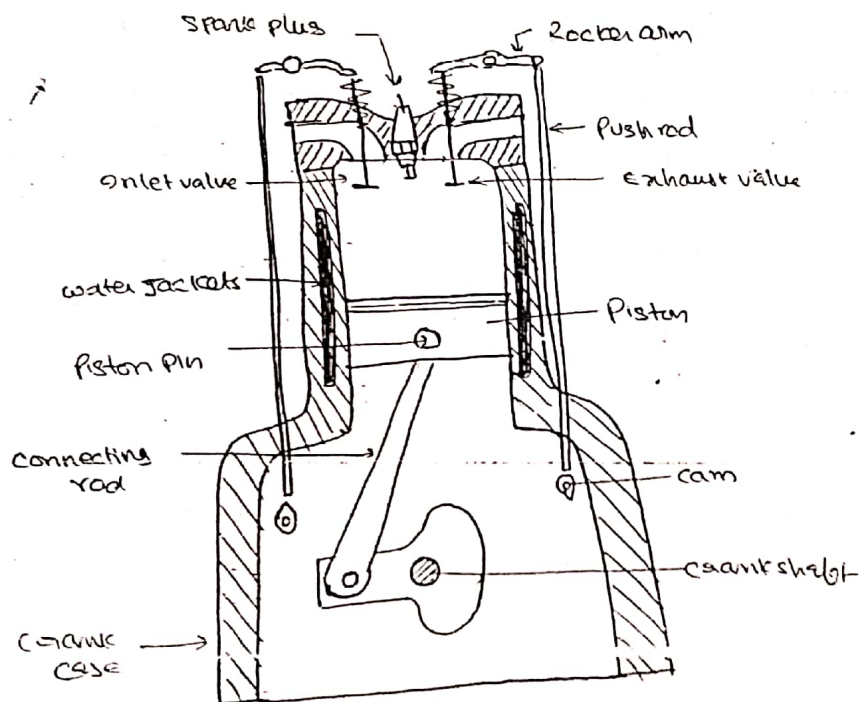
- Bore : Diameter of cylinder
- Stroke : The distance between TDC & BDC
- TDC : Top dead centre (Uppermost position)
- BDC : Bottom dead centre (Lower most position)

stroke volume / swept volume : the volume between TDC & BDC (V_s)

→ clearance volume : the volume between cylinder head & the piston face at the T.D.C (V_c)

→ compression ratio : it is the ratio of total cylinder volume to the clearance volume

$$\eta = \frac{V_c + V_s}{V_c}$$



components of an IC engine

classification of IC engines

I. Based on type of fuel used

- Petrol engines
- Diesel engines
- Gas engines

II. Based on Nature of cycle

- Otto cycle engines
- Diesel cycle engines

III. Based on no. of strokes

- Four stroke engines
- Two stroke engines

IV. Based on method of ignition

- Spark ignition (SI) engines
- Compression ignition (CI) engines

V Based on no. of cylinders — { single cylinder
multi cylinders

VI Based on method of cooling — { Air cooled
water cooled

VII Based on method of governing — { Quantity governing
Quality governing

VIII Based on arrangement of cylinders

Horizontal



Vertical



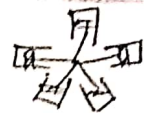
V-engine



In-line



radial



opposed piston



opposed cylinder



IX Based on application — { Stationary
Marine
Locomotive
Aircraft
Automobile

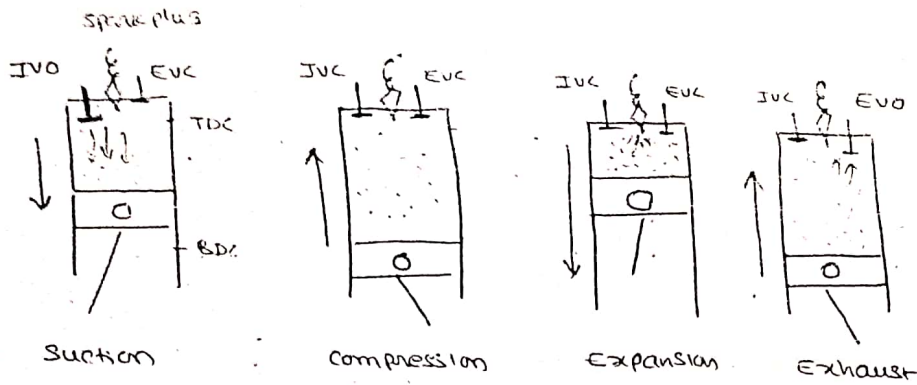
The operations required to produce power in the cylinder

1. suction
2. compression
3. Power (Expansion)
4. Exhaust

→ If all the above 4 operations are completed in 4-strokes of the piston then it is called 4-stroke engine

→ If all the above 4 operations are completed in 2-strokes of the piston then it is called 2-stroke engine

Working of 4-Stroke SI engine (Petrol engine)



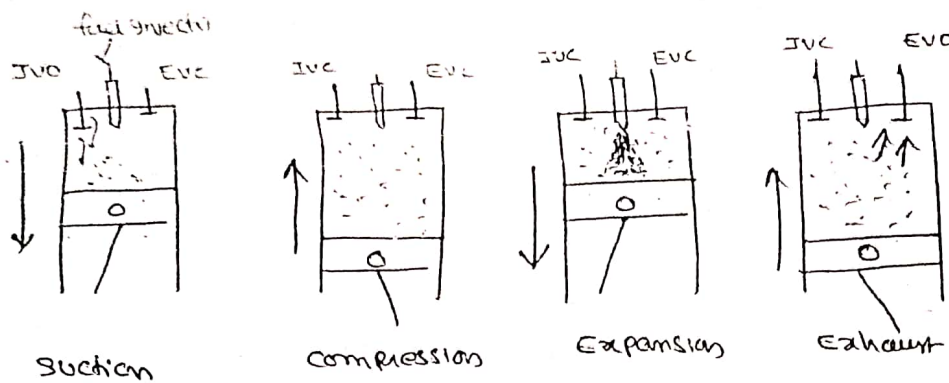
In 4-stroke engine all the operations are completed in 4-strokes of the piston or 2-revolutions of crankshaft. In this Air-fuel mixture is ignited by spark given by spark plug so that it is SI engine.

- 1. Suction stroke:** During the suction stroke inlet valve open and exhaust valve closed. piston moves from TDC to BDC. partial vacuum is created inside the cylinder. Air-fuel mixture from carburettor enters in to the cylinder.
- 2. Compression stroke:** Piston moves from BDC to TDC. Both inlet & outlet valves are closed. Charge (Air+fuel mixture) compressed to high pressure & temperature.
At the end of the compression stroke spark plug gives the spark. The charge burns and applies pressure on the piston.
- 3. Expansion stroke:** This is the power stroke. Piston moves from TDC to BDC (i.e. gases expands & produces power). Both inlet & exhaust valve are in closed position.
- 4. Exhaust stroke:** Piston moves from BDC to TDC. Inlet valve is closed position, Exhaust valve opens. The burnt gases forces out through Exhaust valve.

At the end of this stroke Exhaust valve closes & inlet valve opens. Again cycle repeated.

Working of 4 stroke CI engine (Diesel engine)

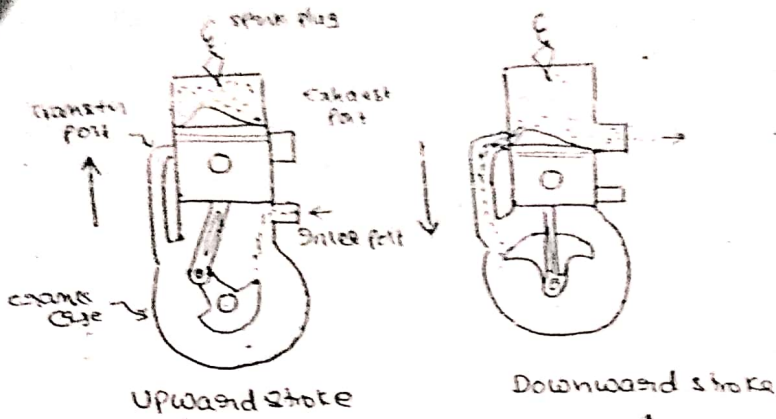
Working of



In 4-stroke engine all 4 operations are completed in 4 strokes of the piston or 2-revolutions of crankshaft. In this Air fuel mixture is ignited due to compression so that it is called C.I engine.

- 1. Suction stroke:** - piston moves from TDC to BDC. Inlet valve open & Exhaust valve closed. Only Air enters in to cylinder.
- 2. Compression stroke:** piston moves from BDC to TDC. Inlet & Exhaust valves are closed. The Air gets compressed. The pressure and temperature of Air increases.
At the end of compression stroke, fuel injector injects fuel in to the compressed air. Ignition occurs spontaneously due to temperature of compressed air.
- 3. power (Expansion) stroke:** the burnt gases apply force on the piston. piston moves from TDC to BDC. Inlet & Exhaust valve are in closed position.
- 4. Exhaust stroke:** Inlet valve is closed position. Exhaust valve open. piston moves from BDC to TDC. The burnt gases forces out through Exhaust valve.
At the end of this stroke Exhaust valve closes & Inlet valve open. Again cycle repeated.

Working of 2 stroke SI engine



In 2-stroke engine all 4 operations (suction, compression, power & exhaust) are completed in 2-strokes of the piston or 1 revolution of the crankshaft.

In this Air-fuel mixture is ignited by spark given by spark plug so that it is called ~~as~~ S.I engine.

1. Upward stroke:

The piston moves from BDC to TDC, covers the transfer port & exhaust port. Due to upward movement of the piston, a partial vacuum is created in the crankcase, and a new charge (Air + fuel mixture) is drawn in to the crankcase through the uncovered inlet port (i.e. suction).

The charge which is already present in the combustion chamber gets compressed by the piston (i.e. compression).

At the end of this 1st stroke the compressed charge is ignited by a spark given by the spark plug.

2. Downward stroke:

As soon as the charge is ignited the hot gases force the piston downwards, rotating the crankshaft thus doing the useful work. During this stroke inlet port is covered by the piston and the new charge is partially compressed in crankcase. Further downward movement of the piston uncovers first the exhaust port and then the transfer port, and hence the exhaust starts through the exhaust port. As soon as the transfer port opens the charge from crankcase enters in to combustion chamber. i.e. expansion & exhaust are completed in this 2nd stroke. Again piston starts moving from BDC to TDC, cycle repeats. (4)

Working of 2 stroke CI engine

→ On two stroke CI engine only Air is entering in to the crank case during suction. Fuel is injected by injector at the end of compression. The remaining operations are same as 2 stroke SI engine.

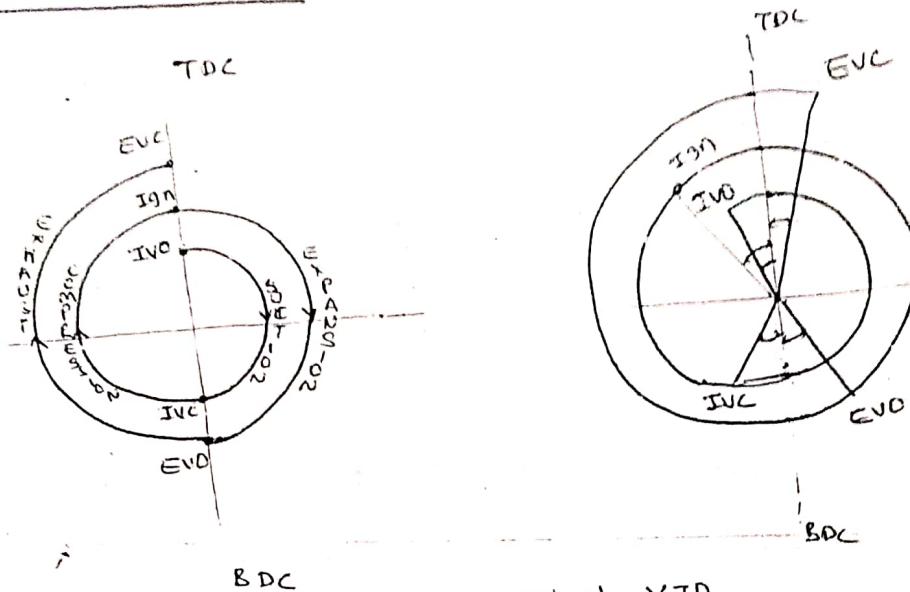
Differences between SI & CI engines

Criterion	SI	CI
1. Fuel used	Petrol, Gas	Diesel
2. Cycle	Otto	Diesel
3. suction of charge	Air + Fuel	only Air
4. Compression ratio	Maximum 10:1	16:1 to 22:1
5. Ignition of charge	by spark plug	self ignition due to compression
6. Thermal efficiency	Up to 25%	Up to 40%
7. Weight	low	Heavy
8. Cost	Initial cost low Running cost more	Initial cost more Running cost low
9. Applications	scooters, Bikes Cars	Cars, Buses, Trucks

Differences between 2-stroke & 4-stroke engines

condition	2-Stroke	4-Stroke
1. Inlets & outlet	It contains ports	It contains valves
2. Entry & charge	charge first enters in to crank case then enters in to cylinder	charge directly enters in to cylinder
3. Cycle completes in	2 strokes of piston or 1 revolution of crankshaft	4-stroke of piston or 2-revolutions of crankshaft
4. Torque	uniform	not uniform
5. Flywheel size	smaller	larger
6. Friction losses	less	more
7. Mechanical efficiency	more	less
8. Thermal efficiency	less	more
9. Volumetric efficiency	less	more
10. Applications	2 strokes	Bikes, Cars, Trucks etc

Valve timing Diagram for 4S-SI engine



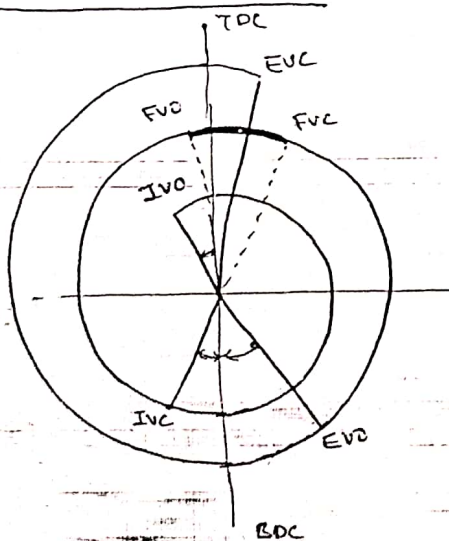
Theoretical V.T.D

IVO at TDC
 IVC at BDC
 EVO at BDC
 EVC at TDC
 Ign at TDC

Actual VTD

IVO 10 to 25° before TDC
 IVC 25 to 45° after BDC
 EVO 30 to 55° before BDC
 EVC 10 to 20° after TDC
 Ign 20 to 35° before TDC

Valve timing Diagram for 4S-CI engine



IVO 25° before TDC
 IVC 30° after BDC
 EVO 45° before BDC
 EVC 15° after TDC
 FVO 5° before TDC
 FVC 25° after TDC

Pont Timing Diagram for SI engine



TPC 35° after BDC

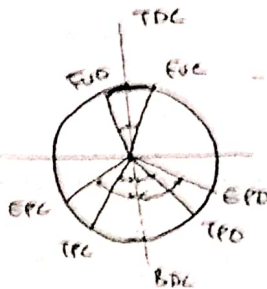
EPC 43° after BDC

EPO 43° before BDC

TPO 35° before BDC

Ion 28° before TDC

PONT timing Diagram for 2S-CI engine



TPC 45° after BDC

EPC 60° after BDC

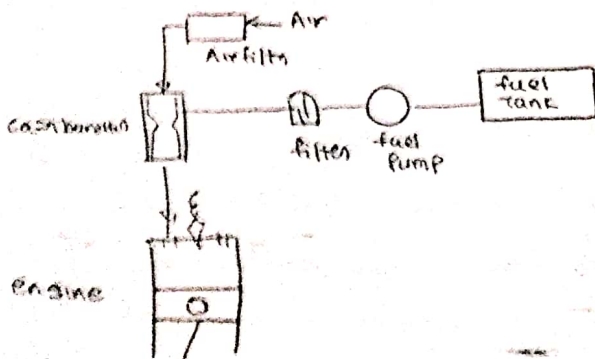
EPO 60° before BDC

TPO 45° before BDC

FVO 15° before TDC

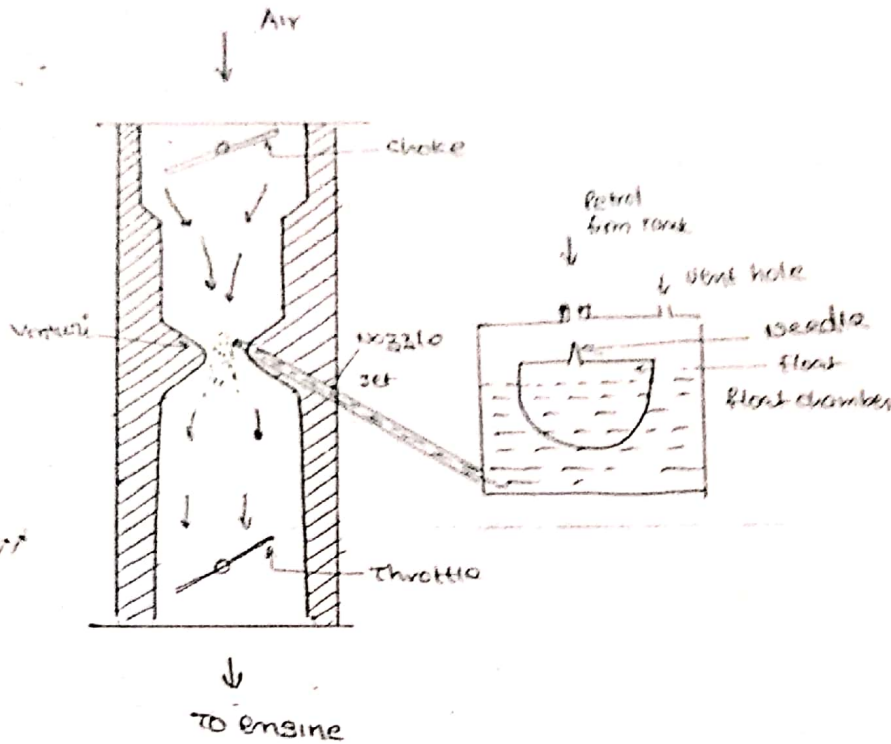
FVC 20° after TDC

Fuel Supply System for S.I engine



Carburettor: To mix Air & Petrol in the Proper ratio

Simple carburettor



- It consists of a float chamber to store the petrol, a Venturi tube, throttle valve, choke, a nozzle jet.
- Float & Needle valve maintain a constant fuel level in the float chamber.
- During the suction stroke, a partial vacuum is created in the engine cylinder and carburettor. Due to this air flows in more carburettor.
- At the narrow section of Venturi tube velocity of air increases and pressure decreases. This creates the pressure difference between float chamber and Venturi, which causes fuel to flow out of fuel jet in the form of spray.
- The fuel delivered by jet in the air stream is finely atomized, mixed with air and vapour to form homogeneous air-fuel mixture which is admitted to the engine cylinder.
- Throttle valve is connected to accelerator. When the accelerator moves throttle valve opens more and supplies more quantity of air-fuel mixture to the engine.
- Choke is used in during the starting of engine to supply rich mixture.

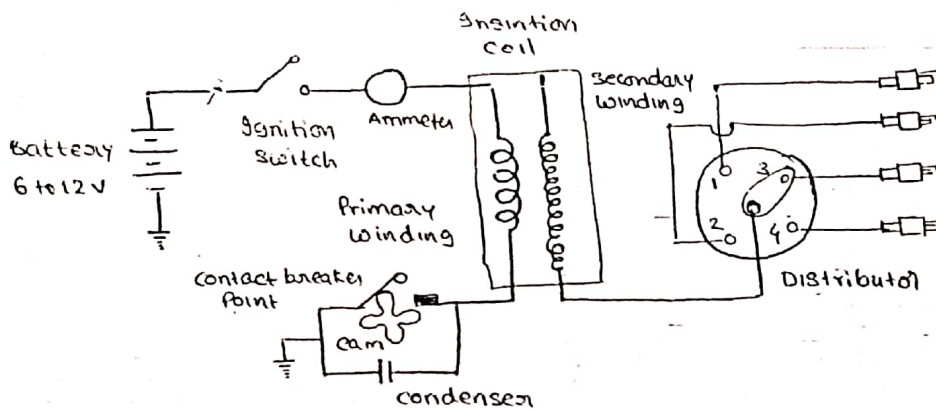
Ignition System

The Ignition System is Used to Initiate the combustion in Engine.

Types of Ignition Systems

1. Battery Ignition (coil Ignition) system
2. Magneto Ignition system.

Battery Ignition System



Components of Battery Ignition System

- It consists of a 6 to 12 Volts Battery, Ignition switch, Ignition coil, contact breaker points, condenser, distributor and spark plug
- Ignition coil consists of Primary winding & Secondary winding
- One end of primary winding is connected to the Battery & the other end connected to the contact breaker point.
- Secondary winding is connected to the Distributor.

Working:

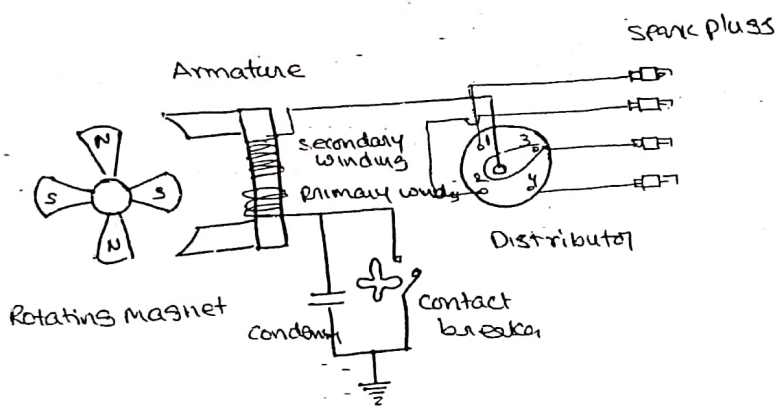
- When the Ignition switch and the contact breaker points are closed, a low voltage current from battery flows through the primary winding and a magnetic field is created around the winding.
- When the breaker points are opened by Cam, magnetic field collapses and current jumps from primary winding to secondary winding. A high voltage current (up to 20,000 volts) develops in secondary winding.

This high voltage current distributed to the different spark plugs according to firing order using Distributor.

The spark occurs due to discharge of high voltage current across the air gap of the spark plug.

Condenser is used to reduce the arcing at the contact breaker points.

Magneto Ignition System



- It consists of a magneto, instead of a battery, which produces and supplies current in the primary winding. The remaining arrangement in this system is same as that in the battery ignition system.
- The magneto consists of a fixed armature and having primary and secondary windings and a rotating magnetic assembly which is driven by the engine.
- When the magnets rotate, current flows in the primary winding, and when the contact breaker point opens current jumps from primary winding to secondary winding, a high voltage current produces in secondary winding.
- This high voltage current distributed to the different spark plugs according to firing order by using Distributor.

Spark plug

Differences between Battery & Magneto Ignition Systems

Criteria	Battery	magneto
1. Source of Energy	6 to 12 volts Battery	magneto
2. Maintenance	costly due to discharge of battery	cheap
3. Quality of Spark	Good, even at low speed	Poor at low speed Good at High speed
4. Efficiency	Decreases as speed increases	Increases as speed increases
5. Space occupied	more	less
6. Application	Cars, Buses	Two wheeled racing cars

Performance of IC Engines

The performance of an IC engine can be evaluated by using the following parameters

1. Power developed

- (a) Indicated power (IP)
- (b) Brake power (BP)
- (c) Friction power (FP)

2. Engine efficiencies

- (a) Mechanical efficiency
 - (i) Indicated thermal efficiency
 - (ii) Brake thermal efficiency
- (c) Relative efficiency
- (d) Volumetric efficiency

3. Air fuel ratio (AFR)

4. Specific fuel consumption (SFC)

5. Heat balance

Indicated power (IP)

IP = The power actually produced inside the engine cylinder is called indicated power

$$IP = \frac{P_{im} L a n k}{60} \text{ kW}$$

P_{im} = Indicated mean effective pressure (i.m.e.p) KN/m^2

L = Stroke length "m"

$a = \frac{\pi}{4} d^2$; d = bore "m"

k = no. of cylinders

$n = \text{no of working strokes}$

$= N$ for 2 stroke engines

$= \frac{N}{2}$ for 4 stroke engines

$N = \text{r.p.m.}$

$$P_{\text{im}} = \frac{\text{Area of Indicator diagram}}{\text{Length of Indicator diagram}} \times \text{Spring constant}$$

Brake Power (B.P)

The power output of an engine at the crank shaft is called Brake power.

$$BP = \frac{2\pi NT}{60} \text{ kW}$$

where $N = \text{r.p.m}$

$T = \text{torque "KN-m"}$

$$T = F \times r$$

$$T = (W - S) \left(\frac{D + d_s}{2} \right)$$

$$T = W \cdot R_m$$

where $W = \text{load "KN"}$

$S = \text{spring balance reading "KN"}$

$D = \text{diameter of brake drum "m"}$

$d_s = \text{diameter of rope "m"}$

Friction power (F.P)

The power wasted due to friction is called friction power

$$FP = IP - BP \text{ kW}$$

Mechanical Efficiency

$$\eta_{\text{Mech}} = \frac{B.P}{I.P}$$

Thermal Efficiency

It is the ratio of power developed to the heat supplied.

$$\eta_{Th} = \frac{\text{Power}}{\text{Heat supplied}}$$

Power may be IP or BP

→ Indicated Thermal Efficiency

$$I.Th.\eta = \frac{IP}{m_f \times C.V}$$

→ Brake thermal efficiency

$$B.Th.\eta = \frac{BP}{m_f \times C.V}$$

m_f = mass of fuel consumed kg/sec

$C.V$ = Calorific Value kJ/kg

IP, BP - in "kW"

Relative Efficiency

It is the ratio of actual efficiency to theoretical efficiency

$$\eta_{rel} = \frac{\text{Thermal } \eta}{\text{Air standard } \eta}$$

Thermal η may be $I.Th.\eta$ or $B.Th.\eta$

Volumetric Efficiency

$$\eta_{vol} = \frac{\text{Volume of charge admitted at NTP}}{\text{Stroke volume}}$$

Air fuel Ratio

$$A.F.R = \frac{\text{mass of Air}}{\text{mass of fuel}}$$

Specific fuel consumption (S.F.C)

The consumption of fuel per unit time per unit of power developed is called S.F.C

$$S.F.C = \frac{m_f}{IP \text{ or } BP}$$

I.S.F.C (Indicated specific fuel consumption)

$$I.S.F.C = \frac{m_f}{IP}$$

$$m_f = \text{kg/hr}$$

$$IP, BP = \text{KW}$$

B.S.F.C (Brake specific fuel consumption)

$$B.S.F.C = \frac{m_f}{BP}$$

$\therefore$ S.F.C units kg/kW-hr

other formulae

$$\rightarrow \text{Piston speed} = \frac{2LN}{60} \text{ m/sec}$$

$\rightarrow$ Brake mean effective pressure: (b.m.e.p)

$$BP = \frac{P_{bm} L A n k}{60}$$

$$P_{bm} = \frac{BP \times 60}{L A n k}$$

$$\rightarrow \text{Mechanical } \eta = \frac{BP}{IP} \text{ or } \frac{B.m.e.p}{I.m.e.p} \text{ or } \frac{I.S.F.C}{B.S.F.C}$$

Air standard efficiency

for SI engines

$$\eta_{\text{Otto}} = 1 - \frac{1}{\gamma r^{\gamma-1}}$$

where γ = compression ratio

γ = cut-off ratio

for CI engines

$$\eta_{\text{Diesel}} = 1 - \frac{1}{\gamma r^{\gamma-1}} \left[\frac{r^{\gamma}-1}{\gamma(\gamma-1)} \right]$$

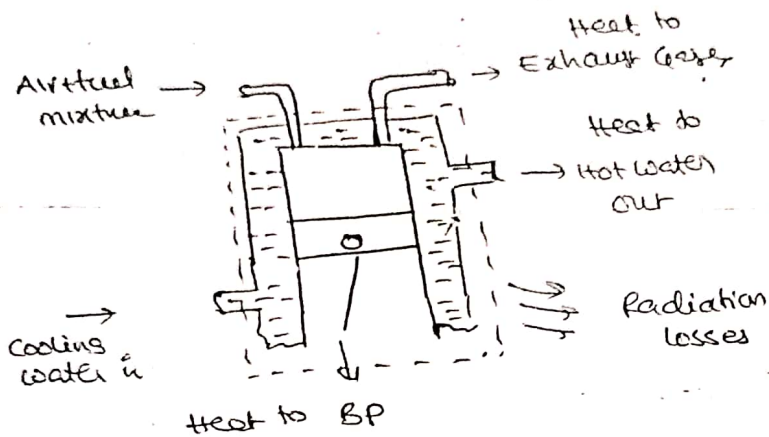
$$\gamma = [1 + K(\gamma-1)]$$

K = fuel cut-off ratio
detection before

Heat Balance

The Heat energy is generated inside the cylinder by combustion of fuel. This heat energy is not completely used to produce B.P at the crank shaft.

The purpose of Heat balance is to determine the distribution of Heat energy in the engine.



I. Heat Generated inside the cylinder

$$Q = m_f \times C.V$$

II. Heat Distribution

(i) Heat utilised to produce BP

$$R.D. = \frac{2 \pi N T}{60}$$

(ii) Heat rejected to cooling water

$$Q_{cw} = m_w \times c_w \times (T_2 - T_1)$$

T_1 = water inlet temperature

T_2 = water outlet temperature

c_w = specific heat of water KJ/kg

m_w = mass of water

(iii) Heat carried away by exhaust gases

$$Q_{eg} = m_g \times c_g \times (T_g - T_r)$$

m_g = mass of exhaust gases

$$m_g = (m_a + m_f) = m_f \left(\frac{m_a}{m_f} + 1 \right) = m_f (A.F.R + 1)$$

T_r = room temperature, T_g = exhaust temp, c_g = sp. heat of gases.

(iv) Radiation losses

$$= Q - (BP + Q_{cw} + Q_{eg})$$

Heat or Energy Balance Sheet

Heat supplied	kJ/sec	%	Heat Distribution	kJ/sec	%
1. Heat generated $Q = m \times C_p \times \Delta T$	X	100	(i) Heat to BP	X_1	$\frac{X_1}{\Sigma X} \times 100$
			(ii) Heat to cooling water	X_2	$\frac{X_2}{\Sigma X} \times 100$
			(iii) Heat to exhaust steam	X_3	$\frac{X_3}{\Sigma X} \times 100$
			(iv) Radiation loss $\approx$ (by difference)	X_4	$\frac{X_4}{\Sigma X} \times 100$
	X	100		ΣX	100

$$\eta_{th} = \frac{BP}{IP} \times 100$$

where V_f = volume of gaseous fuel per min, m^3/min

ΔH = calorific value of fuel, kJ/m^3

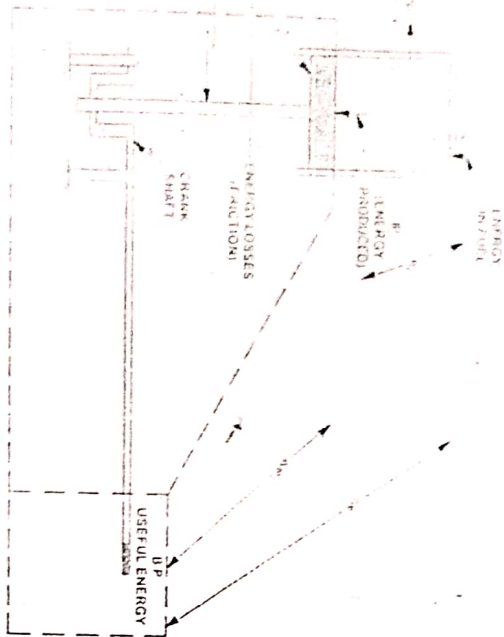


Fig. 6.5 Efficiency of Engine

(iii) **Indicated thermal efficiency** : Indicated thermal efficiency may be defined as the ratio of heat utilized to produce indicated power to heat supplied.

$$\eta_{th,i} = \frac{\text{heat equivalent to I.P.}}{\text{heat supplied}}$$

$$\eta_{th,i} = \frac{I.P. \times 60}{m_f \times \Delta H}$$

where m_f = mass of fuel supplied per minute, kg/min

ΔH = calorific value of fuel, kJ/kg

For gas engine:

$$\eta_{th,i} = \frac{I.P. \times 60}{V_f \times \Delta H}$$

where V_f = volume of gaseous fuel supplied per minute, m^3/kg

ΔH = calorific value of fuel, kJ/m^3

Since η_p and η_m are proportional to BP and IP, the mechanical efficiency can also be expressed in terms of brake and indicated thermal efficiencies.

$$\text{Mechanical efficiency} = \frac{B.Th.E}{I.Th.E}$$

(c) **Relative efficiency (efficiency ratio)** : It is defined as the ratio of the actual thermal efficiency to the ideal efficiency of the engine. The ideal efficiency of the engine is also referred as air standard efficiency of the cycle employed.

$$\text{Relative efficiency} = \frac{\text{thermal efficiency}}{\text{air standard efficiency}}$$

Relative efficiency based on BP :

$$\text{Relative efficiency} = \frac{\text{brake thermal efficiency}}{\text{air standard efficiency}}$$

Relative efficiency based on IP :

$$\text{Relative efficiency} = \frac{\text{indicated thermal efficiency}}{\text{air standard efficiency}}$$

(d) **Volumetric efficiency** : It is defined as the ratio of the actual volume of charge taken into the engine cylinder during suction stroke at NTP to the stroke volume (swept volume).

$$\text{Volumetric efficiency} = \frac{\text{volume of charge admitted at NTP}}{\text{stroke volume}}$$

Engine capacity is directly related to volumetric efficiency. The lower the volumetric efficiency, the less power an engine can develop. This is because with lower volumetric efficiency less fuel is taken in and burnt which results less power. Thus, the power developed by an engine can be increased if its volumetric efficiency is improved.

Example 6.1 : During a trial of a four stroke diesel engine the following observations were recorded :

Area of indicator diagram	= 475 mm^2
Length of indicator diagram	= 62 mm
Spring number	= 1.1 bar/mm
Diameter of the piston	= 100 mm
Length of stroke	= 150 mm
Engine RPM	= 375

Determine : (a) i.m.e.p (b) indicated power in kW

[JNTU April/May 2003, 2005]

$n = 2/2 = 1$ for 4 stroke
 $n = 2$ for 2 stroke

Solution :

$$d = 100 \text{ mm} = 0.1 \text{ m}$$

$$l = 150 \text{ mm} = 0.15 \text{ m}$$

$$N = 375 \text{ RPM}$$

(a) i.m.e.p. :

$$p_{im} = \frac{\text{area of indicator diagram}}{\text{length of indicator diagram}} \times \text{spring number}$$

$$= \frac{475}{62} \times 1.1 = 8.427 \text{ bar Ans.}$$

(b) Indicated power :

$$I.P = \frac{p_{im} \cdot a \cdot l \cdot N}{160 \times 2} = \frac{10^2 \times 8.427 \times \frac{\pi(0.1)^2}{4} \times 0.15 \times 375}{60 \times 2}$$

$$= 3.102 \text{ kW Ans.}$$

Example 6.2 : A single cylinder 4-stroke CI engine has a bore and stroke of 75 mm and 100 mm respectively. Find the BMEP if the torque is 25 Nm.

Solution :

$$d = 75 \text{ mm} = 0.075 \text{ m}$$

$$l = 100 \text{ mm} = 0.1 \text{ m}$$

$$T = 25 \text{ Nm}$$

BP in terms of BMEP,

$$BP = p_{bm} \cdot a \cdot l \cdot \frac{N}{2}$$

Also $BP = 2\pi NT$

Equating Eqn. (i) and (ii)

$$p_{bm} \cdot a \cdot l \cdot \frac{N}{2} = 2\pi NT$$

$$p_{bm} \cdot \frac{\pi(0.075)^2}{4} \times 0.1 \times \frac{N}{2} = 2\pi N \times 25$$

$$p_{bm} = \frac{8 \times 2 \times 25}{0.075^2 \times 0.1} = 7.1 \times 10^5 \text{ N/m}^2$$

$$= 7.1 \text{ bar Ans.}$$

(Chapter 6) Internal Engineering-I

Example 6.3 : The following observations were recorded during a test on a single cylinder oil engine (4 stroke engine)

Bore = 300 mm, Stroke = 450 mm, Speed = 300 RPM

IMEP = 6 bar, Net brake load = 1.5 kN

Brake drum diameter = 1.8 metres

Brake rope diameter = 20 mm

Calculate (i) Indicated power

(ii) Brake power, and

(iii) Mechanical efficiency

(AMIE)

Solution :

$$d = 300 \text{ mm} = 0.3 \text{ m}$$

$$l = 450 \text{ mm} = 0.45 \text{ m}$$

$$N = 300 \text{ RPM}; p_m = 6 \text{ bar} = 600 \text{ kN/m}^2$$

$$(W-S) = 1.5 \text{ kN}; D = 1.8 \text{ m}; d_r = 20 \text{ mm} = 0.02 \text{ m}$$

(i) Indicated power :

$$I.P = \frac{p_m \cdot a \cdot l \cdot N}{60} = \frac{600 \times \pi(0.3)^2 / 4 \times 0.45 \times 300}{60 \times 2}$$

$$= 47.71 \text{ kW Ans.}$$

(ii) Brake power :

$$BP = \frac{(W-S)R_m \cdot 2\pi N}{60}$$

When R_m = mean radius of brake drum

$$= \frac{D + d_r}{2} = \frac{1.8 + 0.02}{2} = 0.91 \text{ m}$$

$$BP = \frac{1.5 \times 0.91 \times 2 \times \pi \times 300}{60} = 42.88 \text{ kW}$$

(iii) Mechanical efficiency :

$$\eta_{mech} = \frac{BP}{IP} = \left[\frac{42.88}{47.71} \right] \times 100$$

$$= 89.87\% \text{ Ans.}$$

(Chapter 6) Internal Engineering-I

Example 6.4 : A single cylinder oil engine has a compression ratio of 10 to 1. The specific fuel consumption is 0.0 kg/kWh. Calculate the value of the fuel oil is 44000 kJ/kg. Calculate

(a) Indicated thermal efficiency

(b) Relative efficiency

Solution : Assume the engine operates on the constant volume cycle and take $\gamma = 1.4$ for air. [JNTU]

Compression ratio, $r = 10$

Fuel consumption = 0.6 kg/kWh

Calorific value of fuel oil = 44000 kJ/kg

For Otto cycle (Otto cycle)

$$\therefore \text{Indicated efficiency} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{10^{1.4-1}} = 1 - \frac{1}{10^{0.4}} = 1 - 0.398 = 0.602 = 60.2\%$$

(a) Indicated thermal efficiency

$$\eta_{th} = \frac{IP \times 60}{\left(\frac{m_f}{60}\right) \times CV} = \frac{IP \times 3600}{m_f \times CV}$$

$$= \frac{1 \times 3600}{\left(\frac{m_f}{60}\right) \times CV} = \frac{1 \times 3600}{0.6 \times 44000} \times 100 \left(\because \frac{m_f}{IP} = s.f.c \right)$$

$$= 13.63\% \text{ Ans.}$$

(b) Relative efficiency :

$$\eta_{rel} = \frac{I.Th.E.}{A.S.E.} = \left[\frac{0.1363}{0.602} \right] \times 100$$

$$= 22.64\% \text{ Ans.}$$

Example 6.5 : A 4 cylinder 4-stroke petrol engine having bore 6 cm and stroke 10 cm develops 65 Nm torque at 3000 RPM. Find the fuel consumption of the engine in kg/hr and brake mean effective pressure, if the relative efficiency of 50% and clearance volume is 60 cm³. Take CV = 40 MJ/kg

[JNTU - H, April/May - 2011]

Solution :

Bore, $d = 6$ cm

Stroke, $l = 10$ cm

Clearance volume, $v_c = 60$ cm³

$$\text{Stroke volume, } v_s = \frac{\pi d^2}{4} \times l = \frac{\pi (6)^2}{4} \times 10 = 282.74 \text{ cm}^3 = 2.827 \times 10^{-4} \text{ m}^3$$

Testing and Performance of I.C. Engines

$$T = 65 \text{ Nm} = 0.065 \text{ kNm}$$

$$N = 3000 \text{ RPM}$$

$$\text{Brake power, BP} = \frac{2\pi NT}{60} = \frac{2\pi \times 3000 \times 0.065}{60} = 20.42 \text{ kW}$$

Compression ratio,

$$r = \frac{v_s + v_c}{v_c} = \frac{282.74 + 60}{60} = 5.71$$

For petrol (Otto cycle) engine,

$$A.S.E. = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{5.71^{0.4}} = 0.502$$

Relative efficiency, $\eta_{rel} = \frac{B.Th.E.}{A.S.E.}$

$$\therefore B.Th.E. = \eta_{rel} \times A.S.E. = 0.5 \times 0.502 = 0.251 = 25.1\%$$

Fuel consumption in per hour (m_f) :

$$B.Th.E. = \frac{BP \times 3600}{m_f \times CV}$$

$$\text{or } m_f = \frac{20.42 \times 3600}{0.251 \times 40,000} = 7.32 \text{ kg/h Ans.}$$

Brake mean effective pressure (p_{bm}) :

$$BP = \frac{k \cdot p_{bm} (a/l) N / 2}{60}$$

$$\therefore p_{bm} = \frac{20.42 \times 60}{4 \times 2.827 \times 10^{-4} \times 3000 / 2} = \frac{1225.2}{1.6962} = 722.3 \text{ kN/m}^2$$

$$= 7.223 \text{ bar Ans.}$$

Example 6.6 : A 6 cylinder, 4-stroke petrol engine consumes 0.4 kg/min fuel when running at 4000 RPM. Bore is 8 cm ; Stroke is 10 cm. Clearance volume is 65 cm³. The torque developed = 11.0 Nm. Calculate BP, BMEP, $\eta_{brake\ thermal}$ and $\eta_{thermal}$. CV = 40 MJ/kg

[JNTU - H, April/May - 2011]

Solution :

Bore, $d = 8$ cm

Stroke, $l = 10$ cm

Clearance volume, $v_c = 65$ cm³

$$\text{Stroke volume, } v_s = \left(\frac{\pi d^2}{4} \right) l = \frac{\pi (8)^2}{4} \times 10$$

$$= 502.65 \text{ cm}^3 = 5.0265 \times 10^{-4} \text{ m}^3$$

$$\frac{65 + 65}{65}$$

sub

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2116

20

2 Ans.

% 100

% 100

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kJ/kg.
2005]

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0417g

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—III—

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80

$$= 2.04$$

卷

$$\frac{2.04^{1.4} - 1}{1.4(2.04 - 1)}$$

$$\frac{BP \times 60}{m_f \times CV} = \frac{1466}{0.8} = 18.325 \text{ kW Ans.}$$

$$\frac{1466}{18.325} \times 100 = 29.98\% \text{ Ans.}$$

$$\frac{BP}{m_f \times CV} = \frac{1466}{0.8} = 18.325 \text{ kW Ans.}$$

$$\frac{BP}{m_f \times CV} = \frac{1466}{0.8} = 18.325 \text{ kW Ans.}$$

Indicated fuel consumption

$$\frac{m_f}{IP} = \frac{4}{18.325} = 0.218 \text{ kg/kWh Ans.}$$

Indicated thermal efficiency

$$\frac{IP \times 60}{m_f \times CV} = \frac{18.325 \times 3600}{4 \times 42000} \times 100 = 37.48\% \text{ Ans.}$$

Note: 1 Th.E. and 4 Th.E. obtained from the equation

$$\frac{Th.E.}{1 \text{ Th.E.}}$$

$$\frac{B.Th.E.}{\eta_{mech}} = \frac{0.2998}{0.8} \times 100 = 37.48\% \text{ Ans.}$$

Example 6.9 A 4-cylinder single acting I.C. engine 12 cm bore and 15 cm stroke has a piston speed of 480 m/min. It develops 48 kW BP and has mechanical efficiency of 75%. The mean effective pressure is 4.42 bar. The specific fuel consumption is 0.25 kg per kW of BP hour. If the fuel is at 42000 kJ/kg, determine

- Indicated thermal efficiency
- Indicated fuel consumption

Solution

$$d = 12 \text{ cm} = 0.12 \text{ m}, l = 15 \text{ cm} = 0.15 \text{ m}, BP = 48 \text{ kW}$$

$$\eta_{mech} = 75\%, m_f = 4.42 \text{ bar} = 442 \text{ kN/m}^2, bsfc = 0.25 \text{ kg/kWh}, CV = 42000 \text{ kJ/kg}$$

$$\frac{BP}{m_f \times CV} = \frac{48}{0.25} = 60 \text{ kW}$$

Rating and Performance of I.C. Engines

$$IP = \frac{k \cdot P_m \cdot d \cdot l \cdot n}{60}$$

Where k = number of cylinders = 6

n = number of working strokes per minute

$$n = \frac{0.942 \times \pi (0.12)^2 / 4 \times 0.15 \times 480}{60} = 300$$

$$n = \frac{6 \times 442 \times \pi (0.12)^2 / 4 \times 0.15}{60} = 300$$

Number of working strokes per min = 300

$$\text{Also, } IP = 480$$

$$N = \frac{480}{2 \times 0.15} = 1600 \text{ RPM}$$

since $n = \frac{N}{2}$, it is a four-stroke engine.

(b) Brake thermal efficiency:

$$\frac{B.Th.E.}{m_f \times CV} = \frac{BP \times 3600}{m_f \times CV} = \left[\frac{1 \times 3600}{0.25 \times 42000} \right] \times 100 = 34.28\% \text{ Ans.}$$

Example 6.10 A 4-cylinder, 4-stroke petrol engine develops 40 kW at 40 RPS. The m.e.p. on each piston is 8 bar and the mechanical efficiency is 80%. Calculate the diameter and stroke of each cylinder if s.c.r. to bore ratio is 1.5. Calculate BSFC if brake thermal efficiency is 24%. CV of fuel is 44 MJ/kg

Solution:

$$IP = 40 \text{ kW}, N = 40 \text{ RPS}, \eta_{mech} = 80\%, P_m = 8 \text{ bar} = 800 \text{ kN/m}^2$$

$$\frac{IP}{\eta_{mech}} = \frac{40}{0.8} = 50 \text{ kW}$$

$$\text{Also, } IP = k \cdot P_m \cdot d \cdot l \cdot n$$

$$50 = 4 \times 800 \times \frac{\pi d^2}{4} \times 1.5d \times 40 \quad (\because l = 1.5d)$$

$$d = 0.069 \text{ m} = 69 \text{ mm}$$

$$\text{and } l = 1.5 \times 69 = 103.5 \text{ m Ans.}$$

$$\frac{B.Th.E.}{m_f \times CV} = \frac{BP \times 3600}{m_f \times CV}$$

$$m_f = \frac{1 \times 3600}{0.24 \times 44000} = 0.341 \text{ kg/kWh}$$

$$BSFC = 0.341 \text{ kg/kWh Ans.}$$

Example 6.10 : In a motor on 4-cylinder, 4 stroke, petrol engine of 101.6 mm bore and 127 mm stroke, the net dynamometer load is 183 N at a radius of 508 mm when the engine is 2500 RPM. At this speed and throttling opening the engine required 4.6 kW to motor it with injection switched off. At this speed and power, engine consumes 0.2 kg of petrol per minute, the calorific value of petrol is 45310 kJ/kg.

Calculate (i) the mechanical efficiency and indicated mean effective pressure
(ii) indicated thermal efficiency

Solution :

$$d = 101.6 \text{ mm} = 0.1016 \text{ m}, l = 127 \text{ mm} = 0.127 \text{ m}$$

$$(W - S) = 183 \text{ N} = 0.183 \text{ kN}, R_m = 508 \text{ mm} = 0.508 \text{ m}$$

$$N = 2500 \text{ RPM}; FP = 4.6 \text{ kW}$$

$$BP = \frac{(W - S) R_m 2\pi N}{60} = \frac{0.183 \times 0.508 \times 2\pi \times 2500}{60}$$

$$= 24.34 \text{ kW}$$

$$IP = BP + FP = 24.34 + 4.6 = 28.94 \text{ kW}$$

(i) Mechanical efficiency and IMEP :

$$\eta_{mech} = \frac{BP}{IP} \times 100$$

$$= \frac{24.34}{28.94} \times 100$$

$$= 84.1\% \text{ Ans.}$$

$$IP = \frac{k \cdot P_{im} \cdot a \cdot l \cdot N/2}{60}$$

$$P_{im} = \frac{28.94 \times 60 \times 4 \times 2}{4 \times \pi (0.1016)^2 \times 0.127 \times 2500} = 337.3 \text{ kN/m}^2$$

$$= 3.373 \text{ bar Ans.}$$

(ii) Indicated thermal efficiency :

$$I.Th.E = \frac{IP \times 60}{\eta_f \times CV} = \left[\frac{28.94 \times 60}{0.2 \times 45310} \right] \times 100$$

$$= 19.16\% \text{ Ans.}$$

✓ **Example 6.11 :** A four stroke petrol engine has a cylinder 250 mm in diameter and the stroke of the piston is 450 mm. It works on the following conditions.

Revolutions per minute = 180

Number of explosion per minute = 82

Mean effective pressure = 6.5 bar

Mechanical efficiency = 80%

Calculate (i) Number of misfires (ii) IP (iii) BP (iv) FP

Solution :

$$N = 180 \text{ RPM}$$

$$n = \frac{N}{2} = 90 \text{ (without misfires)}$$

$$P_{im} = 6.5 \text{ bar} = 650 \text{ kPa}$$

$$d = 250 \text{ mm} = 0.25 \text{ m}$$

$$l = 450 \text{ mm} = 0.45 \text{ m}$$

(i) Number of misfires / min.

$$\text{Number of misfires/min} = \frac{N}{2} - n = \frac{180}{2} - 82$$

$$= 90 - 82 = 8 \text{ Ans.}$$

(ii) Indicated power, $IP = \frac{P_{im} \cdot a \cdot l \cdot n}{60}$

Where n = actual number of explosions / min = 82

$$IP = \frac{650 \times \pi (0.25)^2 / 4 \times 0.45 \times 82}{60}$$

$$= 19.62 \text{ kW Ans.}$$

($\because n \neq N/2$ due to misfires)

(iii) Brake power, $BP = \eta_{mech} \times IP = 0.8 \times 19.62$

$$= 15.696 \text{ kW Ans.}$$

(iv) Frictional power, $FP = IP - BP = 19.62 - 15.696$

$$= 3.924 \text{ kW Ans.}$$

Example 6.12 : A four cylinder two stroke cycle petrol engine, 75 mm bore by 90 mm stroke operates on the constant volume cycle and has a compression ratio of 6 : 1. The efficiency ratio being 55%, calculate the indicated thermal efficiency. When running at 40 RPS engine develops b.m.e.p of 360 kPa and uses 9.2 kg of fuel per hour of calorific value 44000 kJ/kg. Calculate the brake thermal efficiency, mechanical efficiency and the indicated specific fuel consumption in kg/kWh.

[JNTU - April/May - 2006]

Solution :

$$d = 75 \text{ mm} = 0.075 \text{ m}, l = 90 \text{ mm} = 0.09 \text{ m}$$

$$r = 6, \eta_{rel} = 55\%, N = 40 \text{ RPS}$$

$$\text{b.m.e.p} = 360 \text{ kPa}; m_f = 9.2 \text{ kg/h}, CV = 44000 \text{ kJ/kg}$$

For constant volume cycle (Otto cycle),

$$ASE = 1 - \frac{1}{r^{\gamma}} = \left[1 - \frac{1}{6^{1.4}} \right] \times 100 = 51.15\%$$

$$\eta_{rel} = \frac{I.Th.E}{ASE}$$

$$1 \text{ Th.E.} = 9 \times \text{A.S.E.} = 0.55 \times 0.5116 = 0.2814 = 28.14\% \text{ Ans.}$$

$$BP = \frac{P_{\text{ind}} \times A \times N}{4} \times \frac{\pi (0.075)^2}{4} \times 0.09 \times 40$$

$$= 22.9 \text{ kW} \quad (\text{for 2-stroke engine } n = N)$$

$$\frac{BP}{\eta_{\text{mech}} \times CV} = \left[\frac{22.9 \times 3600}{9.2 \times 44000} \right] \times 100 = 20.36\% \text{ Ans.}$$

$$\frac{BP}{\eta_{\text{mech}} \times CV} = \frac{0.2036}{0.2814} \times 100 = 72.35\% \text{ Ans.}$$

$$\frac{IP}{\eta_{\text{mech}} \times CV} = \frac{1 \times 3600}{\left(\frac{m_1}{IP} \right) \times CV} = \frac{1 \times 3600}{\eta_{\text{mech}} \times CV}$$

$$\frac{IP}{\eta_{\text{mech}} \times CV} = \frac{1 \times 3600}{0.2814 \times 44000} = 0.2907 \text{ kg/kWh, based on IP Ans.}$$

Note: η_{mech} can also be obtained as follows:

$$\frac{BP}{IP} = \frac{22.9}{31.6} = 0.7235$$

$$\eta_{\text{mech}} = \frac{BP}{IP} = \frac{9.2}{31.65} = 0.2907 \text{ kg/kWh Ans.}$$

Example 6.13: A four cylinder engine running at 20 RPS gave 19 kW of brake power. The average torque when one cylinder was cut off was 103 Nm. Calculate the indicated thermal efficiency if CV = 43680 kJ/kg and engine uses 0.34 kg of petrol per kWh of brake power.

[JNTU-H April/May 2012]

Solution:

$$\text{B.P. of engine } B = 19 \text{ kW}$$

$$m_1 = \frac{0.34 \times 19}{3600} = 1.79 \times 10^{-3} \text{ kgs}$$

$$\text{Torque when 1 cyl. cut off}$$

$$B = \frac{2 \times \pi T}{60} = \frac{2 \times 20 \times 103}{60}$$

$$= 12.94 \text{ kW}$$

$$\text{Indicated power } = B - B_1 = 19 - 12.94 = 6.06 \text{ kW}$$

$$\text{Indicated efficiency } = \frac{B - B_1}{B} = \frac{6.06}{19} = 0.319$$

$$\frac{21.24}{11.79 \times 10^3 \times 43680} \times 100$$

$$32.16\% \text{ Ans.}$$

Example 6.13.1: In a test with a four cylinder four stroke petrol engine, the following results are obtained for a particular setting and speed:

BP with all cylinders working	= 24.0 kW
BP with No.1 cylinder cut off	= 16.2 kW
BP with No.2 cylinder cut off	= 16.7 kW
BP with No.3 cylinder cut off	= 16.8 kW
BP with No.4 cylinder cut off	= 17.3 kW

Estimate the IP of the engine and its mechanical efficiency. [JNTU-April/May 2003, 2006]

Solution:

$$B = 24.0 \text{ kW, } B_1 = 16.2 \text{ kW, } B_2 = 16.7 \text{ kW, } B_3 = 16.8 \text{ kW, } B_4 = 17.3 \text{ kW}$$

$$\text{IP of cylinder No.1} = B - B_1 = 24.0 - 16.2 = 7.8 \text{ kW}$$

$$\text{IP of cylinder No.2} = B - B_2 = 24.0 - 16.7 = 7.3 \text{ kW}$$

$$\text{IP of cylinder No.3} = B - B_3 = 24.0 - 16.8 = 7.2 \text{ kW}$$

$$\text{IP of cylinder No.4} = B - B_4 = 24.0 - 17.3 = 6.7 \text{ kW}$$

$$\text{IP of engine} = (7.8 + 7.3 + 7.2 + 6.7) = 29.0 \text{ kW Ans.}$$

$$\text{Mechanical efficiency} = \frac{BP}{IP} = \left[\frac{24.0}{29.0} \right] \times 100 = 82.75\% \text{ Ans.}$$

Example 6.14: A four cylinder four stroke petrol engine of 6.35 cm bore and 8.9 cm stroke was tested at full throttle and constant speed. The fuel supply was fixed at 0.064 kg/min and the plugs of the cylinders were short circuited without change of speed, the brake torque being correspondingly adjusted. The power measurements were as follows:

With all cylinders	= 12.3 kW
With cylinder No.1 cut out	= 8.90 kW
With cylinder No.2 cut out	= 8.80 kW
With cylinder No.3 cut out	= 8.70 kW
With cylinder No.4 cut out	= 8.85 kW

Estimate the indicated power of the engine under these conditions. Find indicated thermal efficiency of the engine if the calorific value of fuel used is 43680 kJ/kg. Compare this efficiency with the air standard value, if clearance volume of one cylinder is 70.3 cm³. [JNTU - April/May 2003]

Solution:

$$d = 6.35 \text{ cm, } l = 8.9 \text{ cm; } m_1 = 0.064 \text{ kg/min, CV} = 43680 \text{ kJ/kg}$$

$$B = 12.3 \text{ kW, } B_1 = 8.9 \text{ kW, } B_2 = 8.8 \text{ kW, } B_3 = 8.7 \text{ kW, } B_4 = 8.85 \text{ kW}$$

$$\text{IP of cylinder No.1} = B - B_1 = 12.3 - 8.9 = 3.4$$

$$\text{IP of cylinder No.2} = B - B_2 = 12.3 - 8.8 = 3.5$$

$$\text{IP of cylinder No.3} = B - B_3 = 12.3 - 8.7 = 3.6$$

$$\text{IP of cylinder No.4} = B - B_4 = 12.3 - 8.85 = 3.45$$

$$\text{IP of engine} = (3.4 + 3.5 + 3.6 + 3.45) = 13.95 \text{ kW Ans.}$$

$$\frac{13.95}{0.064 \times 43680} \times 100$$

$$29.94\% \text{ Ans.}$$

Clearance volume, $V_c = 70.3 \text{ cm}^3$

$$V_s = \left(\frac{\pi d^2}{4} \right) \cdot l = \frac{\pi (6.35)^2}{4} \times 8.9 = 281.85 \text{ cm}^3$$

$$\text{Compression ratio } r = \frac{V_c + V_s}{V_c} = \frac{70.3 + 281.85}{70.3} = 5$$

$$\text{ASE} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{5^{1.4-1}}$$

$$= [1 - 0.525] \times 100 = 47.5\%$$

$$\text{Relative efficiency} = \frac{\eta_{\text{Th.E}}}{\text{ASE}} = \left[\frac{0.2994}{0.475} \right] \times 100$$

$$= 63.03\% \text{ Ans.}$$

Example 6.15 : A four-cylinder, four stroke petrol engine of cylinder bore and stroke each equal to 77 mm, has a compression ratio of 8.5 : 1. The relative efficiency is 50% based on s.f.c based in i.p is 0.28 kg/kWh. Determine (i) CV of the petrol in MJ/kg and (ii) the petrol consumption in kg/h, given that the i.m.e.p is 950 kPa when the engine speed is 3000 RPM. Take for air $\gamma = 1.4$. [AIME]

Solution :

$$d = l = 77 \text{ mm} = 0.077 \text{ m}$$

$$N = 3000 \text{ RPM} ; P_{\text{m.e.p.}} = 950 \text{ kPa} ; r = 8.5 ; \eta_{\text{rel}} = 50\%$$

$$\text{i.e. s.f.c} = 0.28 \text{ kg/kWh} ; \gamma = 1.4 \text{ for air}$$

CV of petrol :

For petrol engine,

$$\text{ASE} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{8.5^{1.4-1}} = (1 - 0.4248) \times 100$$

$$= 57.52\%$$

$$\text{Also } \eta_{\text{rel}} = \frac{\eta_{\text{Th.E}}}{\text{ASE}}$$

$$\therefore \eta_{\text{Th.E}} = \eta_{\text{rel}} \times \text{ASE} = [0.50 \times 0.5752] \times 100$$

$$= 28.76\%$$

$$\eta_{\text{Th.E}} = \frac{\text{IP} \times 3600}{m_f \times \text{CV}} = \frac{1 \times 3600}{\left(\frac{m_f}{\text{IP}} \right) \times \text{CV}} = \frac{1 \times 3600}{0.28 \times \text{CV}}$$

$$\text{CV} = \frac{1 \times 3600}{0.2876 \times 0.28} = 44704.9 \text{ kJ/kg}$$

$$= 44.7049 \text{ MJ/kg Ans.}$$

(iii) Petrol consumption in kg/h :

$$\text{IP} = \frac{k \cdot P_{\text{m.e.p.}} \cdot a \cdot L \cdot N}{60}$$

$$= \frac{4 \times 950 \times \pi (0.077)^2 / 4 \times 0.077 \times 3000}{60 \times 2} = 34.06 \text{ kW}$$

$$\text{Petrol consumption} = \text{i.s.f.c} \times \text{IP} = 0.28 \times 34.06$$

$$= 9.537 \text{ kg/h Ans.}$$

Example 6.16 : A 2-stroke diesel engine develops a brake power of 420 kW. The engine consumes 19.5 kg/h of fuel and air-fuel ratio is 22 : 1. Calorific value of the fuel is 42000 kJ/kg. If 76 kW power is required to overcome frictional losses, calculate

(i) Mechanical efficiency

(ii) Air consumption, and

(iii) Brake thermal efficiency

Solution :

$$\text{BP} = 420 \text{ kW}$$

$$\text{FP} = 76 \text{ kW} ;$$

$$m_f = 19.5 \text{ kg/h} ; \text{CV} = 42000 \text{ kJ/kg} ; \frac{m_a}{m_f} = 22$$

$$\text{IP} = \text{BP} + \text{FP} = 420 + 76 = 496 \text{ kW}$$

(i) Mechanical efficiency :

$$\eta_{\text{mech}} = \frac{\text{BP}}{\text{IP}} = \left[\frac{420}{496} \right] \times 100$$

$$= 84.67\% \text{ Ans.}$$

(ii) Air consumption :

$$\frac{m_a}{m_f} = 22$$

$$\therefore m_a = 22 \times m_f = \frac{22 \times 19.5}{60}$$

$$= 71.5 \text{ kg/min Ans.}$$

(iii) Brake thermal efficiency

$$\text{B.Th.E} = \frac{\text{BP} \times 3600}{m_f \times \text{CV}} = \left[\frac{420 \times 3600}{19.5 \times 42000} \right] \times 100$$

$$= 18.46\% \text{ Ans.}$$

✓ **Example 6.17** Following data relates to 4-cylinder four stroke petrol engine. Air-fuel ratio by weight = 15, compression ratio of the fuel = 45200 kJ/kg, mechanical efficiency = 82%, air standard efficiency = 78%, relative efficiency = 70%, volumetric efficiency = 78%, stroke/bore ratio = 1.25, for the conditions = 1 bar 25°C, RPM = 2400, power at brakes = 72 kW

(a) Indicated thermal efficiency

(b) Brake specific fuel consumption

(c) Bore and stroke

Solution:

$$k = 1.4, \quad \eta_{me} = 16, \quad CV = 45200 \text{ kJ/kg}$$

$$\eta_{ac} = 82\%, \quad \eta_{ad} = 52\%, \quad \eta_{rel} = 70\%, \quad \eta_v = 78\%$$

$$\eta_f = 1.25, \quad N = 2400 \text{ RPM}, \quad BP = 72 \text{ kW}$$

(i) Compression ratio

$$A.S.E. = 1 - \frac{1}{r^{1.4}} = 1 - \frac{1}{r^{1.4}} = 0.52$$

$$\frac{1}{r^{0.4}} = 0.48 \quad \text{or} \quad r^{0.4} = \frac{1}{0.48}$$

$$r = \left(\frac{1}{0.48} \right)^{\frac{1}{0.4}} = 6.2 \text{ Ans.}$$

(ii) Indicated thermal efficiency:

$$\eta_{ac} = \frac{\eta_{ad}}{\eta_{rel}}$$

$$\eta_{ac} = \eta_{ad} \times \eta_{rel} = [0.7 \times 0.52] \times 100 = 36.4\% \text{ Ans.}$$

(iii) Brake specific fuel consumption:

$$\eta_{me} = \frac{\eta_{ac}}{\eta_{rel}}$$

$$\eta_{me} \times \eta_{rel} = \eta_{ac} = [0.82 \times 0.364] \times 100 = 29.85\%$$

$$\frac{BP}{\eta_{me} \times CV} = \frac{72}{0.2985 \times 45200} = 0.2985$$

$$0.6622 \text{ kg} = 19.188 \text{ kJ/h}$$

$$BP = 19.188$$

$$BP = 72$$

$$0.2665 \text{ kg/kWh Ans.}$$

[JNTU]

(iv) Bore and Stroke:

$$\dot{m}_f = 0.00533 \times 60 = 0.3198 \text{ kg/min}$$

$$\text{Mass of charge} = (m_u + m_f) = m_f \left(\frac{m_u}{m_f} + 1 \right) = m_f (AFR + 1)$$

$$= 0.3198 (16 + 1) = 5.4366 \text{ kg/min}$$

$$\text{Also, } p = 1 \text{ bar} = 1 \times 10^5 \text{ N/m}^2, \quad T = 25 + 273 = 298 \text{ K, assume } R = 0.287 \text{ kJ/kg.K}$$

∴ Volume of charge (air-fuel mixture)

$$V = \frac{mRT}{p} = \frac{5.4366 \times 0.287 \times 298}{1 \times 10^5} = 4.6497 \text{ m}^3/\text{min}$$

$$\text{Volumetric efficiency} = \frac{\text{volume of charge supplied}}{\text{swept volume}}$$

$$\therefore \text{Swept volume} = \frac{4.6497}{0.78} = 5.961 \text{ m}^3/\text{min}$$

$$\text{Also, swept volume} = K \cdot \frac{\pi d^2}{4} \cdot l \cdot \frac{N}{2}$$

$$5.961 = 4 \times \frac{\pi d^2}{4} \times 1.25 d \cdot \frac{N}{2}$$

$$5.961 = 4 \times \frac{\pi d^3}{4} \times 1.25 \times \frac{2400}{2}$$

$$5.961 = 4712.389 d^3$$

$$\text{or } d = \sqrt[3]{\frac{5.961}{4712.389}} = 0.108 \text{ m}$$

$$l = 1.25 \times 0.108 = 0.135 \text{ m}$$

$$\text{Bore} = 108 \text{ mm}$$

$$\text{Stroke} = 135 \text{ mm Ans.}$$

Example 6.18: A 4-stroke gasoline engine develops a brake power of 41.10 kW. The engine consumes 120 kg of fuel in one hour and the air consumption is 40 kg/min. If the mechanical efficiency is 87% and the fuel heating value is 43000 kJ/kg, determine the following

(i) Fuel ratio

(ii) Indicated and brake thermal efficiencies

Solution:

$$BP = 41.10 \text{ kW}, \quad \dot{m}_f = 120 \text{ kg/h}, \quad \dot{m}_a = 40 \text{ kg/min}$$

$$\eta_{me} = 87\%, \quad CV = 43000 \text{ kJ/kg}$$

[JNTU]

(ii) Air-fuel ratio :

$$m_f = 120 \text{ kg/h} = \frac{120}{60} = 2 \text{ kg/min}$$

$$\therefore \text{Air-fuel ratio} = \frac{m_a}{m_f} = \frac{40}{2} = 20$$

i.e. 20 : 1 Ans.

(iii) Indicated and brake thermal efficiencies :

$$\eta_{\text{mech}} = \frac{\text{BP}}{\text{IP}}$$

$$\therefore \frac{\text{BP}}{\text{IP}} = \frac{410}{0.87} = 471.26 \text{ kW}$$

$$\text{I.Th.E} = \frac{\text{IP} \times 3600}{m_f \times \text{CV}} = \left[\frac{471.26 \times 3600}{120 \times 43000} \right] \times 100$$

$$= 32.88\% \text{ Ans.}$$

$$\eta_{\text{mech}} = \frac{\text{B.Th.E}}{\text{I.Th.E}}$$

$$\therefore \text{B.Th.E} = \eta_{\text{mech}} \times \text{I.Th.E}$$

$$= [0.87 \times 0.3288] \times 100$$

$$= 28.6\% \text{ Ans.}$$

Note : Brake thermal efficiency can also be calculated from the equation,

$$\text{B.Th.E} = \frac{\text{BP} \times 3600}{m_f \times \text{CV}} = \left[\frac{410 \times 3600}{120 \times 43000} \right] \times 100$$

$$= 28.6\% \text{ Ans.}$$

Example 6.19 : A 4-stroke, four cylinder petrol engine, 90 mm bore $\times$ 120 mm stroke develops brake power of 27 kW while running at 1500 r.p.m when using a 20% rich mixture, if the volume of air drawn into the cylinders when measured at 15°C and 760 mm of Hg is 70% of the swept volume, the theoretical air-fuel ratio is 15 : 1. The calorific value of fuel is 43960 kJ/kg. The mechanical efficiency of the engine is 90%. Determine the indicated thermal efficiency and bmep.

Solution :

$$d = 90 \text{ mm}, l = 120 \text{ mm}, N = 1500 \text{ r.p.m} ; \text{BP} = 27 \text{ kW}$$

$$\eta_{\text{mech}} = 90\%, \text{A : F ratio} = 15, \text{CV} = 43960 \text{ kJ/kg}$$

$$\frac{\text{BP}}{\eta_{\text{mech}}} = \frac{27}{0.9} = 30 \text{ kW}$$

Volume of air drawn at 15°C and 760 mm Hg

$$= 0.7 \times \text{swept volume}$$

$$= 0.7 \times \frac{\pi d^2}{4} \times l \times k$$

$$= 0.7 \times \frac{\pi (0.09)^2}{4} \times 0.12 \times 4$$

$$= 0.00214 \text{ m}^3/\text{stroke}$$

$$\text{Also, } p = 760 \text{ mm Hg} = 1.013 \times 10^5 \text{ N/m}^2, T = (15 + 273) = 288 \text{ K}$$

Assume $R = 0.287 \text{ kJ/kg K}$

$$\therefore m_a = \frac{pv}{RT} = \frac{1.013 \times 10^5 \times 0.00214}{0.287 \times 288}$$

$$= 0.00262 \text{ kg/stroke}$$

Theoretical A : F ratio = 15

$$\text{i.e., } \frac{m_a}{m_f} = 15 \quad \text{or} \quad m_f = \frac{m_a}{15} = \frac{0.00262}{15}$$

$$= 0.000174 \text{ kg/stroke}$$

Theoretical mass of fuel/min

$$= \text{mass of fuel / stroke} \times \text{number of strokes / min}$$

$$= 0.000174 \times \frac{1500}{2} = 0.131 \text{ kg}$$

Actual mass of fuel

$$m_f = 0.131 \times 1.2 = 0.1572 \text{ kg/min}$$

i.e. 20% rich mixture

$$\text{I.Th.E} = \frac{\text{IP} \times 60}{m_f \times \text{CV}} = \left[\frac{30 \times 60}{0.1572 \times 43960} \right] \times 100$$

= 26.04% Ans.

$$\text{BP} = \frac{k \cdot p_{\text{mep}} \cdot a \cdot l \cdot N}{60 \times 2}$$

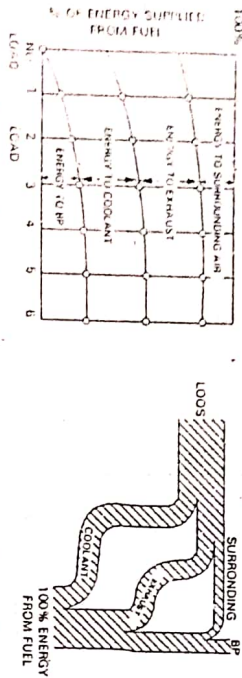
$$p_{\text{mep}} = \frac{\text{BP} \times 60 \times 2}{k \cdot a \cdot l \cdot N} = \frac{27 \times 60 \times 2}{4 \times \frac{\pi (0.09)^2}{4} \times 0.12 \times 1500}$$

= 707.35 kN/m² Ans.

Table 6.1
ENERGY (HEAT) BALANCE SHEET

Energy supplied	kJ/min	%	Energy Distribution	kJ/min	%
(i) Energy supplied in Fuel	X	100	(i) Energy to BP	X_1	$\frac{X_1}{X} \times 100$
			(ii) Energy to cooling	X_2	$\frac{X_2}{X} \times 100$
			(iii) Energy to exhaust gas	X_3	$\frac{X_3}{X} \times 100$
			(iv) Energy to surroundings (by difference)	X_4	$\frac{X_4}{X} \times 100$
	X	100	ΣX	ΣX	100

The energy balance can also be represented graphically or by a sankey diagram as shown in Fig. 6.11.



(a) Graphical representation

(b) Sankey diagram

Fig. 6.11 Heat Balance Diagrams

Performance curves :

Performance curves are graphical representation of engine characteristics. They are obtained from the test results. The typical curves for variable speed test are shown below.

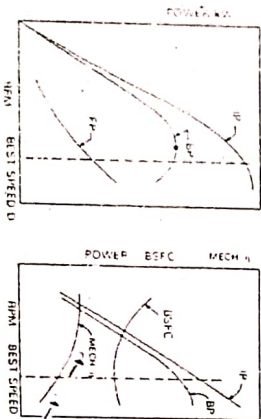


Fig. 6.12 Performance Curves

Example 6.23 : A single cylinder engine working on 4-stroke cycle has a bore of 110 mm and stroke 130 mm and runs at 600 RPM. The mean effective pressure is 6 bar. It consumes 10 cc of fuel in 28 seconds. The diesel oil used is having a C.V of 42000 kJ/kg and specific gravity of 0.85. The brake wheel diameter is 850 mm and rope diameter is 20 mm. The net load on the brake is 0.11 kN. Calculate

- (a) I.P (b) B.P (c) Mechanical efficiency (d) Indicated thermal efficiency
(e) Brake thermal efficiency

Solution :

$$d = 0.11 \text{ m}; l = 0.13 \text{ m}; P_m = 6 \text{ bar} = 600 \text{ kN/m}^2; N = 600 \text{ rpm},$$

$$\text{Fuel consumption} = 10 \text{ cc}$$

$$\text{Specific gravity} = 0.85$$

$$\therefore \text{Mass of fuel} = \frac{10}{28} \times 0.85 = 0.303 \text{ gm/s}$$

$$= 0.000303 \text{ kgs}$$

$$\text{Net brake load, } (W - S) = 0.11 \text{ kN}$$

Mean radius

$$R_m = \left(\frac{0.85 + 0.02}{2} \right) = 0.435 \text{ m}$$

$$(a) \text{ I.P.} = \frac{P_m \cdot d \cdot l \cdot N}{60} = \frac{600 \times \left(\frac{\pi \times 0.11^2}{4} \right) \times 0.13 \times 600 / 2}{60} = 3.706 \text{ kW Ans.}$$

$$(b) \text{ B.P.} = \frac{(W - S) R_m \cdot 2\pi N}{60} = \frac{0.11 \times 0.435 \times 2\pi \times 600}{60} = 3.006 \text{ kW Ans.}$$

$$(c) \text{ Mechanical efficiency} = \frac{\text{B.P.}}{\text{I.P.}} = \left[\frac{3.006}{3.706} \right] \times 100 = 81.12\% \text{ Ans.}$$

$$(d) \text{ Indicated thermal } \eta = \frac{\text{I.P.}}{m_f \times \text{C.V.}} = \left[\frac{3.706}{0.000303 \times 42000} \right] \times 100 = 29.12\% \text{ Ans.}$$

$$(e) \text{ Brake thermal } \eta = \frac{\text{B.P.}}{m_f \times \text{C.V.}} = \left[\frac{3.006}{0.000303 \times 42000} \right] \times 100 = 23.62\% \text{ Ans.}$$

Example 6.24 : A six cylinder, four stroke CI engine is tested against a water brake dynamometer

for which $BP = \frac{WN}{17 \times 10^3}$ in kW, where W is brake load in newtons and N is the speed of the engine in rpm. The air consumption was measured by means of a sharp edged orifice. During the test following observations were taken :

Bore	= 10 cm
Stroke	= 14 cm
Speed	= 2500 rpm
Brake load	= 480 N
Barometer reading	= 76 cm of Hg
Orifice diameter	= 3.3 cm
Coefficient of discharge	= 0.62
Pressure drop across orifice	= 14 cm of Hg
Room temperature	= 25°C
Fuel consumption	= 0.32 kg/min

Calculate the following

- The volumetric efficiency
- Brake mean effective pressure
- The engine torque and
- The brake specific fuel consumption

Solution :

$$P_a = 76 \text{ cm of Hg} = 1.013 \text{ bar}$$

$$T_a = 25 + 273 = 298 \text{ K}$$

(i) Volumetric efficiency :

$$\begin{aligned} \text{Density of air : } \rho_a &= \frac{P_a}{R_a T_a} = \frac{1.013 \times 10^2}{0.287 \times 298} \\ &= 1.1844 \text{ kg/m}^3 \end{aligned}$$

Head causing flow of air,

$$h_a = \frac{h \times \rho}{\rho_a} = \frac{14 \times 13.6 \times 1000}{1.1844} = 1607.56 \text{ m of air}$$

$$\begin{aligned} \text{Volume of air flow} &= C_d \cdot A \cdot \sqrt{2g h_a} \\ &= 0.62 \times \frac{\pi (0.033)^2}{4} \times \sqrt{2 \times 9.81 \times 1607.56} \\ &= 0.094 \text{ m}^3/\text{s} \end{aligned}$$

$$(\therefore \rho_a = 0.287 \text{ kJ/kg K})$$

$$\therefore \text{Air consumption/stroke} = \frac{0.094}{2500} = \frac{0.094 \times 120}{2500} = 0.0045 \text{ m}^3$$

$$\text{Swept volume} = \frac{\pi d^2}{4} l \times k = \frac{\pi (0.1)^2 \times 0.14 \times 6}{4} = 0.00659 \text{ m}^3$$

$$\text{Volumetric efficiency} = \frac{\text{volume of air admitted/stroke}}{\text{swept volume}}$$

$$= \left[\frac{0.00451}{0.00659} \right] \times 100 = 68.4\% \text{ Ans}$$

(ii) Brake mean effective pressure :

$$BP = \frac{WN}{17 \times 10^3} = \frac{480 \times 2500}{17 \times 10^3} = 70.588 \text{ kN}$$

$$\text{Also, } BP = \frac{k p_{bm} \cdot a \cdot l \cdot N}{60 \times 2}$$

$$\therefore p_{bm} = \frac{BP \times 60 \times 2}{k \cdot a \cdot l \cdot N} = \frac{70.588 \times 60 \times 2}{\pi (0.1)^2 \times 0.14 \times 2500} = 513.57 \text{ kN/m}^2 \text{ Ans}$$

(iii) Engine torque :

$$T = \frac{BP}{2\pi N} = \frac{70.588 \times 10^3}{2\pi \times 2500/60} = 269.626 \text{ Nm Ans}$$

(iv) Brake specific fuel consumption

$$bsfc = \frac{m_f}{BP} = \frac{0.32 \times 60}{70.588} = 0.272 \text{ kg/kWh Ans}$$

Example 6.25 : A two-stroke cycle, 210 mm bore $\times$ 280 mm stroke single cylinder oil engine gives the following results ω_{test}

- Speed = 350 RPM ; Net brake load = 620 N
- Diameter of brake drum = 1 m, oil consumption = 4.25 kg/hr
- i.m.e.p = 2.75 bar, Heating value of fuel used = 45000 kJ/kg
- Air-fuel ratio by mass = 32
- Room temperature = 20°C
- Exhaust gas temperature = 370°C

Calculate (i) IP, (ii) BP, (iii) Indicated thermal efficiency (net brake thermal efficiency) and (iv) percentage heat loss to exhaust gases.

Assume $C_p = 1.007 \text{ kJ/kgK}$ for the exhaust gases

Solution:

$$d = 210 \text{ mm} = 0.21 \text{ m}, l = 280 \text{ mm} = 0.28 \text{ m}$$

$$P_m = 2.75 \text{ bar} = 275 \text{ kN/m}^2, N = 350 \text{ RPM}$$

$$(W - S) = 620 \text{ N} = 0.62 \text{ kN}, R_m = \frac{D}{2} = \frac{1}{2} = 0.5 \text{ m}$$

$$\text{Piston area, } A = \frac{\pi (0.21)^2}{4} = 0.03462 \text{ m}^2$$

$$(i) \quad I.P. = \frac{P_m \cdot A \cdot N}{60} = \frac{275 \times 0.03462 \times 350}{60} = 15.55 \text{ kW Ans.}$$

$$(ii) \quad B.P. = \frac{(W - S) R_m \cdot 2\pi N}{60} = \frac{0.62 \times 0.5 \times 2\pi \times 350}{60} = 11.356 \text{ kW Ans.}$$

$$(iii) \quad \text{B.Th.E} = \frac{\text{BP} \times 3600}{m_f \times \text{CV}} = \frac{11.356 \times 3600}{4.25 \times 45000} \times 100 = 21.37\% \text{ Ans.}$$

(iv) Percentage of heat lost in exhaust gases:

$$\text{Heat to exhaust} = m_g \times C_p (t_g - t_i)$$

Where m_g = (mass of air + mass of fuel)

$$= (m_g + m_f) = m_f \left(\frac{m_s}{m_f} + 1 \right) = m_f (\text{AFR} + 1)$$

$$= 4.25 \times (32 + 1)$$

$$= 140.25 \text{ kg/hr} = 0.03896 \text{ kg/s}$$

$$\text{Heat to exhaust} = 0.03896 \times 1007 (370 - 20) = 13.73 \text{ kJ/s}$$

$$\text{Heat supplied in fuel} = m_f \times \text{CV}$$

$$= \frac{4.25}{3600} \times 45000 = 53.125 \text{ kJ/s}$$

Percentage heat lost to exhaust gases,

$$= \left(\frac{13.73}{53.125} \right) \times 100$$

$$= 25.84\% \text{ Ans.}$$

Rating and Performance of I.C. Engines

Example 6.26: A six cylinder four stroke cycle marine oil engine has cylinder diameters of 610 mm and a piston stroke of 1250 mm. When the engine speed is 120 rpm it uses 340 kg of fuel oil of calorific value 44.2 MJ/kg in one hour. The cooling water amounts to 19200 kg/h, entering at 15°C and leaving at 63°C. The torque transmitted at the engine couplings is 108 k N-m and the indicated mean effective pressure is 775 kPa. Determine

- the i.p.,
- the b.p.,
- the mechanical efficiency,
- b.m.e.p
- brake thermal efficiency
- the percentage of energy supplied per kg of fuel lost to the cooling water
- the fuel used per kWh on a brake power basis.

Solution:

$$\text{Diameter, } d = 610 \text{ mm} = 0.61 \text{ m}$$

$$\text{Stroke, } l = 1250 \text{ mm} = 1.25 \text{ m}$$

$$\text{Engine speed, } N = 120 \text{ RPM}; p_m = 775 \text{ kPa}$$

$$\text{Torque, } T = 108 \text{ k N-m}$$

(i) The indicated power:

$$I.P. = \frac{k \cdot p_m \cdot a \cdot l \cdot N}{2} = \frac{6 \times 775 \times \pi (0.61)^2 / 4 \times 1.25 \times 120 / 2}{60} = 1698.68 \text{ kW Ans.}$$

(ii) The brake power:

$$B.P. = \frac{2\pi NT}{60} = \frac{2\pi \times 120 \times 108}{60} = 1357.17 \text{ kW}$$

(iii) The mechanical efficiency:

$$\eta_{\text{mech}} = \left(\frac{\text{BP}}{\text{IP}} \right) \times 100 = \left(\frac{1357.17}{1698.68} \right) \times 100 = 79.89\% \text{ Ans.}$$

(iv) Brake mean effective pressure (P_{bm}):

$$B.P. = \frac{k \cdot P_{bm} (a) \cdot l \cdot N}{2} = \frac{P_{bm} \cdot 6 \times \pi (0.61)^2 / 4 \times 1.25 \times 120 / 2}{60}$$

$$1357.17 = 2.192 P_{bm}$$

$$P_{bm} = \frac{1357.17}{2.192} = 619.2 \text{ kN/m}^2 \text{ Ans.}$$

Note: Mech. efficiency can also be obtained as:

$$\eta_{\text{mech}} = \frac{P_{\text{brake}}}{P_{\text{ind}}} = \left(\frac{619.2}{775} \right) \times 100 = 79.89\% \text{ Ans.}$$

(e) Brake thermal efficiency:

$$\text{B.Th.E} = \frac{\text{BP} \times 3600}{m_f \times \text{CV}} = \left[\frac{1357.17 \times 3600}{340 \times 44.2 \times 10^3} \right] \times 100 = 32.51\% \text{ Ans.}$$

(f) The percentage of energy supplied per kg of fuel lost to the cooling water:

Mass of cooling water per kg of fuel,

$$m_c = \frac{19200}{340} = 56.47 \text{ kg/kg of fuel}$$

Energy lost to cooling water

$$= m_c \times c_w (t_2 - t_1) = 56.47 \times 4.18 (63 - 15) = 11330.18 \text{ kJ}$$

Energy supplied per kg of fuel

$$= 44.2 \times 10^3 \text{ kJ}$$

\therefore % energy lost to cooling water

$$= \left(\frac{11330.18}{44.2 \times 10^3} \right) \times 100 = 25.63\% \text{ Ans.}$$

(g) The fuel used per kWh on a brake power basis (bs/c):

$$\text{bs/c} = \frac{340}{1357.17} = 0.25 \text{ kg/kWh Ans.}$$

Example 6.27: In a full load test on a two cylinder, four stroke oil engine, the following readings were taken: mean effective pressure = 775 kPa, net brake load 2000 N, effective brake diameter = 1.5 m, mean speed = 300 rpm, fuel consumption 14.4 kg per hour, quantity of cylinder cooling water = 12 kg per min; cylinder cooling water inlet temperature = 16°C, cylinder cooling water outlet temperature = 70°C; quantity of exhaust calorimeter cooling water = 20 kg/min exhaust cooling water inlet temperature = 16°C; exhaust cooling water outlet temperature 55°C; temperature of gases leaving calorimeter = 86°C; air/fuel ratio by mass = 20; engine room temperature 18°C.

The calorific value of the fuel oil was 43000 kJ/kg and the mean specific heat of the exhaust gases was 1.004 kJ/kg K. The stroke volume per cylinder was 15 litres and each cylinder may be assumed to have contributed equally towards the work done. Find the mechanical and brake thermal efficiencies. Draw up a heat balance for the test expressed in kJ/min. [JNTU April/May - 2006]

Solution:

$$\text{Stroke volume, } V_s = 15 \text{ litres} = 15 \times 10^{-3} = 0.015 \text{ m}^3$$

$$\text{Mean effective pressure, } P_m = 775 \text{ kPa}$$

$$\text{Mean speed, } N = 300 \text{ RPM}$$

$$\text{Fuel consumption, } m_f = 12 \text{ kg/min}$$

$$\text{Calorific value of fuel, CV} = 43000 \text{ kJ/kg}$$

$$\text{Net brake load, (W - S)} = 2000 \text{ N} = 2 \text{ kN}$$

$$\text{Effective brake radius, } R_m = \frac{1.5}{2} = 0.75 \text{ m}$$

$$\text{IP} = \frac{k P_m (a) \times N / 2}{60} = \frac{2 \times 775 \times 0.015 \times 300 / 2}{60} = 58.125 \text{ kW}$$

$$\text{BP} = \frac{(W - S) R_m \cdot 2\pi N}{60} = \frac{2 \times 0.75 \times 2 \times \pi \times 300}{60} = 47.124 \text{ kW}$$

$$\text{Mechanical efficiency} = \left(\frac{\text{BP}}{\text{IP}} \right) \times 100 = \left(\frac{47.124}{58.125} \right) \times 100 = 81.07\% \text{ Ans.}$$

$$\text{Brake thermal efficiency} = \frac{\text{BP} \times 3600}{m_f \times \text{CV}} = \left[\frac{47.124 \times 3600}{14.4 \times 43000} \right] \times 100 = 27.4\% \text{ Ans.}$$

Heat balance sheet (minute basis)

$$\begin{aligned} \text{Heat supplied in fuel} &= m_f \times \text{CV} = \frac{14.4}{60} \times 43000 \\ &= 10,320 \text{ kJ/min} \end{aligned}$$

Heat distribution:

$$(i) \text{ Heat to BP} = 47.124 \times 60 = 2827.44 \text{ kJ/min}$$

(ii) Heat to exhaust gases:

$$\text{AFR} = 20$$

$$\text{Mass of exhaust gas, } m_g = m_f (\text{AFR} + 1)$$

$$= \frac{14.4}{60} (20 + 1) = 5.04 \text{ kg/min}$$

(a) Heat in exhaust gases

$$= m_g \times C_p (t_g - t_a)$$

$$= 5.04 \times 1 (86 - 18) = 342.72 \text{ kJ/min}$$

(b) Heat lost by exhaust gases in calorimeter

= Heat to circulating cooling water

$$= m_w \cdot C_w (t_2 - t_1) = 20 \times 4.18 (55 - 16) = 3260.4 \text{ kJ/min}$$

Total heat to exhaust gases

$$= (342.72 + 3260.4) = 3603.12 \text{ kJ/min}$$

(iii) Heat to cooling water = $m_w \cdot C_w \cdot (t_2 - t_1)$

$$= 12 \times 4.18 (70 - 16) = 2708.64 \text{ kJ/min}$$

(iv) Heat to surrounding (By difference)

$$= 10320 - (2827.44 + 3603.12 + 2708.64)$$

$$= 1180.8 \text{ kJ/min}$$

Heat balance sheet (minute basis)

Heat supplied	kJ/min	Heat distribution	kJ/min
Heat in fuel	10,320	(i) Heat to BP	2827.44
		(ii) Heat to exhaust gases	3603.12
		(iii) Heat to cooling water	2708.64
		(iv) Heat to surrounding unaccounted losses by (difference)	1180.8
	10,320		10320.00

Example 6.28 : A two stroke diesel engine was motored when meter reading was 1.5 kW. Then the test on the engine was carried with following results :

Brake torque = 120 N m, RPM = 600 ; fuel used = 2.5 kg, CV of fuel = 41 kJ/kg; cooling water used = 820 kg. Rise in cooling water temperature is 10°C. Exhaust gas temperature = 350°C ; Room temperature = 25°C ; A : F = 32 : 1 ; Calculate BP, IP, Mechanical and indicated thermal efficiencies and heat balance on percentage basis.

(JNTU - H, April/May - 2011)

Solution :

$$\text{Brake torque, } T = 120 \text{ NM} = 0.12 \text{ kNm}$$

$$\text{Speed, } N = 600 \text{ RPM}$$

$$BP = \frac{2\pi NT}{60} = \frac{2\pi \times 600 \times 0.12}{60}$$

$$= 7.54 \text{ kW Ans.}$$

$$IP = 7.54 + 1.5 = 9.04 \text{ kW Ans.}$$

$$\text{Mechanical efficiency} = \left(\frac{BP}{IP} \right) = \left(\frac{7.54}{9.04} \right) \times 100$$

$$= 83.4\% \text{ Ans.}$$

Testing and Performance of I.C. Engines

$$I Th. E. = \frac{IP \times 3600}{2.5 \times 41000}$$

(Assume $m_f = 2.5 \text{ kg/h}$)

$$= \left[\frac{9.04 \times 3600}{2.5 \times 41000} \right] \times 100 = 31.75\% \text{ Ans.}$$

Heat balance sheet (percentage basis) :

$$\text{Heat in fuel} = \frac{m_f}{3600} \times CV$$

$$= \frac{2.5 \times 41,000}{3600} = 28.47 \text{ kJ/s} = 100\%$$

Heat distribution :

$$(i) \text{ Heat to BP} = 7.54 \text{ kJ/s} = \frac{7.54}{28.47} \times 100 = 26.48\%$$

(ii) Heat to exhaust gases

Mass of exhaust gases,

$$m_g = m_f (AFR + 1) = \frac{2.5(32 + 1)}{3600} = 0.0229 \text{ kg/s}$$

Heat to exhaust gases

$$= m_g \cdot C_p (t_2 - t_1) = 0.0229 \times (350 - 25)$$

$$= 7.44 \text{ kJ/s} = 26.13\%$$

(iii) Heat to cooling water

$$= m_w \cdot C_w (t_2 - t_1) = \frac{820}{3600} \times 4.18 (10)$$

$$= 9.52 \text{ kJ/s} = 33.44\%$$

(iv) Heat to surrounding (By difference)

$$= 28.47 - (7.54 + 7.44 + 9.52)$$

$$= 3.97 \text{ kJ/s} = 13.95\%$$

Heat balance sheet (percentage basis) :

Heat supplied	kJ/s	%	Heat distribution	kJ/s	%
Heat in fuel	28.47	100	(i) Heat to BP	7.54	26.48
			(ii) Heat to exhaust gases	7.44	26.13
			(iii) Heat to cooling water	9.52	33.44
			(iv) Heat to surrounding (By difference)	3.97	13.95
	28.47	100		28.47	100

Example 6.29 : In a full load on an oil engine the following results were obtained.

$$IP = 30 \text{ kW, BP} = 24 \text{ kW, Fuel consumption} = 0.128 \text{ kg/min}$$

Cylinder circulating water	= 5.9 kg/min
Temperature rise of cooling water	= 49.5°C
Temperature of exhaust gas	= 387.8°C
Temperature of engine room	= 18.4°C
Air to fuel ratio	= 20
Calorific value of oil	= 45200 kJ/kg
Specific heat of exhaust gas	= 1.05 kJ/kg K
Specific heat of water	= 4.2 kJ/kg K

Determine the mechanical and indicated thermal efficiencies and draw up an energy balance on the basis of kJ/min and in percentage.

Solution :

$$\text{Mechanical efficiency} = \frac{BP}{IP} = \frac{24}{30} \times 100$$

$$= 80\% \text{ Ans.}$$

$$\text{Indicated thermal efficiency} = \frac{IP \times 60}{m_f \times CV} = \left[\frac{30 \times 60}{0.128 \times 45200} \right] \times 100$$

$$= 31\% \text{ Ans.}$$

$$\text{Heat energy supplied/min} = 0.128 \times 45200 = 5785.6 \text{ kJ/min}$$

$$\text{Energy to B.P.} = 24 \times 60 = 1440 \text{ kJ/min}$$

$$\text{Energy to cooling water} = m_c C_w (t_2 - t_1) = 5.9 \times 4.2 (49.5)$$

$$= 1226.6 \text{ kJ/min}$$

$$\text{Energy to exhaust gas} = m_g \times C_p (t_g - t_r)$$

$$\text{Where } m_g = m_f (AFR + 1) = (20 + 1) 0.128 = 2.688 \text{ kg/min}$$

$$\text{Energy to exhaust gas} = 2.688 \times 1.05 (387.8 - 18.4)$$

$$= 1042.56 \text{ kJ/min}$$

$$\text{Energy to surroundings (By difference)}$$

$$= 5785.6 - (1440 + 1226.6 + 1042.56)$$

$$= 2076.44 \text{ kJ/min}$$

Table 6.2

ENERGY BALANCE SHEET

Energy supplied	kJ/min.	%	Energy Distribution	kJ/min.	%
Energy in-fuel	5788.6	100	Energy to B.P.	1440	24.88
			Energy to cooling water	1226.6	21.19
			Energy to exhaust gas	1042.56	18.01
			Energy to surroundings	2076.44	35.92
	5788.6	100		5788.6	100

Example 6.30 : The following observations were made during the trial of a Diesel engine. The power developed by the engine is used for driving a D.C generator. The output of the generator was 200 amperes at 200 volts. The efficiency of generator being 85%.

The quantity of fuel supplied to the engine was 0.18 kg/min. Calorific value of fuel being 42500 kJ/kg. The air-fuel ratio was 20 : 1. The exhaust gases were passed through a exhaust gas calorimeter for which the observations were as under.

Water circulated through exhaust gas calorimeter = 9.5 kg/min

Temperature rise of water through calorimeter = 36°C

Temperature of exhaust gases at exit from calorimeter = 106°C.

Ambient temperature = 20°C

If the specific heat of exhaust gases is 1.0 kJ/kgK, Draw up the heat balance sheet on minute basis. Heat lost to jacket cooling water is 25% of total heat supplied.

Solution :

$$\text{Output of DC generator} = VI = 200 \times 200 = 40000 \text{ W}$$

$$= 40 \text{ kW}$$

$$\text{Input to the generator} = \text{BP of the engine}$$

$$= \frac{40}{0.85} = 47.06 \text{ kW}$$

$$\text{Heat supplied} = m_f \times CV = 0.18 \times 42500$$

$$= 7650 \text{ kJ/min}$$

$$\text{Mass of exhaust gases, } m_g = m_f (AFR + 1) = 0.18(20 + 1) = 3.78 \text{ kg/min}$$

Heat distribution :

$$(a) \text{ Heat to BP} = \text{BP} \times 60 = 47.06 \times 60 = 2823.6 \text{ kJ/min}$$

$$(b) \text{ Heat carried away by exhaust gases}$$

$$= m_g \cdot C_p (t_{g2} - t_r) + m_w \cdot C_w (t_{w2} - t_{w1})$$

$$= 3.78 \times 1(100 - 20) + 9.5 \times 4.2 (36)$$

$$= 302.4 + 1436.4 = 1738.8 \text{ kJ/min}$$

Heat lost to cooling water

$$= \frac{2.5}{100} \times \text{Heat supplied}$$

$$= 0.25 \times 7650 = 1912.5 \text{ kJ/min}$$

(a) Heat to surroundings (by difference)

$$= 7650 - (2823.6 + 1738.8 + 1912.5)$$

$$= 1175.1 \text{ kJ/min}$$

Table 6.3

HEAT BALANCE SHEET

Heat supplied	kJ/min.	% age	Heat Distribution	kJ/min.	% age
Heat in fuel	7650	100	(a) Heat to BP	2823.6	36.9
			(b) Heat to exhaust gases	1738.8	22.73
			(c) Heat to cooling water	1912.5	25.0
			(d) Heat to surroundings (By difference)	1175.1	15.37
	7650	100		7650	100

Example 6.31 A full load test was conducted on 2-stroke engine and the following results were obtained: Speed = 500 rpm, brake load = 500 N, imep = 3 bar, oil consumption = 5 kg/h, jacket water temperature rise = 35°C, jacket water flow rate = 7 kg/min; Air-fuel ratio by mass = 30, Exhaust gas temperature = 350°C; Room temperature = 25°C; Atmospheric pressure = 1 bar; cylinder diameter = 22 cm; stroke = 28 cm; brake diameter = 1.6 m; fuel calorific value = 42000 kJ/kg.

Proportion of hydrogen in mass in fuel = 15%

Specific heat of dry exhaust gases = 1.0 kJ/kg K

Specific heat of dry steam = 2.0 kJ/kg K

Determine

(a) the indicated thermal efficiency

(b) specific fuel consumption

(c) volumetric efficiency

Draw up the heat balance sheet.

Solution:

$$(W - S) = 500 \text{ N} = 0.5 \text{ kN}; R_m = \frac{1.6}{2} = 0.8 \text{ m}; N = 500 \text{ rpm}$$

$$P_{br} = 3 \text{ bar}; d = 0.22 \text{ m}; l = 0.28 \text{ m}; m_f = 5 \text{ kg/h}; CV = 42000 \text{ kJ/kg}$$

Testing and Performance of I.C. Engines

$$P = \frac{(W - S) R_m 2\pi N}{60} = \frac{0.5 \times 0.8 \times 2\pi \times 500}{60} = 20.94 \text{ kW}$$

$$P_{ind} = \frac{P_m \times l \times N}{60} = \frac{3 \times 10^2 \times \pi (0.22)^2 / 4 \times 0.28 \times 500}{60} = 26.61 \text{ kW}$$

(a) Indicated thermal efficiency:

$$\eta_{th} = \frac{P_{ind} \times 3600}{m_f \times CV} = \left[\frac{26.61 \times 3600}{5 \times 42000} \right] \times 100 = 45.61\% \text{ Ans.}$$

(b) Specific fuel consumption:

$$\text{bsfc} = \frac{m_f}{BP} = \frac{5}{20.94} = 0.2387 \text{ kg/kWh Ans.}$$

(c) Volumetric efficiency:

$$\text{Swept volume} = \frac{\pi d^2}{4} \cdot l \cdot N = \frac{\pi (0.22)^2}{4} \times 0.28 \times 500 = 5.3218 \text{ m}^3/\text{min}$$

$$\frac{m_a}{m_f} = 30$$

$$\therefore \text{mass of air } m_a = \frac{30 \times 5}{60} = 2.5 \text{ kg/min}$$

$$\text{Density of air, } \rho_a = \frac{P}{RT} = \frac{1 \times 10^2}{0.287 \times (25 + 273)} = 1.169 \text{ kg/m}^3$$

$$\text{Volume of air} = \frac{m_a}{\rho_a} = \frac{2.5}{1.169} = 2.1386 \text{ m}^3/\text{min}$$

$$\text{or Volume of air} = \frac{m_{air}}{P} = \frac{2.5 \times 0.287 \times 298}{100} \approx 2.1386 \text{ m}^3/\text{min}$$

$$\text{Volumetric efficiency} = \frac{\text{volume of air admitted}}{\text{swept volume}} = \frac{2.1386}{5.3218} \times 100 = 40.18\% \text{ Ans.}$$

Heat balance sheet (minute basis) :

Heat supplied :

$$\begin{aligned} \text{Heat supplied in fuel} &= m_f \times CV = \frac{5}{60} \times 42000 \\ &= 3500 \text{ kJ/min} \end{aligned}$$

Heat distribution :

(a) Heat to BP = $20.94 \times 60 = 1256.4 \text{ kJ/min}$

(b) Heat to exhaust gases :

$$= m_g \cdot C_p (t_g - t_f)$$

Where m_g = mass of exhaust gases

$$= m_g [(AFR + 1) - 9 H_2], \quad 9 H_2 = \text{mass of steam formed per kg of fuel}$$

$$= \frac{15}{60} [(30 + 1) - 9 \times 0.15] = 2.47 \text{ kg/min}$$

∴ Heat to exhaust gases

$$\begin{aligned} &= 2.47 \times 1.0 (350 - 25) \\ &= 802.75 \text{ kJ/min} \end{aligned}$$

Steam in the exhaust gas exist at 1 bar and 350°C

$$m_s = 9 H_2 \times m_f = 9 \times 0.15 \times \frac{5}{60} = 0.1125 \text{ kg/min}$$

(c) Heat to steam is exhaust gases,

$$\begin{aligned} &= m_s (h_{\text{sup}} - h_f) \\ \text{Where } h_{\text{sup}} &= h_f + C_p (t_{\text{sup}} - t_f) = 2675.4 + 2 (350 - 99.63) \\ &= 3176.14 \text{ kJ/kg} \end{aligned}$$

Heat to steam = $0.1125 (3176.14 - 4.2 \times 25)$

$$= 345.5 \text{ kJ/min}$$

(d) Heat to cooling water

$$= m_w \cdot C_w (t_2 - t_1) = 7 \times 4.2 (35) = 1029 \text{ kJ/min}$$

(e) Heat to surroundings (By difference)

$$\begin{aligned} &= 3500 - (1256.4 + 802.75 + 345.5 + 1029) \\ &= 66.35 \text{ kJ/min} \end{aligned}$$

Table 6.4
HEAT BALANCE SHEET

Heat supplied	kJ/min.	% age	Heat Distribution	kJ/min.	% age
Heat in fuel	3500	100	(a) Heat to BP	1256.4	35.89
			(b) Heat to exhaust gases	802.75	22.93
			(c) Heat to steam in exhaust gases	345.5	9.87
			(d) Heat to cooling water	1029.0	29.4
			(e) Heat to surroundings (By difference)	66.35	1.91
	3500	100		3500	100

SUMMARY

$$1. \text{ IMEP} = \frac{\text{area of indicator diagram}}{\text{length of indicator diagram}} \times \text{spring number} = \frac{\text{work done in cycle}}{\text{swept volume}}$$

2. Indicated power

$$\begin{aligned} \text{IP} &= \frac{P_{\text{im}} \cdot a \cdot l \cdot n}{60} ; n = \frac{N}{2} \text{ for 4-stroke engine} \\ &= N \text{ for 2-stroke engine} \end{aligned}$$

3. Brake power

$$\text{BP} = \frac{(W - S) R_m \cdot 2\pi N}{60} = \frac{2\pi NT}{60}$$

4. Efficiencies :

$$(a) \eta_{\text{mech}} = \frac{\text{BP}}{\text{IP}} = \frac{\eta_{b,th}}{\eta_{i,th}} = \frac{b_{\text{mep}}}{i_{\text{mep}}}$$

$$(b) \eta_{b,th} = \frac{\text{BP} \times 60}{m_f \times CV}$$

$$(c) \eta_{i,th} = \frac{\text{IP} \times 60}{m_f \times CV}, \text{ Where } m_f = \text{mass of fuel/min}$$

$$(d) \eta_{\text{rel}} = \frac{\text{thermal efficiency}}{\text{air standard efficiency}}$$



LUBRICATION

UNIT-2 (4)
(108)

Whenever two metallic surfaces move over each other under direct contact dry or solid friction is produced. This is due to the irregularities on the two surfaces interlocking each other. The dry friction thus created produces a lot of heat & results in wear of the metal surface.

Introducing the lubricating oil between the moving parts to reduce the friction is termed as Lubrication.

Objectives of Lubrication

1. To reduce friction between moving parts } primary objectives
2. To reduce wear of the moving parts
3. To act as cooling medium for removing heat
4. To keep engine parts clean
5. To form a good seal between piston rings and cylinder wall
6. To absorb shocks between bearings and other engine parts thus reducing the engine noise and extending the engine life.
7. To prevent deposition of carbon, soot.

Lubrication Systems

1. Engine lubrication
2. Chassis lubrication

→ A single system lubricates the entire engine of all present day motor vehicles. Only equipments outside the engine like starter, water pump... are lubricated separately.

① ②

Lubricant

→ In chassis lubrication system each lubricating point is lubricated separately. About '34' points are to be lubricated in the chassis of modern car. (It may vary from 21 to 45 depending on design).

Required properties of Lubricants

1. Viscosity
2. Flash & Fire point
3. Pour point
4. Resistant against oxidation
5. Physical stability
6. Chemical stability

Types of Lubricants

1. Solid — Graphite, mica, soap stone
2. Liquid — mineral oil, Vegetable oil, animal oil
3. semisolid — Grease.

Specifications of Lubricating oil

SAE — society of Automotive Engineers

API — American petroleum Institute

SAE 10, SAE 20, SAE 30 - - -

Engine parts which are to be Lubricated

- | | |
|-----------------------------|----------------------------------|
| 1. Main crankshaft bearings | 5. Piston rings & Cylinder walls |
| 2. G & end bearings | 6. Timing gear |
| 3. Small end bearings | 7. valve mechanism. |
| 4. camshaft bearings | |

Lubricating Systems

UNIT-2

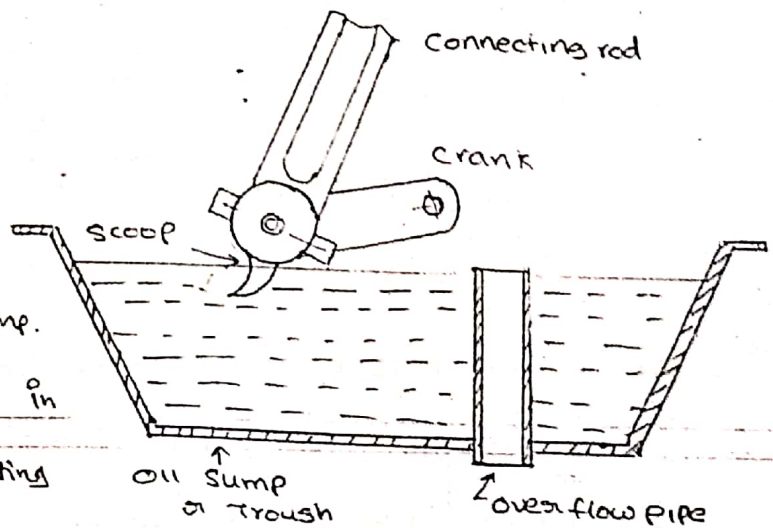
1. Petrol system
2. Splash system
3. Pressure system { Wet sump
Dry sump

Petrol system (Mist lubrication)

This system of lubrication is generally adopted in 2-stroke petrol engine. It is the simplest form of lubricating system. It does not consist of any separate parts like oil pump. But the lubricating oil is mixed in to petrol itself while filling in the petrol tank of the vehicle, in a specified ratio. When the fuel goes into the crank chamber during the engine operation, the oil particles go deep in to the bearing surfaces and lubricate them.

Splash system

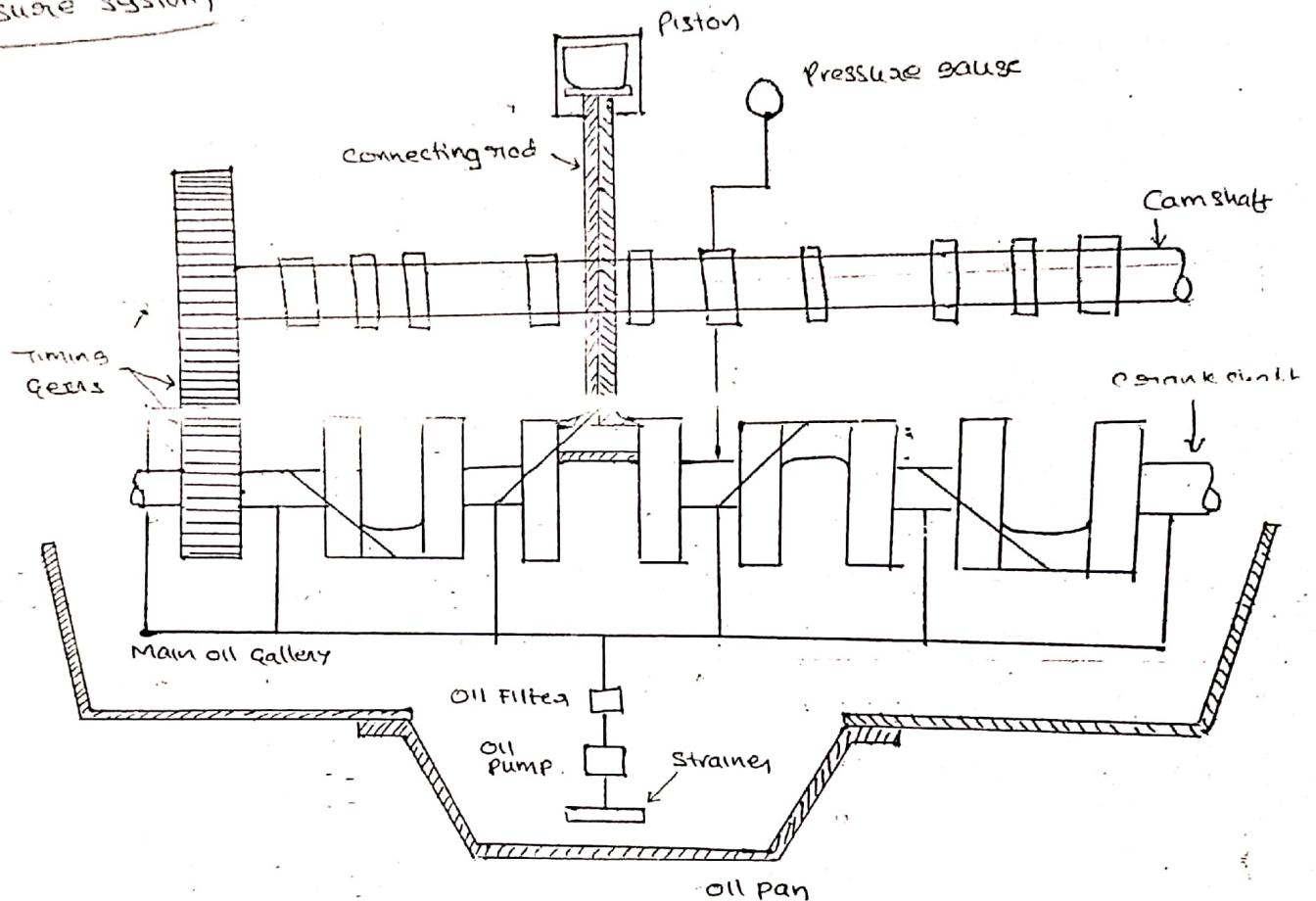
In this system of lubrication, the lubricating oil is stored in an oil sump. A scoop or dipper is made in the lowest part of connecting rod.



When the engine runs, the dipper dips in the oil once in every revolution of crank shaft and cause the oil to splash on the cylinder walls. This action affects the lubrication of the engine walls, p. crank shaft bearings.

splash system mostly works in connection with pressure system
 an engine, some parts being lubricated by splash system and
 the other by pressure system

Pressure System



In this system of lubrication, the engine parts are lubricated under pressure feed. The lubricating oil is stored in separate tank or the sump, from where an oil pump takes oil through a strainer and delivers it through a filter to the main oil gallery at a pressure of 200 to 400 kPa. The oil from the main oil gallery goes to the main bearings, from where some of it lubricates the main bearings, falls back to the sump, some is splashed to lubricate the cylinder walls and

remaining goes through a hole to the crank pin. From the crank pin it goes to the piston pin through a hole in the connecting rod web, where it lubricates the piston rings.

For lubricating cam shaft and timing gears, the oil is led through a separate oil line from the oil gallery.

An oil pressure gauge at the instrument panel indicates the oil pressure in the system. Oil filters & strainers in the system clean off the dust & metal particles.

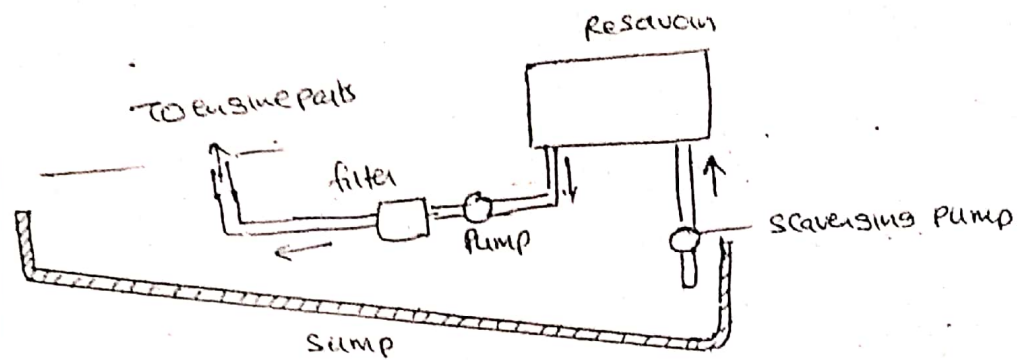
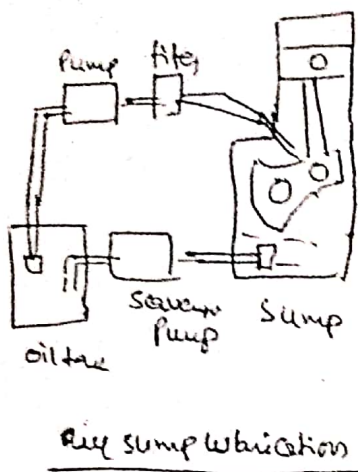
Dry Sump System

A system in which the lubricating oil is stored in the oil pump is called wet sump system. But the system in which the sump is kept dry and the oil collected in the sump after lubrication is taken out for filtering and cooling and again pumped in to the bearings is called dry sump system.

The dry sump lubricating system has a separate oil tank, relatively small oil pan, and two oil pumps. The one oil pump is used to feed oil and other to collect oil.

The dry sump system is generally used where the up and down movement of vehicle is very frequent.

In this system there is no break down of oil supply.



Parts of lubricating system

1. oil sump or tank
2. oil pump
3. oil cooler
4. oil filter & strainer
5. oil pressure gauge

Oil sump:

It is the lowest part of crank chamber. It provides a covering for the crank shaft and contain oil in it.

It is usually made of steel pressing, sometimes made of aluminium or cast iron.

Oil pump

The function of oil pump is to supply oil under pressure to various engine parts.

Types: 1. Gear pump 2. Rotor pump 3. Plunger pump 4. Vane

(X) (6)

oil cooler

The purpose of an oil cooler is to cool the lubricating oil in heavy duty engines where oil temperatures become quite high. Because the viscosity of the oil decreases with temperature rise the oil must be kept cold in the lubricating system.

An oil cooler is just like a simple heat exchanger. The oil may be cooled in it either by cold water from radiator or the air stream.

Oil filter

1. Bypass system

2. Full flow system

→ In bypass system, the whole of the oil does not pass through the filter at the same time, but some of the oil without being filtered goes to the bearings. Remaining oil passes through the filter and then goes to the bearings. When the engine is run continuously for a long period, the whole oil is however filtered.

→ In full flow system, the whole oil passes first through the filter, then goes to the bearings. If the filter is closed due to any reason, the system fails completely.

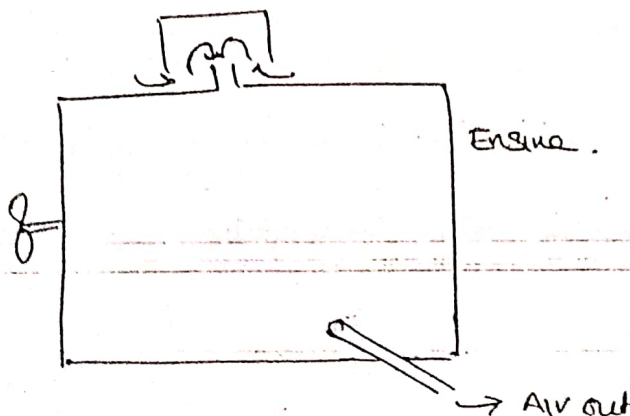
Crank case ventilation

Sometimes the products of combustion may leak in to the crank case through the piston rings. Combustion products containing water vapour, sulphuric acid, fuel vapour. When they leaked in to the crank case they may dilute the lubricating oil and may corrode the crank case.

Crank case ventilation removes all these leaked gases from crank case before they mix with lubricating oil.

In this Air is drawn in to the crank case through the breather cap and discharged through an outlet tube at the side of the engine. The circulation of air in the crank case takes place due to the motion of the vehicle.

In a positive crank case ventilation, the crank case vapours are returned to the engine through the intake manifold.



COOLING SYSTEM

UNIT-2 C.

(1411)

All the heat produced by the combustion of fuel in the cylinders is not converted into useful power at the crank shaft.

A typical distribution of heat energy is given below

Useful work at crankshaft = 25 %

Loss to cylinder walls = 30 %

Loss in Exhaust Gases = 35 %

Loss in friction = 10 %

If the heat from cylinder walls is not removed it would result

1. burning of lubricating oil

2. Distortion of parts

3. Preignition

So that excess heat from cylinder walls must be removed.

→ Cooling system removes the excess heat from cylinder walls and maintains the engine at operating temperature

→ However, cooling beyond optimum limits is not desirable, because it decreases overall efficiency.

Methods of cooling

1. Air cooling (Direct cooling)

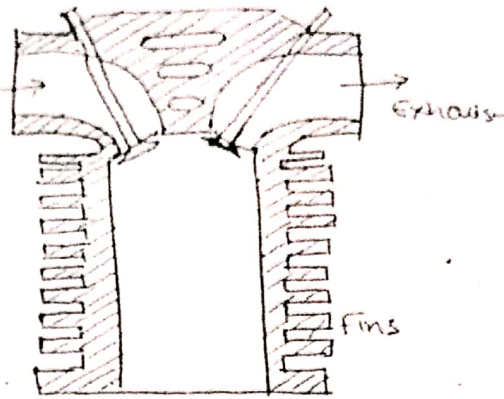
2. Water cooling (Indirect cooling)

Air cooling

In Air cooling system Air is directly flows over the heated metal surface (i.e. cylinder) from where heat is to be removed. The heat dissipated in air cooling depends upon the following factors.

①

1. surface area of metal
2. mass flow rate of air
3. temperature difference
4. conductivity of metal



thus surface area of cylinder is increased by providing fins or ribs around the cylinder.

Advantages of Air cooling

1. Air cooled engines are lighter because of the absence of the radiator and coolant, jackets
2. can be operated in extreme cold climates, where water may freeze
3. low maintenance
4. Air cooled engines get warmed up faster than water cooled engines

Disadvantages

1. It cannot provide uniform cooling around the engine cylinder
2. more noisy in operation
3. coeff of heat transfer of air is less than water.

Water cooling

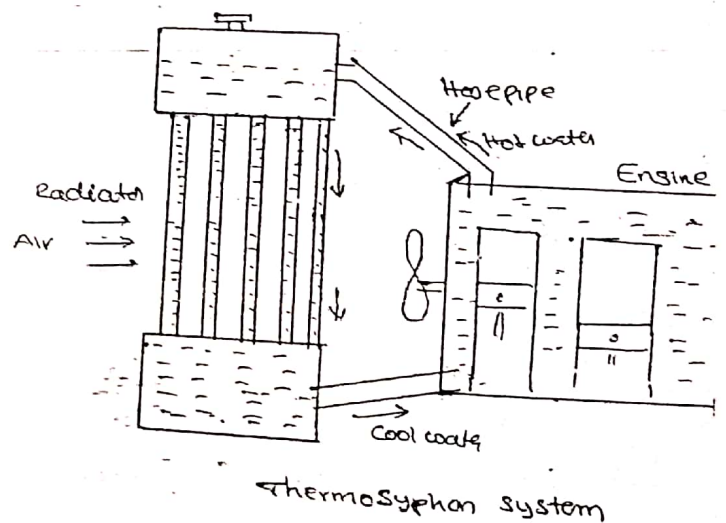
In water cooling system: water is used as a cooling medium. The water jackets are provided around the each engine cylinder, cylinder head, valve openings. The water circulates through these jackets and takes away the heat. The heat flows from the cylinder walls to the water, which goes to the radiator where it is cooled by air through an array of cooling fins. Then the coolant is recirculated to the engine.

types

1. Thermosyphon system
2. Pump circulation system.

Thermosyphon system

- It consists of a radiator connected to engine through flexible hoses.
- In this system circulation of water is obtained from the difference in densities of the hot and cold regions of the cooling water.



Advantages

1. Simple in construction
2. Low initial cost

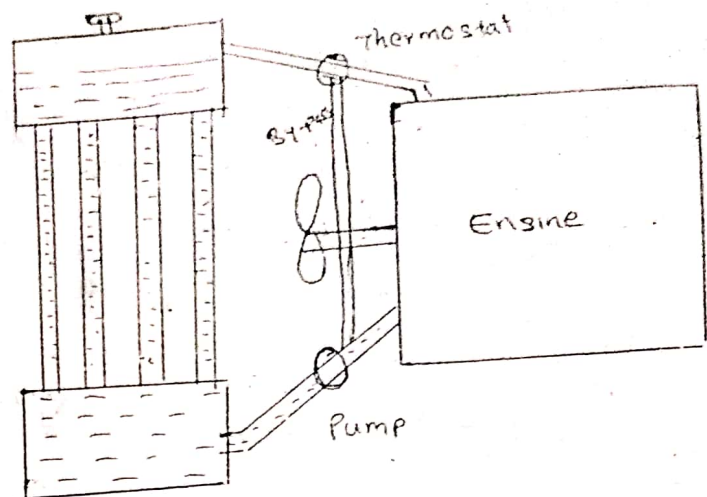
Disadvantages

1. Slow cooling
2. Always it should maintain certain level of water
3. The quantity of water required is large and hence it takes more time to reach the operating temperatures.

Pump Circulation System

→ It is similar to thermosiphon system. The only difference is, it has a pump to circulate the water and a thermostat to control the flow of coolant.

→ Pump is driven by means of a belt from the engine crank shaft.



→

1. Circulation of coolant is proportional to both load & speed

2. Better cooling

3. Radiator can be placed anywhere (Need not place above the engine level)

Disadvantages

1. More costly

2. Complicated

Components of Water Cooling System

1. Radiators

2. thermostat

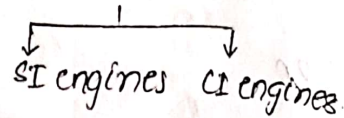
3. Pump

4. Fan

5. Water jackets

UNIT-3: combustion in IC Engines.

Combustion:



It is a chemical process in which certain elements of the fuel (Carbon, Hydrogen) combine with oxygen causing an increase in temperature of the gases.

→ the conditions necessary for combustion are:

- (1) the presence of combustible mixture and some means of initiating the combustion process

→ combustion in SI engines:

We need petrol + air mixture.

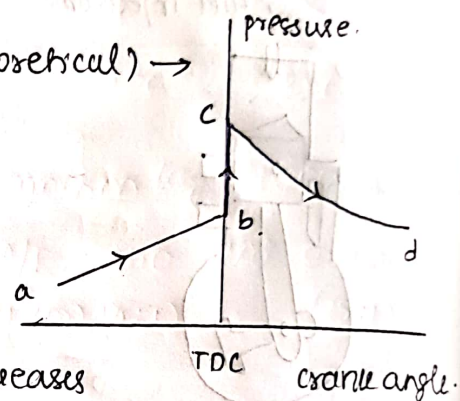
and ignition by spark. (theoretical) →

piston from BDC to TDC

compression, at TDC.

Spark is given at constant volume, so, pressure increases

so, bc remains as pressure increases



Theoretical pressure vs crank angle diagram

at CD → expansion

Actual pressure vs crank angle diagram:

** stages of combustion in SI engine:

Stage I (a-b): Ignition loss.

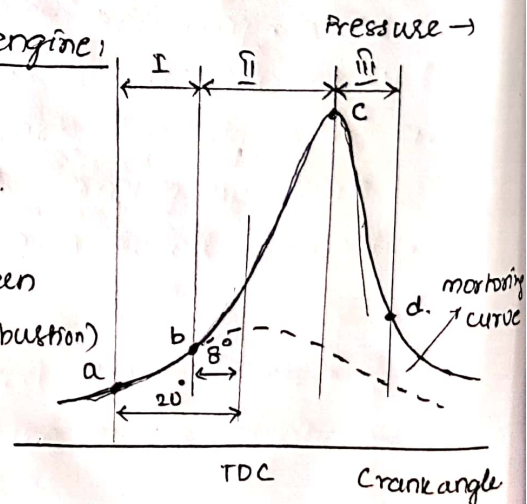
spark is given at a, and at b.

combustion starts. the time.

gap between the points (between spark ignition and start of combustion) known as ignition lag/delay period.

a - spark initiation

b - start of combustion



- (1) physical delay - entering into the first layer of particles the chemical delay and then flame should be generated)

In this lag - chemical reactions take place.

• stage I (b-c): Propagation of flame.

b to c propagation of flame takes place, flame travels from one end to another end and c is at highest pressure.

• stage II (c-d): After burning.

During expansion stroke, fuel burns (only heat increases but pressure doesn't change) Burning of air-fuel mixture during expansion stroke.

• motoring curve: (no ignition spark).

then the curve changes its direction and then after pressure decreases after B point (obtained by hand movement).

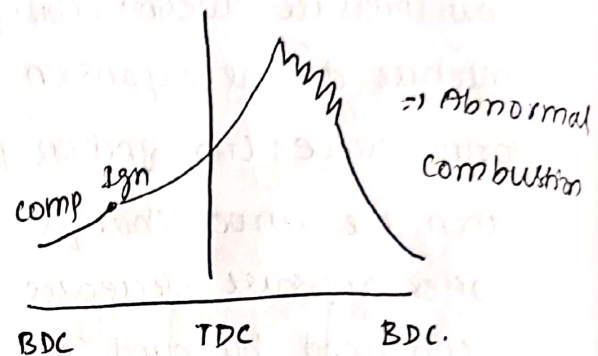
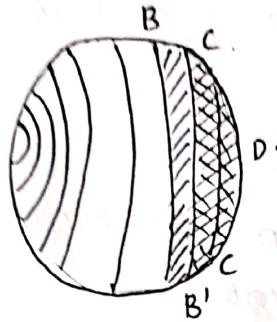
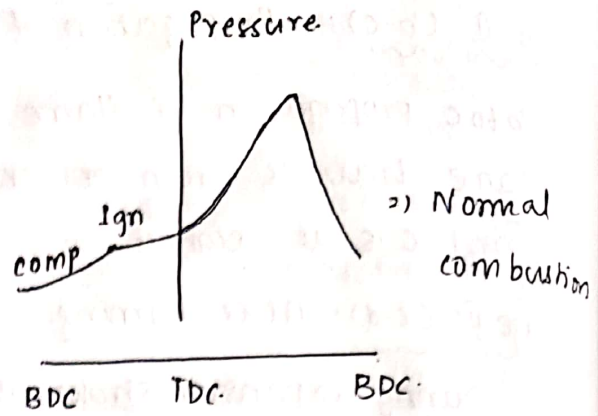
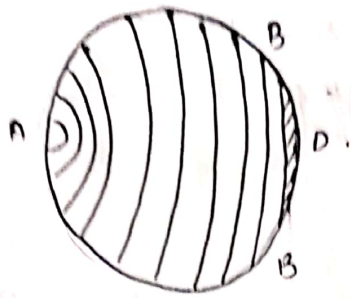
* Effect of engine variables on Ignition lag:

- | | | ↑ | ↓ |
|--------------------------------------|------|---|---|
| 1) compressed gas temp and pressure. | | ↑ | ↓ |
| 2) self Ignition temp. of fuel. | | ↑ | ↑ |
| 3) Air Fuel ratio. | rich | ↑ | ↓ |
| 4) Turbulence. | | ↑ | ↓ |

* Effect of Engine variables on flame preparation:

- | | | ↑ | ↓ |
|-------------------------|--------|---|---|
| 1) Turbulence. | - | ↑ | ↑ |
| 2) A/F ratio | - rich | ↑ | ↑ |
| 3) comp. ratio. | - | ↑ | ↑ |
| 4) Intake press & temp. | - | ↑ | ↑ |
| 5) Engine speed | - | ↑ | ↓ |
| 6) Engine size | - | ↑ | ↓ |

* Abnormal conditions:



- If A/F ratio is rich the Ignition lag decreases
- The ratio between actual A/F ratio and stoichiometric ratio is Equivalence ratio. $(13:1 \text{ (or) } 14:1)$ $(15:1)$ generally in SI engine
- When two pressure wave collide in the engine, lead to knocking which is undesirable. Due to this engine gets heated up and the engine gets damaged gradually
- Abnormal combustion involves knocking, where ~~at one end~~ when the airfuel mixture gets burned up (at one end) due to ignition of spark leading to the end charge reaching its SIT and forming up flame from the other end and leading to a sound formation called knocking
- In normal combustion, the end charge doesn't lead to its SIT as observed from the graphs mentioned

* Abnormal combustion in SI engine,

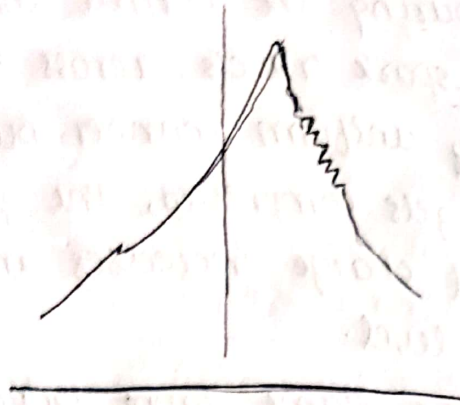
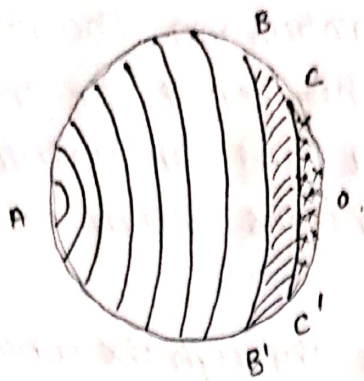
During the normal combustion, the flame initiated by the spark travels across the combustion chamber in a fairly uniform manner. During this period, the end charge gets compressed, the pressure and temperature of the end charge increases and remains below certain critical level.

The flame front propagates through the whole charge and completes the combustion smoothly. But if the pressure and temp. of the end gas attains the value above critical limit (SIT), the (charge) end charge ignites spontaneously before the flame front reaches to it. This is called abnormal combustion.

- When two flames collide with each other, an audible sound is generated known as knocking.
- SIT of fuel should be high to avoid knocking.
- compression ratio of fuel should be low to reduce knocking.
- comp ratio of petrol - 10:1
diesel - 16:1 (or) 22:1 } generally

(IC engines by ganesan)

* Abnormal combustion in SI engines:



* Effect of Engine Variables on knock:

Density factor: (more ratio, more efficiency
or per Otto, for upto 10:1)

			<u>Knocking</u>
1. compression ratio	-	↑	↑
2. Mass of induced charge (more A/F mixture is sent in - known super charging)	-	↑	↑
3. Inlet temp. of mixture	-	↑	↑
4. Temp. of the combustion chamber wall.	-	↑	↑

Time factors:

			<u>Knocking</u>
1. Turbulence	-	↑	↓
2. Engine speed	-	↑	↓
3. Flame travel distance.	-	↑	↓
4. Engine size	-	↑	↑
5. combustion chamber shape.	-	should be compact.	
6. location of spark plug	-	should be placed centrally.	

* Control of Detonation:

- 1) Fuel should have high SI
- 2) F/A ratio increases (about 10%)
- 3) Increase flame speed.
- 4) Remove some heat from the end charge.
- 5) Reducing the flame travel distance
- 6) Avoid high compression.

* super charge is not appreciable for SI engine.

* comp. ratio $\propto \eta$ of engine, so $\uparrow$ in this leads to knocking after 10:1 comp ratio increases.

* Turbulance means rotation of gases in the chamber. in this case flame speed increases.

* Flame travel distance increases, knocking increases.

* We limit the engine size to 150cc, 300cc means we go for multiple cylinders.

* We need to have compact combustion chamber shape.

* Effect of Detonation (or) knocking:

1) overheating

2) Pre-ignition.

3) Loss of power.

4) mechanical Damage.

5) Excessive Noise.

* grade/quality of petrol can be measured by octane number

C_8H_{18} - Iso octane

C_7H_{16} - n-heptane.

} - constitute petrol

octane no. is $\frac{\text{Isooctance}}{\text{Isooctance} + \text{n-heptane}}$

Isooctance has anti-knocking capability

n-heptane has high knocking capability

- * TEL (Tetra ethyl lead) - increases the self ignition temp. this decreases the knocking.

amyl nitrate is also used.

- * quality of petrol is measured by octane number

- * (1) what is abnormal comb. • knocking phenomenon
- 2) factor effecting knocking • comparison of knock in CI and SI
- 3) Factor effecting Ignition lag and flame speed
- 4) Combustion chambers in SI and CI.

- * Octane number is the ratio of percentage of iso-octane in the mixture (iso-octane and normal octane, which produces the same knocking intensity as the given fuel.

isooctane - good anti-knocking property (C_8H_{18})
octane number - 100.

n-heptane - poor anti-knocking property (C_7H_{16})
octane number - 0.

- * combustion chambers: (for SI engine)

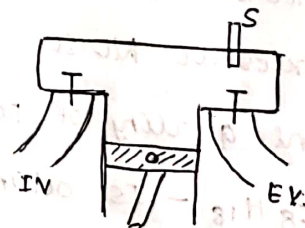
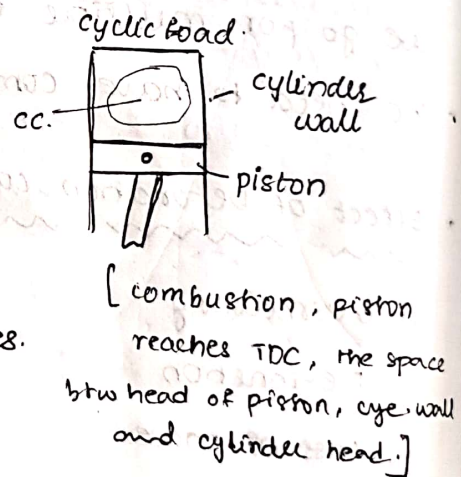
Requirements:

- 1) High power output and thermal efficiency
- 2) smooth operations.
- 3) Reducing knocking possibilities.
- 4) Provide good turbulence.

→ types:

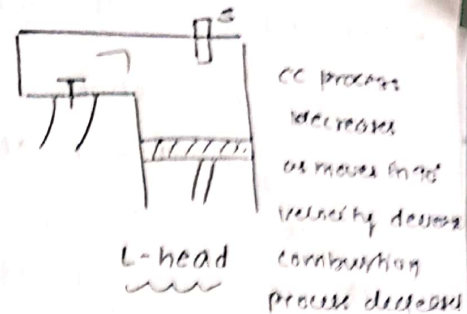
T-head:

- used in early stages.
- Requires two cam shafts.
- knocking tendency is high
- From manufacture's point of view 2 cam shafts is a disadvantage.



L-head:

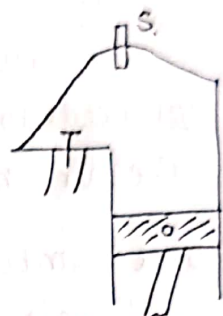
- It is modified 'T' head.
- two valves on the same side.
- operated by single cam shaft.
- cylinder head can be removed without disturbing valves.



I-head:

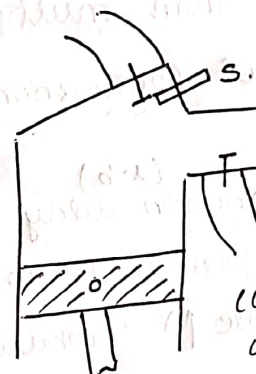
- Both valves are located on cylinder head.
- less surface to volume ratio
- less flame travel distance.
- high η_{vol} .
- widely used.

circular in shape.



F-head:

- compromise b/w L and I head type.
- one valve in cylinder head and other in cylinder block.
- main disadvantage, requires '2' camshafts.
- valve mechanism is complicated.



* For Ricardo's (has less tendency to knock)
In L-head type air flow / air-fuel mixture must take 2 right angle turns to enter into the cylinder, due to this velocity decreases and combustion process becomes slow. To create turbulence, the L-head is modified as Ricardo's turbulent head.

• Combustion in CI engines:

time of injection: time required for injection.

CI engine - It is a heterogeneous mixture.

at fuel is injected, conc of mixture may be more at one place and less at other place.

delays - $\left\{ \begin{array}{l} \text{Physical delay (time taken by the fuel with} \\ \text{air and the process of atomisation} \\ \text{and vapourisation)} \\ \text{Chemical delay} \end{array} \right.$

(fuel needs to mix of / with

the air, then the time taken to burn the fuel).

- Once combustion starts, the other fuel particles which is injected takes less time for combustion.

CI engine - combustion starts from multiple points

• Stages of Combustion in CI engine:

- (a-b) 1. Ignition delay
- (b-c) 2. period of rapid combustion (pressure ↑)
- (c-d) 3. period of controlled combustion.
- (d-e) 4. period of after burning

a - start of injection

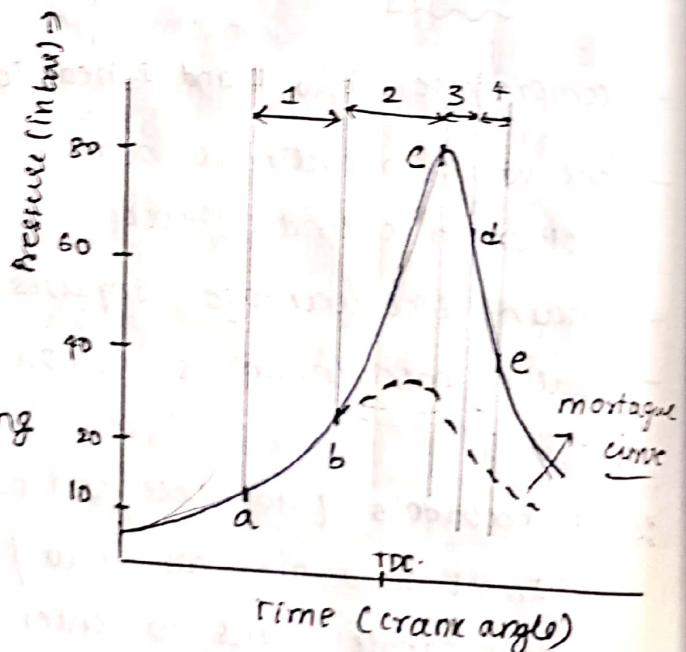
b → start of combustion.

c → peak pressure

d → end of Injection.

e → end of combustion.

(c-d) ⇒ movement of fuel charge into the flame.
(easy combustion)



Factors effecting Delay period:

Ignition delay $\left\{ \begin{array}{l} \text{Physical delay} \\ \text{chemical delay} \end{array} \right.$

In physical delay, atomisation, vapourisation, mixing with air and raising to a certain temp.

In chemical delay, chemical reaction starts and ignition takes place.

factor effecting delay period:

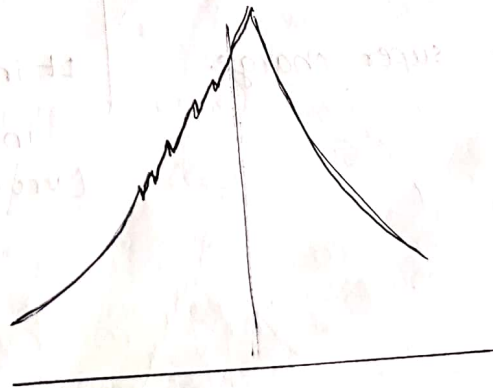
- | | | | |
|-------------------------------|-------------------------|------|------------------|
| 1. comp ratio | ↑ | DP ↓ | |
| 2. Inlet P & temp of air. | ↑ | ↓ | |
| 3. Engine speed. | ↑ | ↓ | |
| 4. Quality of fuel. | ↑ | ↓ | = Centane number |
| 5. combustion temp wall temp. | ↑ | ↓ | |
| 6. SIT of fuel. | ↑ | ↑ | |
| 7. Ignition temp. | — 12° to 20° before TDC | | |
| Injection time. | | | |

* Knocking in CI engine:

If the delay period ↑
the more amount of
fuel particles are accumulated
in the cylinder.

(if delay period ↓, the
fuel particles could burn).

In CI engines, normal combustion process leads to smooth pressure rise. If the ignition delay of the fuel is long a considerable part of fuel accumulates in the combustion chamber. When the ignition begins; it will suddenly burn causing rapid pressure and pressure fluctuates, this will results in audible knocks.



→ The quality of diesel can be measured by using cetane number.

** to reduce knocking, delay period ↓

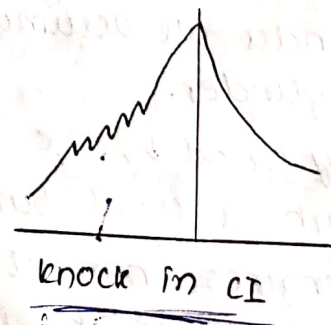
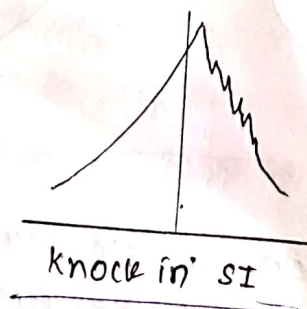
$$\text{cetane number} = \frac{\text{cetane}}{\text{cetane} + \alpha\text{-methyl-naphthalene}}$$

Cetane number for cetane is 100 and for α -methyl naphthalene is zero.

• cetane number for high speed diesel engine is around 45 to 54.

* comparison between knocking in SI and CI engines

1. comp. ratio.	↓ low	↑ High.
2. Inlet temp.	↓ low	↑ high
3. Inlet pressure.	↓ low	↑ high
4. SIT of fuel.	↑ more	↓ less
5. Ignition delay	more	less
6. cylinder size	less	more
7. Combustion wall temperature.	less	more
8. super charge	It increases knock in SI. (promotes knock)	It decreases knock in CI. (reduces knock)



* super charge promotes knocking in SI and decreases knock in CI.

* knocking takes place after combustion in SI engine and before in CI engines

* combustion chambers in CI engines:

methods to create air motion (turbulence) in combustion chamber.

1. suction swirl.
2. Compression turbulence
3. squish.
4. combustion turbulence.

* CI engines combustion chambers:

methods to create air motion (turbulence) in combustion chamber.

(1) Suction Swirl. - enters with a swirl.

(2) compression turbulence. - turbulence is created.

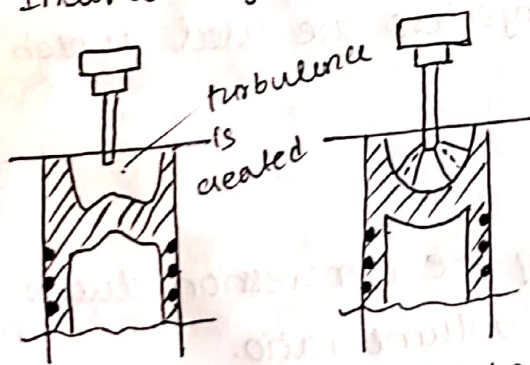
(3) squish. - ordinary motion of air, air fuel mixture at walls needs to come to the centre of cylinder.

(4) combustion turbulence.

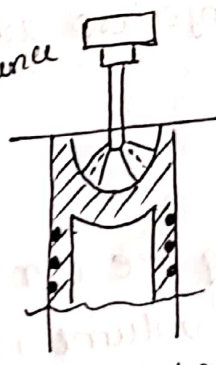
↳ due to combustion, piston moves down, turbulence is created.

Types:

1. Direct Injection combustion. - Direct injection of fuel into the chamber. (in cylinder only).
2. Indirect Injection combustion.



shallow depth chamber.



Hemi-spherical chamber



cylindrical chamber (nucut cylinder)

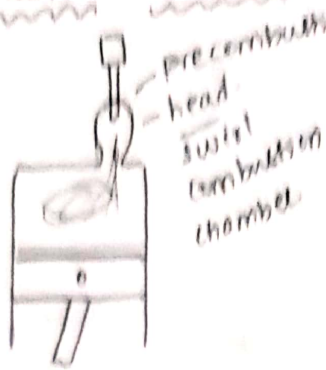


Toroidal chamber.

Direct (or) open combustion chambers.

C.

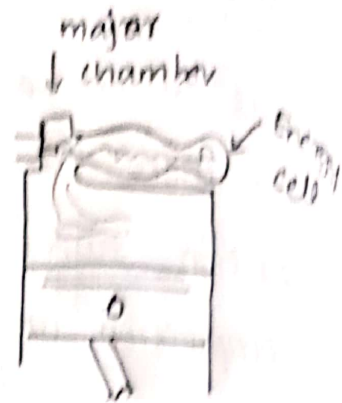
* Indirect combustion chambers



swirl chamber



precombustion chamber



air cell chamber
(better mixing)
(complicated design)

(combustion chamber is divided into two cylinder block and cylinder head

divided head.

⇒ write advantages and disadvantages of each from TB

* Open Combustion chambers:

- It is also called direct injection chambers
- In this type, the entire volume of combustion chamber is located in main cylinder and the fuel is injected into this volume.
- Induction swirl is obtained by either by careful formation of intake passage (or) by masking a portion of circumference of the inlet valves.
- Generally a multi-hole type injectors are used at high pressure (at 100 bar)

advantages:

- 1) minimum heat losses during the compression, due to a lower ~~area~~ surface area to volume ratio.
- 2) No cold starting problems.
- 3) Fine atomisation because of multi-hole nozzle.

Disadvantages:

- 1) high fuel injection pressures complex design of fuel injection pumps.
- 2) accurate metering of fuel is necessary.

* shallow depth: Depth of the cavity provided in the piston is quite small; usually adapted for large engines running at low speeds.

* hemispherical: - has hemispherical cavity.

* cylindrical: It is used in recent diesel engines, the cavity is in the form of truncated cone with base angle 30° .

* toroidal: better squish motion is produced.

Indirect:

In this type, the combustion space is divided into two parts, one part in the main cylinder and other cylinder head.

The fuel injection is takes place in the part that located in the cylinder head.

* Swirl chamber: consists of a spherical shaped chamber located in the cylinder head.

Into this chamber about 50% of air is transferred during the compression stroke.

When the air enters into the ~~swirl~~ ^{Spherical} swirl chamber, a strong swirl is produced.

Suitable for high speed engines

High fuel consumption.

Pre-combustion:

a pre-combustion chamber located in the cylinder head connected to the main chamber through a number of small holes.

It contains a volume, 40% of the total combustion space.

Tendency to knock is minimum.

* Air-cell chamber:

clearance volume is divided into two parts

(1) In the main cylinder

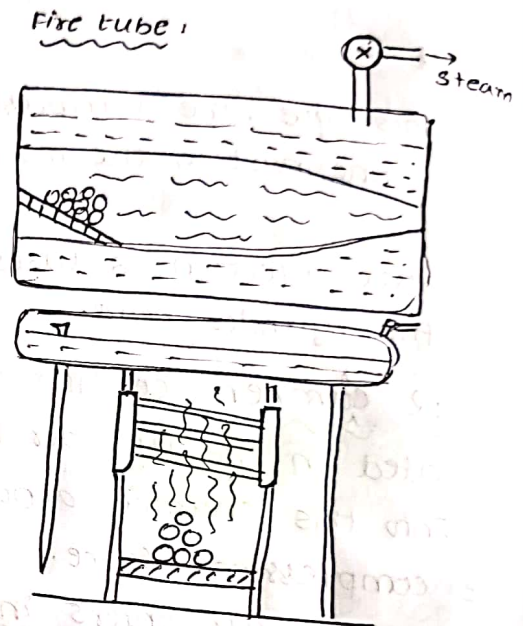
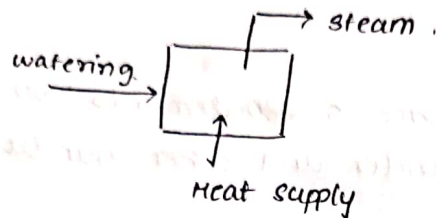
(2) ^{It is} called energy cell

energy cell is divided into two i.e major and minor

- rapid mixing of air and fuel takes place
- complete combustion takes place
- Design is complication.

UNIT-4:

* Steam Boilers:



Types:

- (1) Fire tube boiler
 - (2) Water tube boiler.
 - ↳ around tube hot gases are moving.
- Ex: bab. cock & wilcox, Yarrow, stirling and all. modern Modern.

- (1) Fire: Ex: Cochran, Cornish, simple vertical body, locomotive boiler, Lancashire boiler, Scotch marine boiler.

* classification of boilers:

- (I) Acc. to content in the tubes.
 - (1) Fire tube.
 - (2) Water tube.
- (II) Based on location of furnace.
 - (1) Internally fired
 - (2) Externally fired

Based on axis of shell.

i) vertical

ii) horizontal.

Based on circulation of water.

i) Natural circulation.

ii) Forced circulation.

Based on application

i) stationary

ii) Mobile.

* Cochran Boiler :

vertical

evaporating
capacity /

* Steam generating

capacity = 3000 kg/hr

(max: 4000 kg/hr)

* Dia of tubes

= 60 mm

* Shell dia = 2m.

* Working pressure = 6.5 to 15 bar.

* No. of tubes = 165

* shell life height = 4.42m

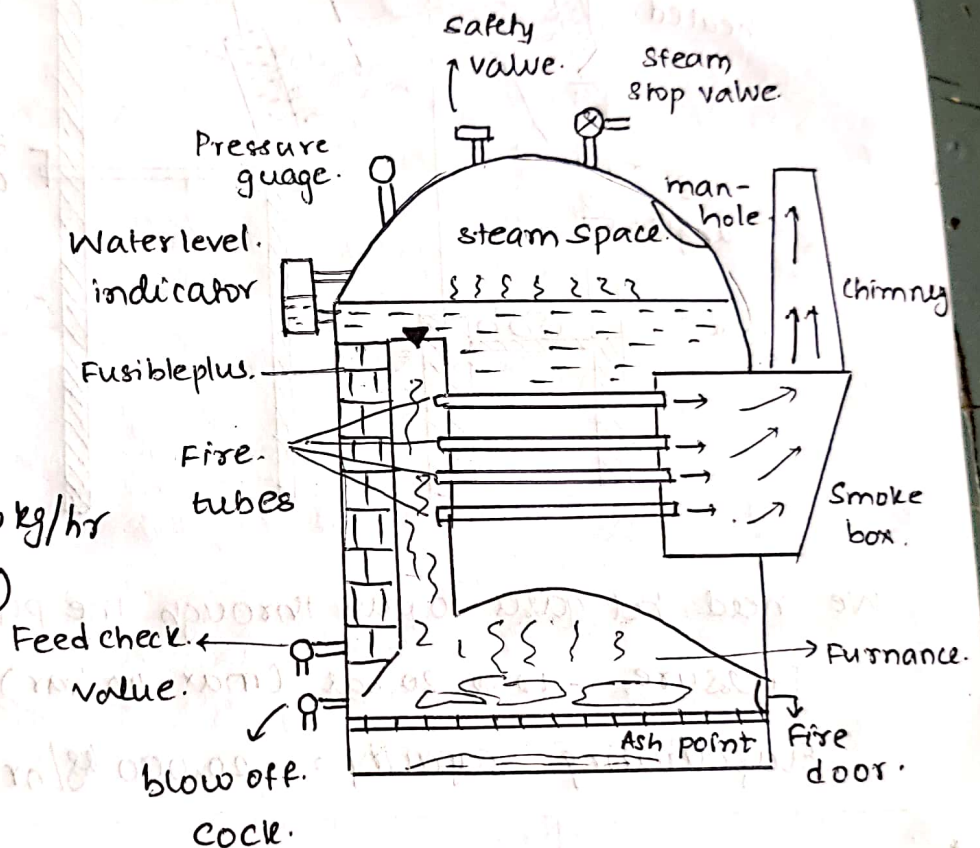
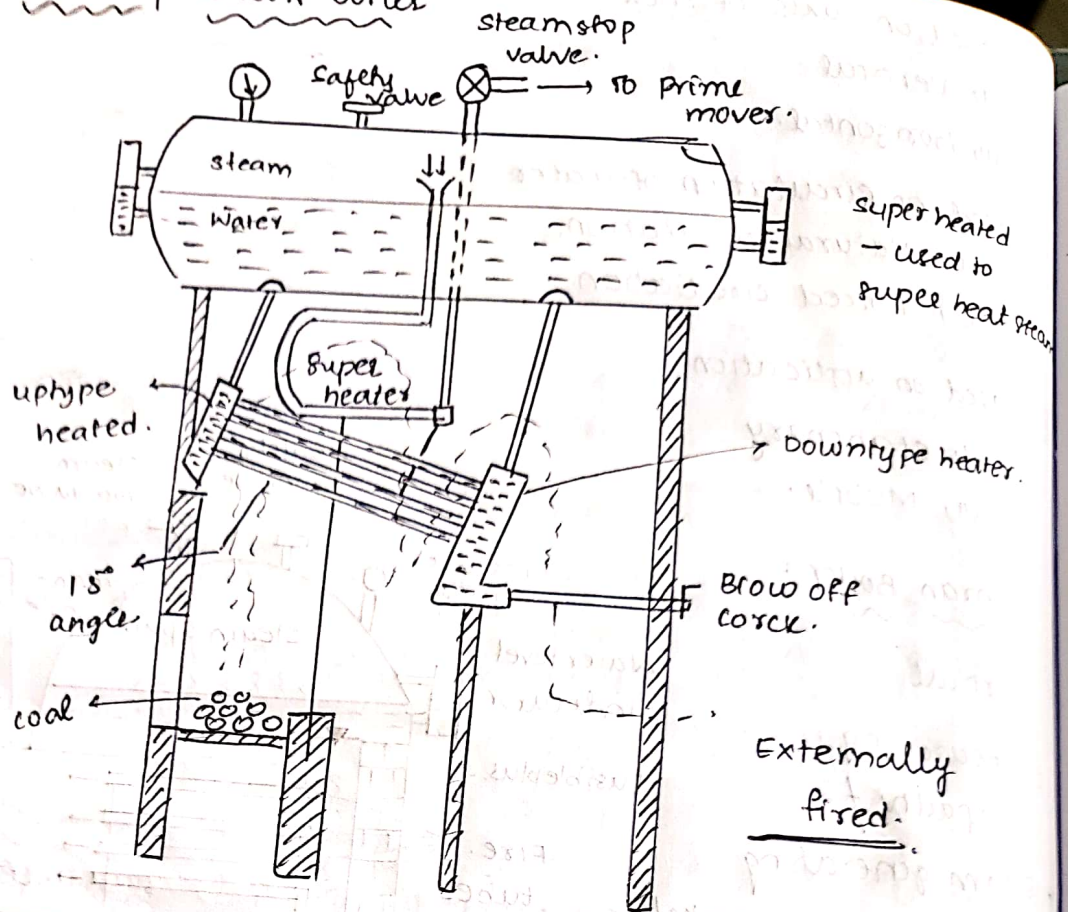


Diagram of water boilers



We need hot gases to pass through the pipe.

Pressure - 10 to 20 bar (max. 40 bar)

Evaporating capacity = 20,000 kg/hr.

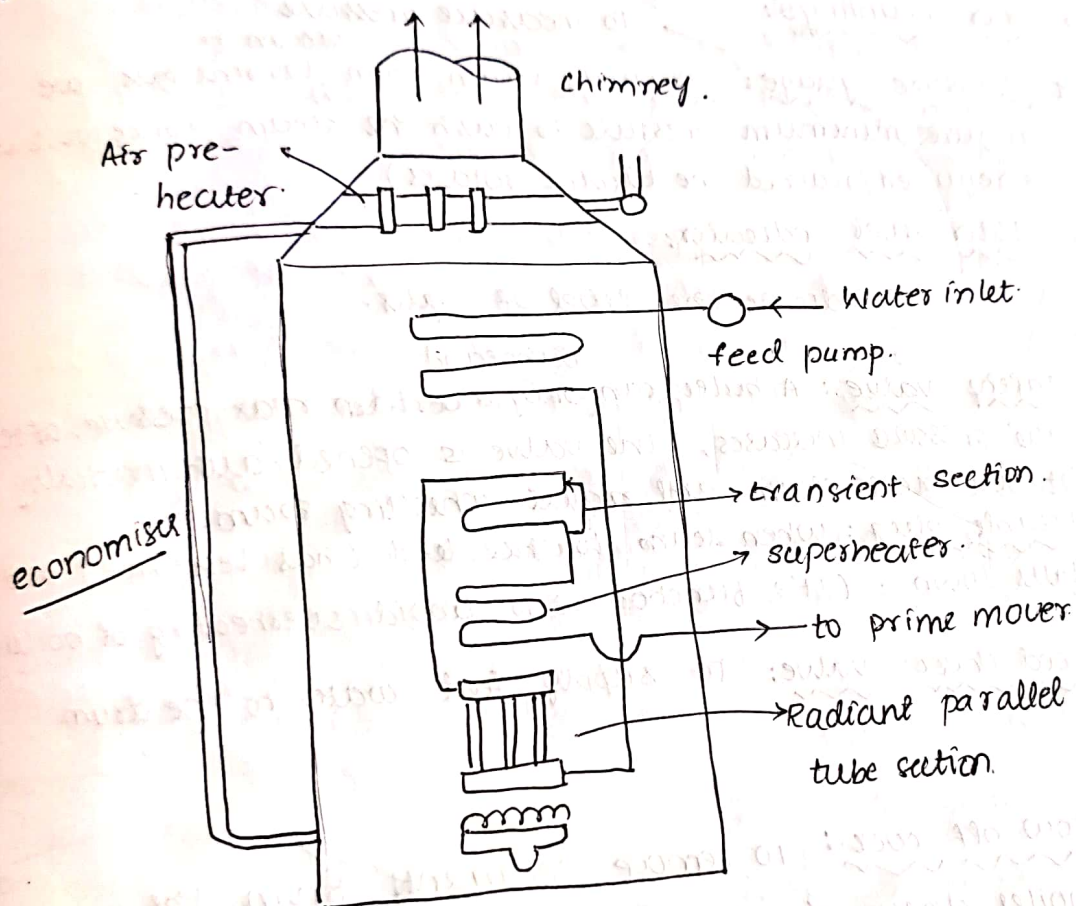
* Criterion.	Fire tube boiler	water tube boilers
(1) Flow of hot gases & water	Hot gases - inside the tube water - around the tubes	water - inside the tubes. Hot gases - around the tubes
(2) location of furnace	Inside the boiler	Outside the boiler
(3) Floor area	Bigger in size (requires more floor area)	Smaller size (requires less area)
(4) capacity	max capacity - 10,000 kg/hr	max - 5 lakhs kg/hr
(5) pressure range	15 to 25 bar	165 to 200 bar
(6) efficiency	upto 80%.	upto 92%.
(7) safety aspects.	more	less.

* modern boilers: (High performance boilers)

1. method of circulation of water and steam.
2. Arrangement of drum & tubing.
3. Improved method of heating - by using Economiser

* Economiser is a device that rises the water temp by using the heat from the chimney. superheater.
Air preheater.

* Benson Boilers / critical boiler / once through boiler



also known as drumless boiler

superheated boiler $\Rightarrow p > 221.2 \text{ bar}$
 $\Rightarrow$ direct steam is produced.

- bubble formation is avoided.

- more efficiency

- 135000 kg/hr - steam generated.

• as pressure $\uparrow$, BP of water $\uparrow$ and latent heat decreases
 at 221.2 bar; $T = 374.15^\circ\text{C}$; $h_{fg} = 0$
 at 221.2 bar, steam is directly formed from water.

Boiler mountings and Accessories:

→ Ex: Safety valve boiler

↳ These are extra devices that are fitted to the boiler drum. (It is compulsory, i.e. for safe and proper functioning). Without accessories can operate the boiler, i.e. it only increases the efficiency.

Ex: Superheater

Before steam moves to prime mover, we superheat, moisture is decreased, more work output, when sent into the turbine i.e. efficiency increases.

* Boiler mountings:

→ To measure pressure steam in

1. Pressure gauge: To supply steam from turbine base. we require minimum pressure to push the steam (once 100 bar is required/reached the turbine rotates).

2. Water level indicator:

To indicate the level of water.

3. Safety valve: A boiler can only withstand a certain max. pressure, after the pressure increases, the valve is opened automatically to let out excess steam and make whistling sound.

4. Fusible plug: When steam touches lead (has less m.p) and falls down. (It's function is to avoid overheating of boiler drum).

5. Feed check valve: To supply feed water to the drum.

6. Blow off cock: To remove sediments from the boiler drum. (The sediments from water).

7. Man hole: It is used in inspection and repair of the boiler.

8. Steam stop valve: Steam to turbine

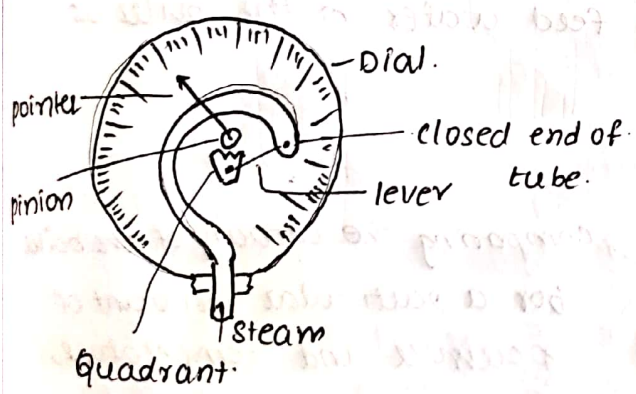
Steam from boiler drum to turbine.

(Without mountings, we cannot operate the boiler)

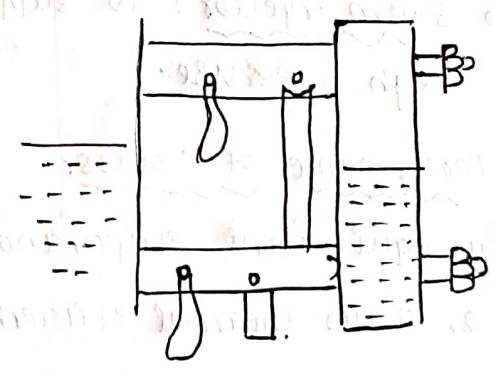
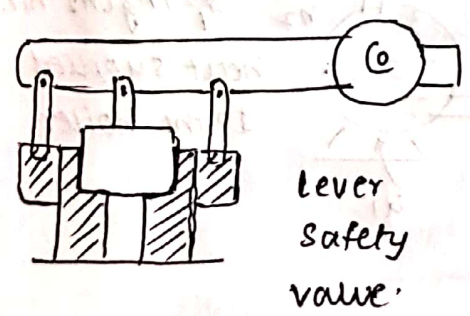
* Boiler Accessories:

1. feed pump:
2. Economiser:
3. Super heater
4. Air Preheater
5. steam trap.
6. steam separator.
7. steam injector.

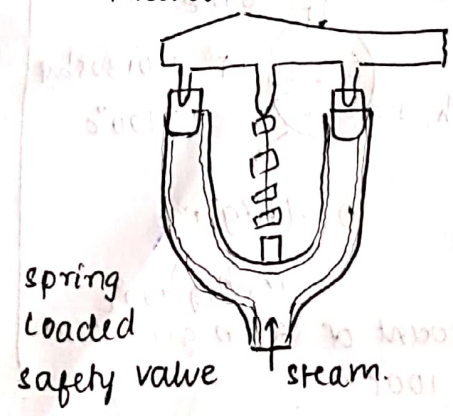
* Boiler Mountings:

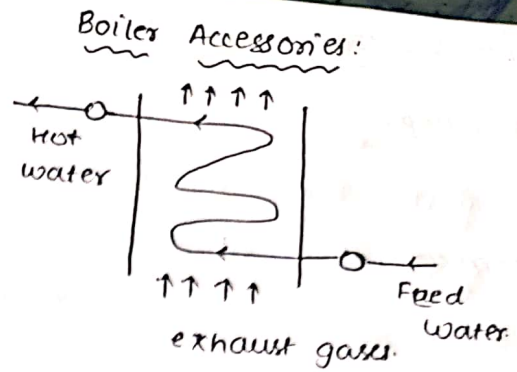
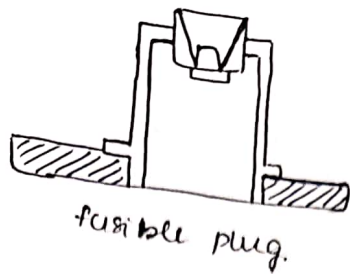


Bourdon Pressure gauge



Water level Indicator



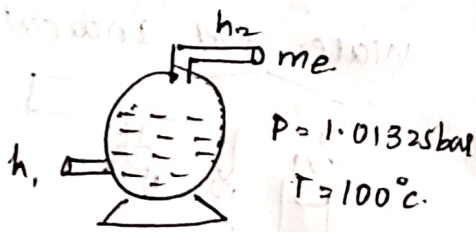


Economiser:

- Feed pump: To supply feed water to the boiler at high pressure. (needs electrical power)
- Economiser: To preheat the feed water before it goes to the boiler by using the heat from exhaust gases.
- Superheater: to superheat the steam before it moves to the prime mover by using Exhaust gases.
- Air preheater: to preheat the air before it moves to the furnace by using the heat from Exhaust gases.
- Steam trap: to separate the condensed steam from pipelines
- steam separator: to separate the moisture content from the steam before it moves to the prime mover
- steam injector: to supply feed water to the boiler at high pressure.

* Performance of Boilers:

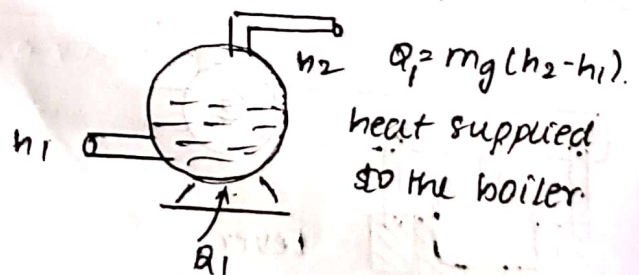
- 1) equivalent evaporation → comparing the capacity of the boiler for a particular amount of pressure and temperature, the amount of steam can be generated is called equivalent evaporation.
- 2) Boiler thermal efficiency



$$Q_2 = (h_{fg}) m_e$$

$$= m_e (h_{fg})_{100^\circ\text{C}}$$

amount of steam generated at 100°C .



heat supplied to the boiler

For equivalent evaporation.

$$Q_1 = Q_2$$

$$m_e (h_{fg}) = m_g (h_2 - h_1)$$

$$m_e = \frac{m_g (h_2 - h_1)}{h_{fg}}$$

$$m_e = \frac{m_g (h_2 - h_1)}{2256.9}$$

m_g - mass of steam at actual conditions.

$$P = 1.01325 \text{ bar}$$

$$T = 100^\circ\text{C}$$

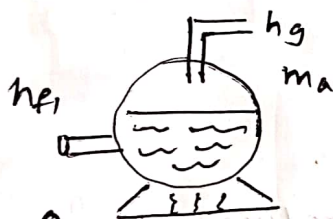
From steam table.

$$2256.9 \text{ kJ/kg}$$

$$2256.9 \text{ kJ/kg}$$

boilers * (The amount of steam

* equivalent evaporation is defined as the mass of water that could be evaporated from and at 100°C at atmospheric pressure (1.01325 bar) to produce dry saturated steam by the same amount of heat which produces the steam under actual conditions.



std. conditions:

$$P = 1.01325 \text{ bar.}$$

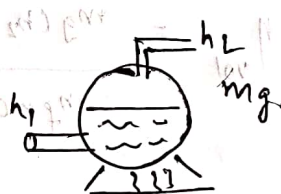
$$T = 100^\circ\text{C.}$$

$$Q_1 = m_e h_{fg}$$

$$Q_1 = Q_2$$

$$m_e h_{fg} = m_g (h_2 - h_1)$$

$$m_e = \frac{m_g (h_2 - h_1)}{h_{fg}}$$



Actual conditions:

$$Q_2 = m_g (h_2 - h_1)$$

$$m_e = \frac{m_g (h_2 - h_1)}{h_{fg}}$$

$$\therefore h_{fg} \text{ at } 100^\circ\text{C is } 2256.9 \text{ kJ/kg}$$

m_g = mass of steam gas kg/hr

h_1 = enthalpy of water at inlet kJ/kg

h_2 = enthalpy of steam at inlet kJ/kg

m_c = kg/hr

$$\frac{h_2 - h_1}{h_{fg}} \text{ , Factor of evaporation}$$

$$m_c = m_g \times \text{Factor of evaporation}$$

* Boiler Thermal Efficiency:

$$\eta_{\text{boiler}} = \frac{\text{Heat utilised to produce steam}}{\text{Heat generated by burn the fuel}}$$

$$\eta_{\text{vol}} = \frac{m_g (h_2 - h_1)}{m_f \times cv}$$

m_g = mass of steam produced (kg/hr)

m_f = mass of fuel burned (kg/hr)

cv = calorific value of fuel (kJ/kg)

h_1, h_2 = enthalpy (kJ/kg)

* The capacity to evaporate 15.563 kg of water per hour from ~~100~~ and at 100°C is called one boiler hp.

$$\text{Boiler HP} = \frac{m_g (h_2 - h_1)}{2256.9 \times 15.563}$$

steam	h'	v'	s'
wet	$h_{wet} = h_f + x h_{fg}$	$v_{wet} = x v_g$	$s_{wet} = s_f + x s_{fg}$
dry	$h_{dry} = h_g$	$v_{dry} = v_g$	$s_{dry} = s_g$
superheated	$h_{sup} = h_g + c_p (T_{sup} - T_g)$	$v_{sup} = \frac{T_{sup}}{T_g} v_g$	$s_{sup} = s_g + c_p \log \left(\frac{T_{sup}}{T_g} \right)$

* A boiler produces steam at a pressure of 10 bar and dryness fraction 0.95 while evaporating 8 kg of water per kg of coal burnt. The cv of fuel is 28800 KJ/kg. Determine the equivalent evaporation and thermal efficiency of the boiler. Consider feed water enters the boiler at 38°C.

at $P = 10 \text{ bar}$, $h_f = 762.9 \text{ KJ/kg}$

$h_{fg} = 2013.6 \text{ KJ/kg}$

at 38°C, $h_f = 159.1 \text{ KJ/kg}$

$\Rightarrow h_{wet} = 762.9 + (0.95)(2013.6)$

$h_{h_2} = 2675.82 \text{ KJ/kg}$

$$\eta = \frac{HP = 8 \times (2675.82 - 159.1)}{2256.9 \times 15.563}$$

$$m_e = \frac{8 (2675.82 - 159.1)}{2256.9}$$

$m_e = 8.92 \text{ kg/hr}$

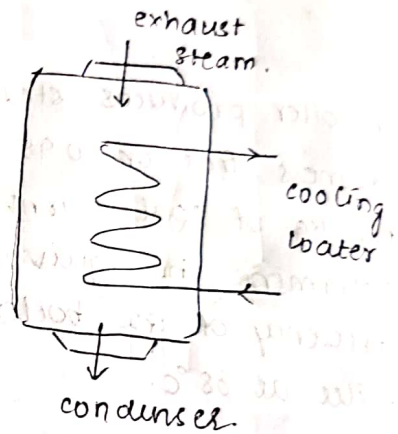
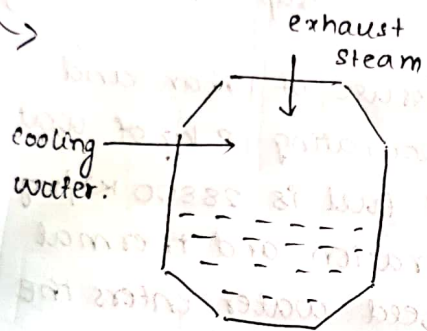
$$\eta_b = \frac{m_s(h_2 - h_1)}{m_p \times c_v}$$

$$= \frac{8 \times (2673.82 - 159.1)}{1 \times 28800}$$

$$= 69.1\%$$

* condensers:

- (1) Jet condensers
- (2) Surface condensers.

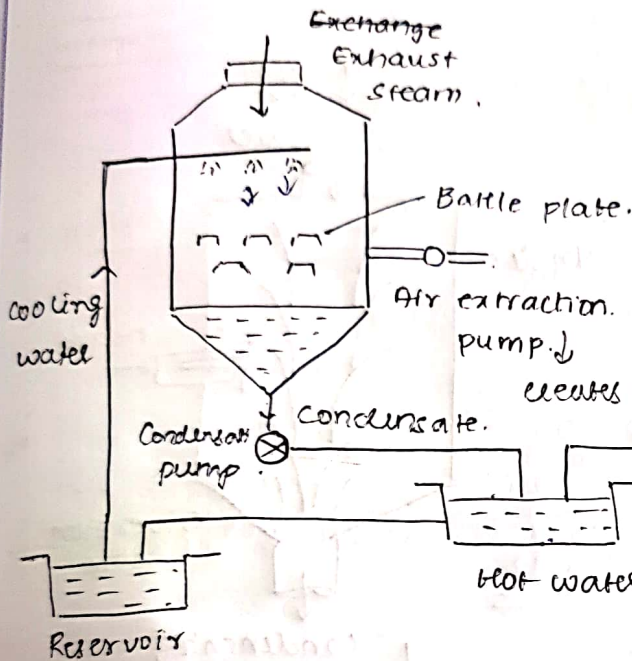


- | | |
|---|--------------------------------------|
| (1) suitable for small plants. | (1) For large plants. |
| (2) occupying less space. | (2) more space. |
| (3) simple design. | (3) complicated design. |
| (4) water and steam are in direct contact. | (4) not in direct contact. |
| (5) less quantity of cooling water is quantity. | (5) large. |
| (6) Low vacuum is maintained. | (6) more. |
| (7) condensate cannot be reused as feed water. | (7) can be reused as hot feed water. |

* types of jet condensers :

1. lowlevel Jet condenser. Parallel flow.
2. Highlevel Jet condenser (only counter flow). counter flow.
3. Ejector type

lowlevel parallel flow (Jet condenser)

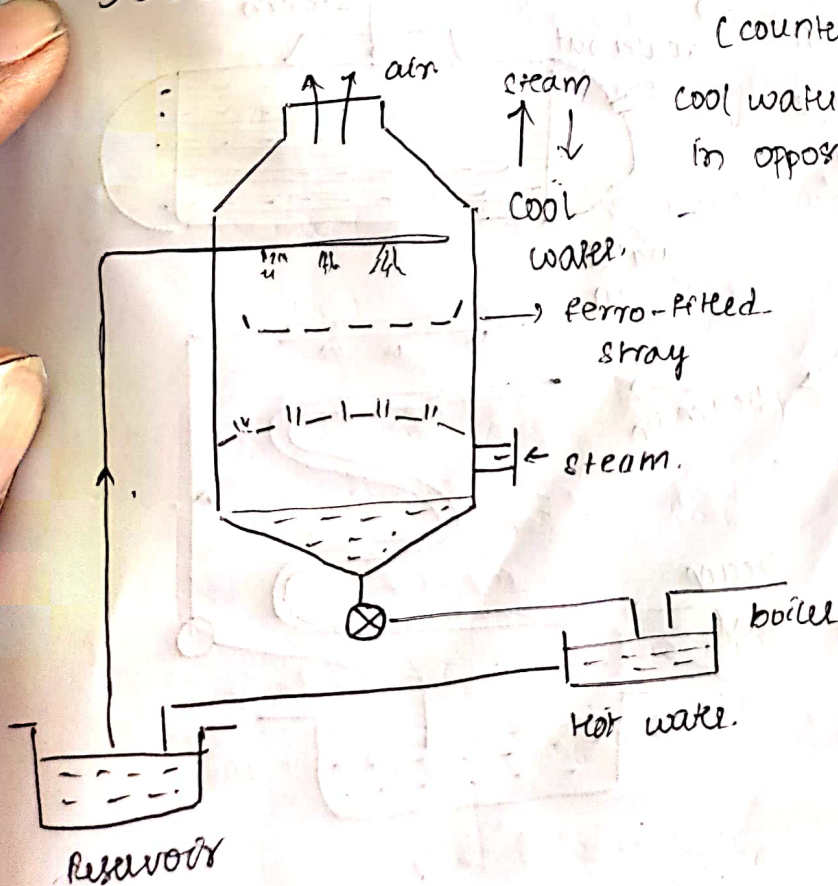


Baffle plate increases path steam and water droplets condense at the baffle plates moves to the bottom of tank.

cool water & steam moves in same direction. (parallel flow)

as this creates, high pressure is achieved and water moves upwards.

low level counter flow (Jet condenser)

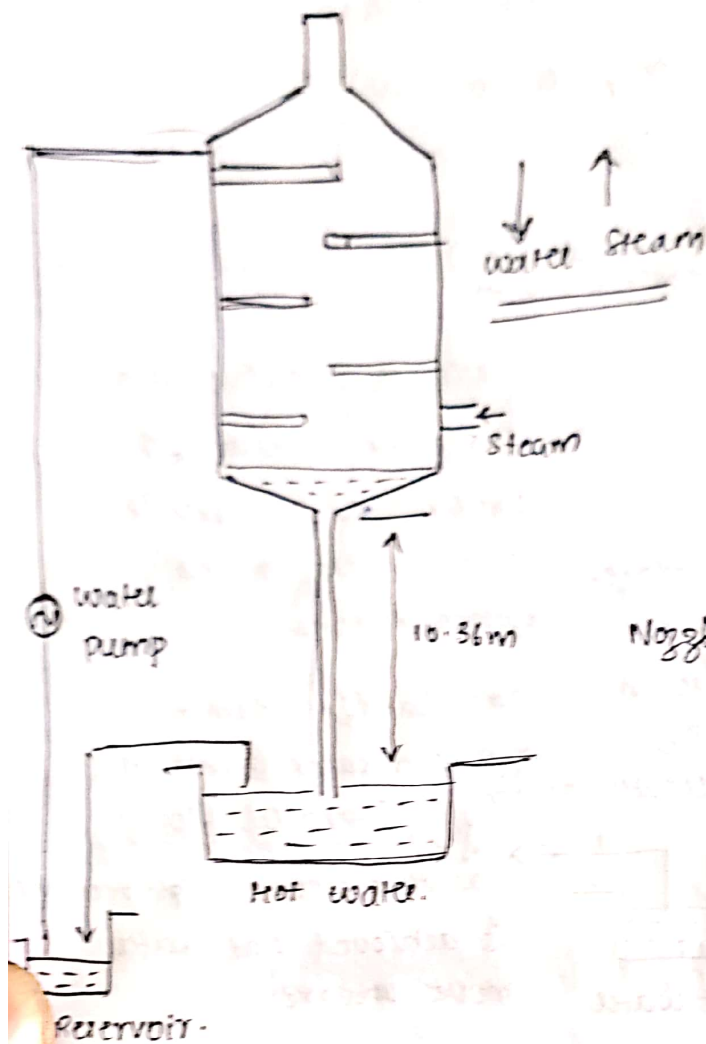


(counter flow)

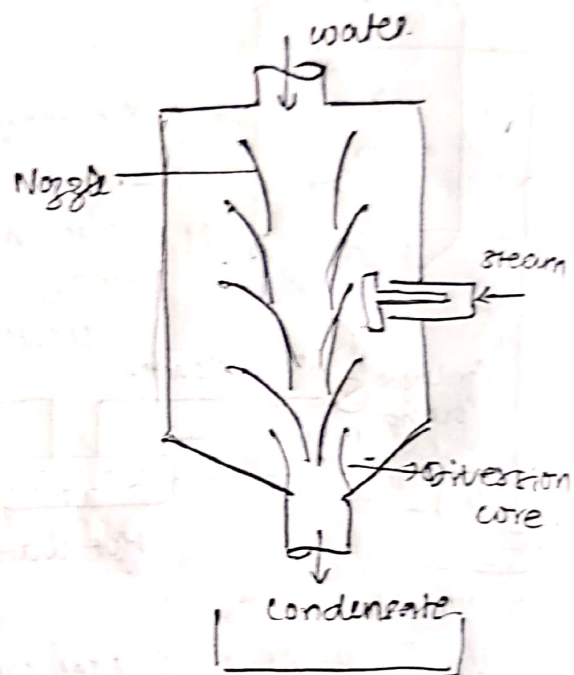
cool water and steam moves in opposite direction

lowlevel is one in which we don't require

* High Level jet condenser:

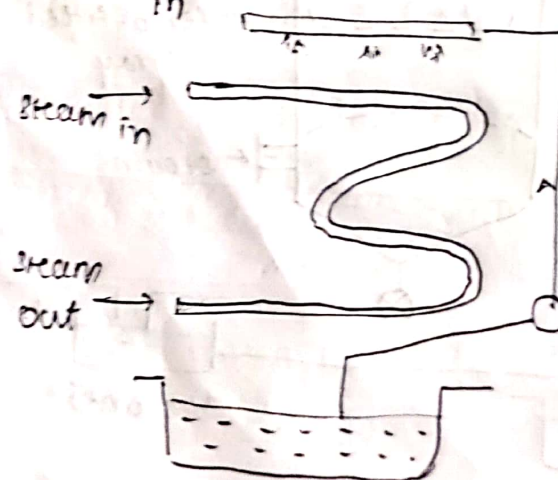
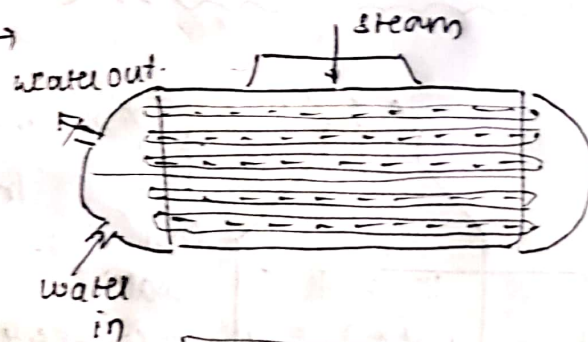


* Ejector type:

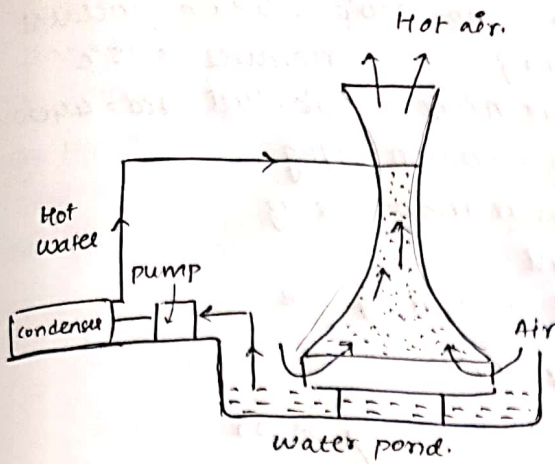


* Super condensers:

1. shell & tube type
2. Evaporative type.



* cooling tower:



Condenser - converts
cool to hot water.
and set it to the
tower.

* the following reading were obtained during a boiler trial for
a 6 hours duration

data, mean steam pressure = 12 bar

mass of steam generated = 40,000 kg

mean dryness fraction = 0.85

mean feed water temp = 30°C

Coal used = 4000 kg

calorific value of coal = 33400 kJ/kg

calculate (a) factor of equivalent evaporation
b, equivalent evaporation from and at 100°C
c, efficiency of the boiler.

$$(a) \frac{h_2 - h_1}{2256.9} \quad (h_{wet} = 798.4 + 0.85(1984.3))$$

$$h_f = 125.7$$

$$= \frac{2485.055 - 125.7}{2256.9} = 1.045$$

$$(b) m_e = m_g \left(\frac{h_2 - h_1}{2256.9} \right) = 40,000 \times \left(\frac{2485.055 - 125.7}{2256.9} \right)$$

$$= 41600 \text{ kg}$$

$$(c) \eta_{vol} = \frac{m_g (h_2 - h_1)}{m_f \times CV} = \frac{41600 (2485.055 - 125.7)}{4000 \times 33400} = 70.6\%$$

* the following observations are made on a boiler plant during 1 hour, pressure is 20 bar, steam temp. 260°C , steam generated 37500 kg, temp. of water entering the economiser is 15°C temp. of water leaving the economiser is 90°C , fuel used = 4400 kg, energy of combustion of fuel = 30,000 kJ/kg

- calculate equivalent evaporation per kg
- thermal efficiency of plant
- % heat energy of the fuel utilized by the economiser.

20 bar $T_{\text{sat}} = 212.4^{\circ}\text{C}$.

$T = 260^{\circ}\text{C}$

$\Rightarrow h_2 = 2797.2 + C_p(260 - 212.4)$

$= 2797.2 + (4 \times 47.6)$

$\Rightarrow \frac{3025.0 - 2902.4}{50}$

h_f for 260°C is 2926.92

h_f for $15^{\circ}\text{C} = 62.9$

$\Rightarrow \eta = \frac{37500 (2926.92 - 62.9)}{30,000 \times 4400}$

$= 81.36\%$

$\Rightarrow m_g \frac{(2926.92 - 62.9)}{2256.9}$

$\Rightarrow \frac{47587.73}{4400}$

$\Rightarrow \frac{47587.78}{4400}$

$= 10.81 \text{ kg / kg of coal}$

5. % of heat utilised in economiser

Heat utilised in Boiler = $m_s (h_{f'} - h_{f1})$

% of Heat utilised = $\frac{m_s (h_{f'} - h_{f1})}{m_f \times CV} \times 100$

$h_{f1} = 62.9$ at 15°C

$h_{f'} = 316.9$ at 90°C

$= \frac{37500 (316.9 - 62.9)}{4400 \times 30,000} \times 100$

$= 8.92\%$

- * A steam generator delivers steam at 100 bar, 500°C. The feed water inlet temp. is 160°C, the steam generation is 100000 kg/hr and the steam generator efficiency is 88%. Determine
- fuel burning rate in kg/hr if cv = 21 MJ/kg
 - the % of total heat absorbed in economiser, evaporator and superheater. (Assume that only latent heat is absorbed in the evaporator and neglect any pressure drop.)

Pressure = 100 bar

Temp = 500°C.

$h_f = 675.5$

$h_1(\text{Temp}) = 160^\circ\text{C}.$

100 bar $\Rightarrow T_{\text{sat}} = 311.0^\circ\text{C}.$

$T_{\text{sat}} < T_{\text{emp}}.$

$\Rightarrow h_f = \underline{\underline{3374.6}}$

$m_g = 100000 \text{ kg/hr}.$

$\eta_{\text{vol}} = \frac{m_g (h_2 - h_1)}{m_f \times \text{cv}}$

$\Rightarrow m_f = \frac{1,00,000 (3374.6 - 675.5)}{0.88 \times 21,000}$

$m_f = 14605.51 \text{ kg/hr}.$

Heat added by evap, economiser and superheater.

evaporator = $(h_g - h_{f1})$

from ST

at $P = 100 \text{ bar}$

$h_{f1}' = 1408 \text{ kJ/hr}$

$h_g = 2727.7 \text{ kJ/kg}$

$= (h_g - h_{f1}') \underline{\underline{48\%}}$

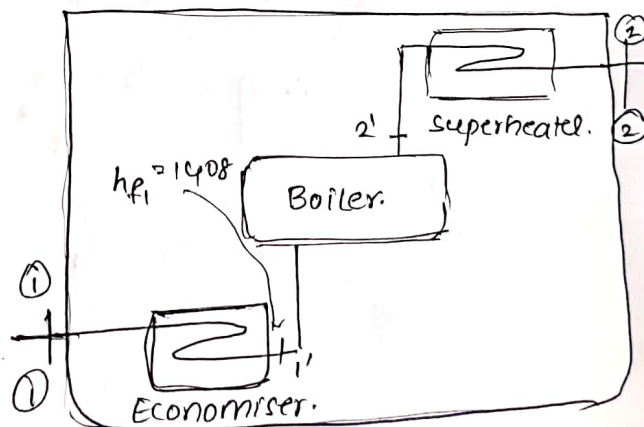
$= (2727.7 - 1408)$

$\Rightarrow 27.14\% \quad Q_{\text{ev}} = 1319.7 \text{ kJ}.$

$\Rightarrow \underline{\underline{\text{Economiser}}}: (h_{f1}' - h_{f1})$

$= (1408 - 675.5)$

$= 732.5 \text{ kJ}$



$\Rightarrow \underline{\underline{\text{Superheater}}}: (h_2 - h_g)$

$= (3374 - 2727.7)$

$\downarrow$
 $23\% = 646.9 \text{ kJ}$
 $=$

total heat added in steam generator

$$h = (h_g - h_{f1})$$

$$= (9374.7 - 695.5)$$

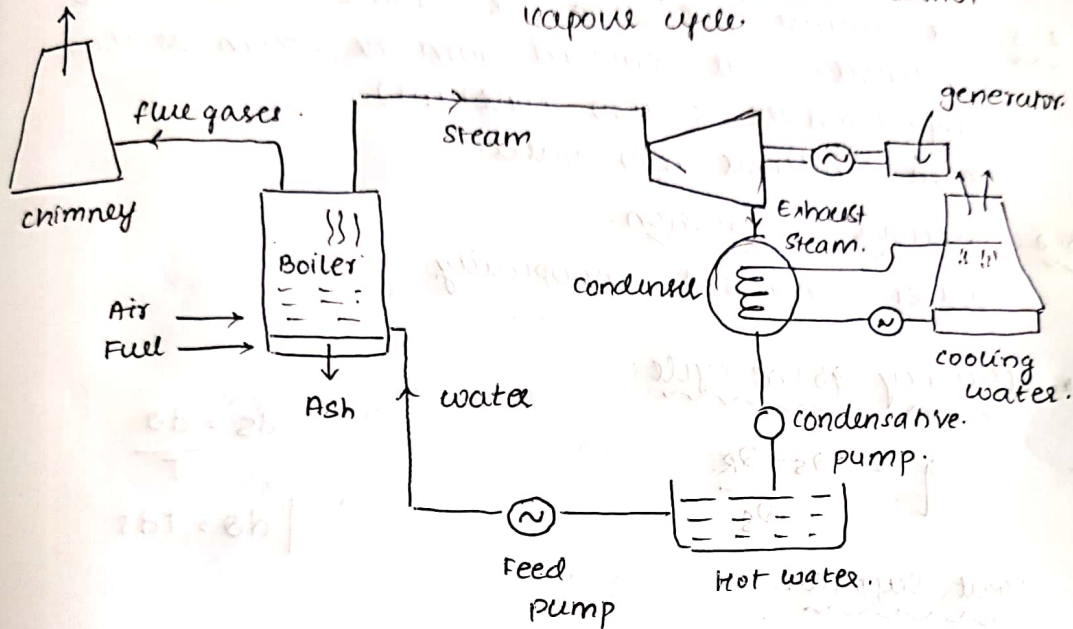
$$Q_{ht} = 2698.5 \text{ kJ}$$

UNIT-5:

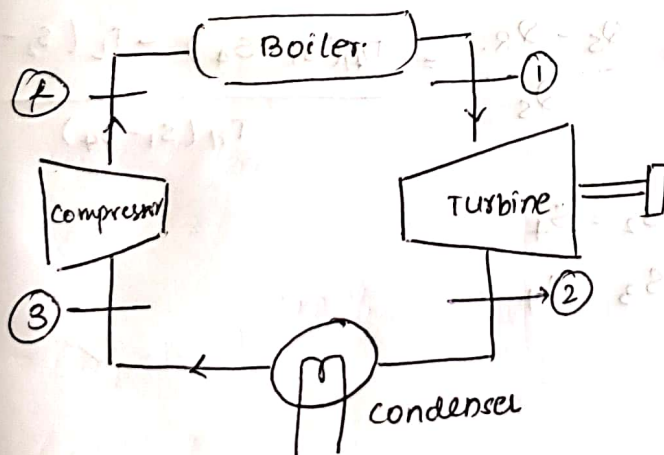
* steam Power Plant:

Rankine cycle is the working cycle in the steam power plant.

* Ideal cycle for working of steam powerplant is carnot vapour cycle.



* Carnot vapour cycle: (Ideal cycle for working of steam from plant is carnot vapour cycle).

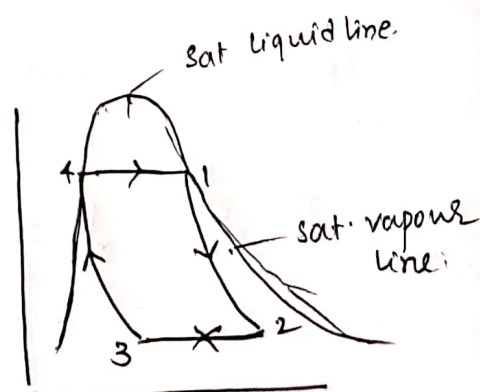
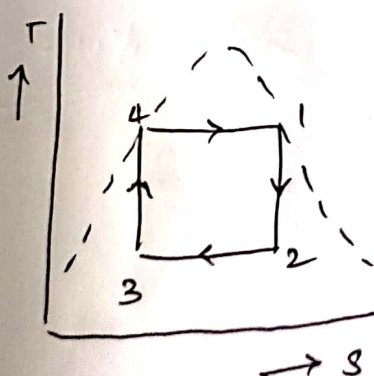


4-1: Isothermal Heat addition.

1-2: Isentropic expansion.

2-3: Isothermal Heat rejection.

3-4: Isentropic compression.



4-1: Isothermal Heat addition.

Heat is added to water at const. temp (pressure const.) (because when ball changes) in the boiler.

1-2: Saturated steam from the boiler enters into the turbine. steam expands isentropically.

2-3: The exhaust ^{steam} heat from the turbine enters into the condenser. Heat removed from the steam at const temp. (Pressure is also constant). steam condense into water.

3-4: Isentropic compression:

water compressed isentropically.

* Efficiency of the cycle:

$$\eta = \frac{Q_S - Q_R}{Q_S}$$

Heat supplied:

$$Q_S = T_H (s_1 - s_4)$$

Heat rejected: $Q_R = T_L (s_2 - s_3)$

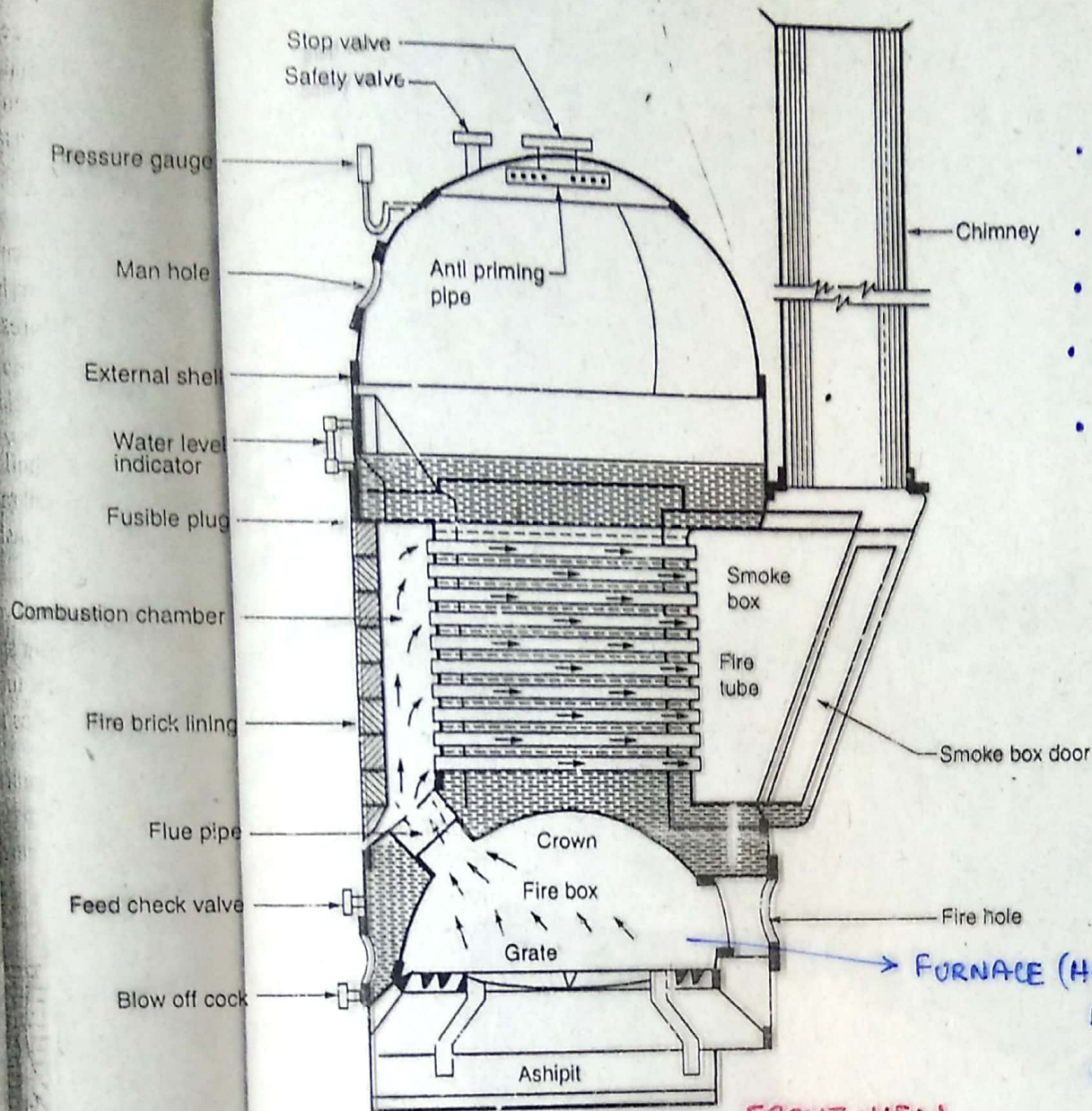
$$\therefore \eta = \frac{Q_S - Q_R}{Q_S} = \frac{T_H (s_1 - s_4) - T_L (s_2 - s_3)}{T_H (s_1 - s_4)}$$

but $s_2 = s_4$

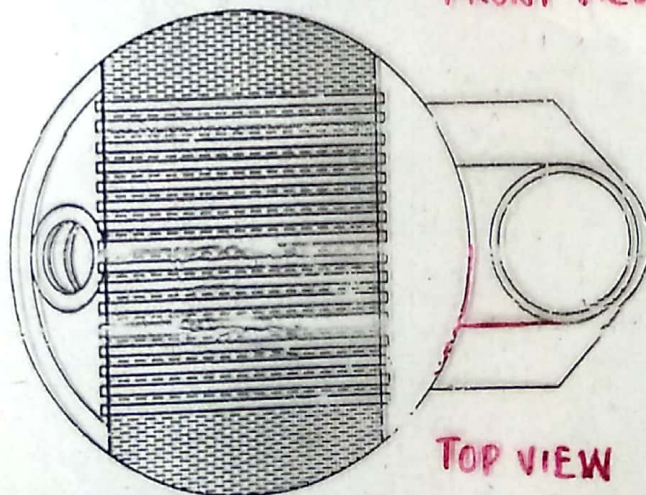
$s_3 = s_4$

$$ds = \frac{dq}{T}$$

$$dq = T ds$$



FRONT VIEW



TOP VIEW

COCHRAN BOILER

• FIRE-TUBE BOILER

• VERTICAL

• MULTI-TUBULAR

• INTERNALLY FIRED

• STATIONARY

→ FURNACE (Hemispherical)

↳ Strongest

↳ maximum absorption of radiant heat

Adv:

• Can use any kind of fuel

* wood, paddy husk, oil fuel, coal

• Steam production

rates: 20 kg/hr to

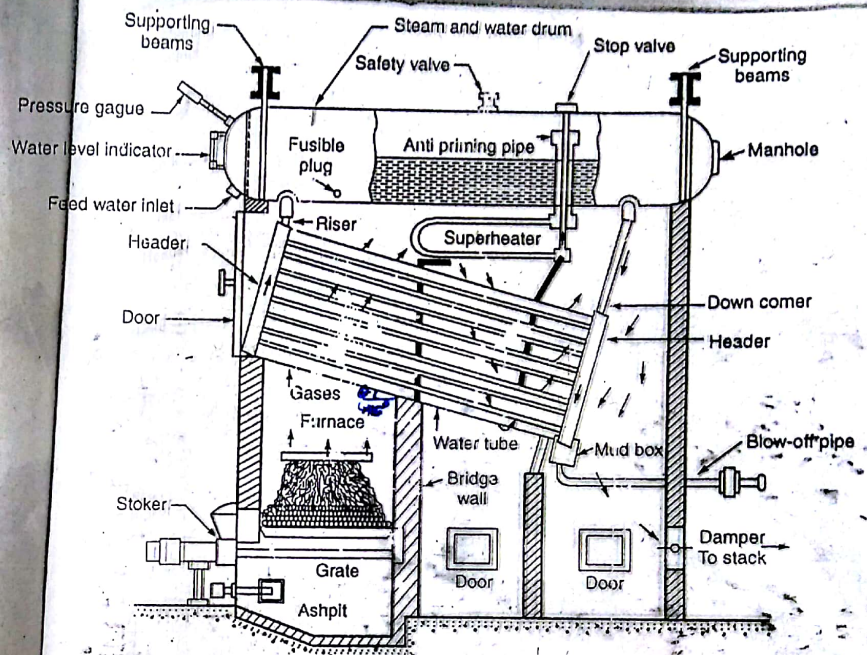
3000 kg/hr

• Steam Pressures: 20 bar (max)

• $\eta = 70-75\%$

• Cochran

- Water Tube Boiler
- HORIZONTAL
- Natural Circulation Boiler



BABCOCK & WILCOX BOILER

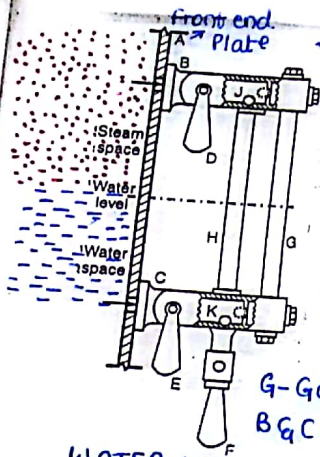
Steam Capacity: 40,000 kg/hr (max)

Steam Pressure: 40 bar to 125 bar.

0° to 15°

≈ 100 bar

BOILER MOUNTINGS



- Fittings & devices which are necessary for the boiler safety and control

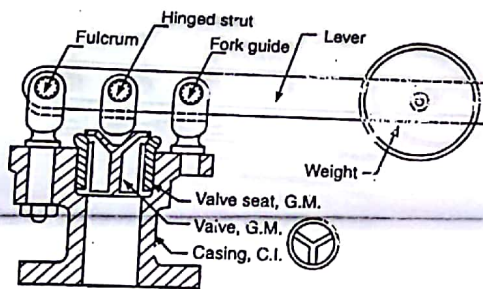
- Integral part of the boiler

- Mounted over the boiler shell

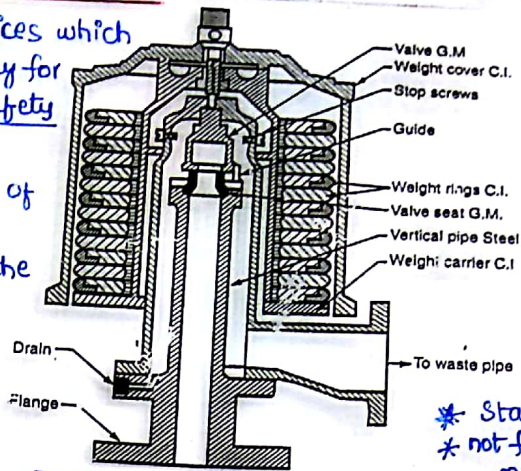
G - Gauge Glass tube
B & C: Stuffing Boxes

WATER LEVEL INDICATOR

- Shows the exact level of water at any instant

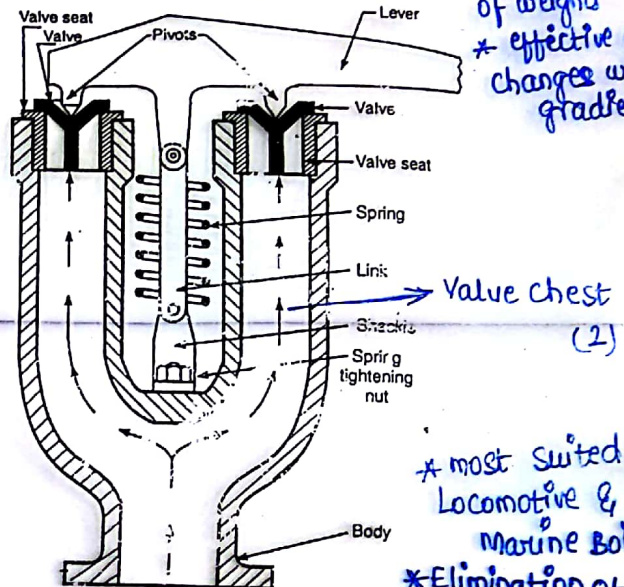


LEVER SAFETY VALVE



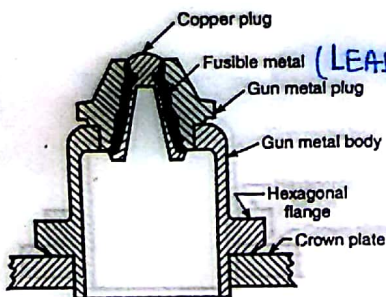
DEAD WEIGHT SAFETY VALVE

* Stationary Boilers only
* not for Locomotive & marine Boilers, as they need large amounts of weights
* effective wt changes with gradient



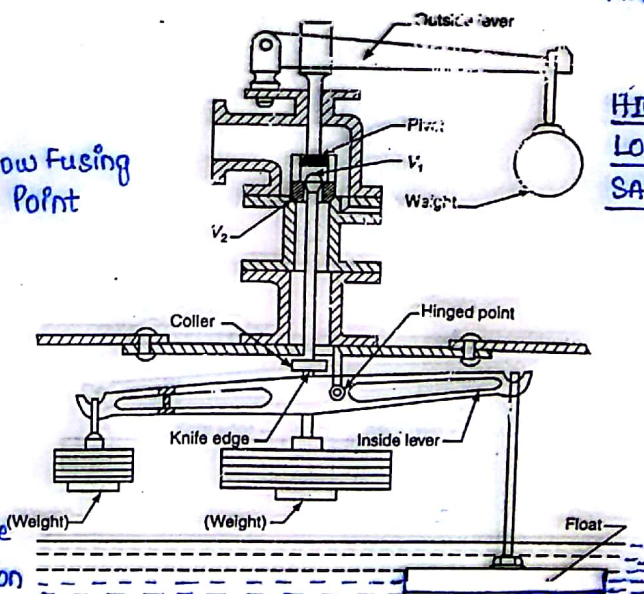
SPRING LOADED SAFETY VALVE

* most suited for Locomotive & marine Boilers
* Elimination of heavy wts.
most widespread use

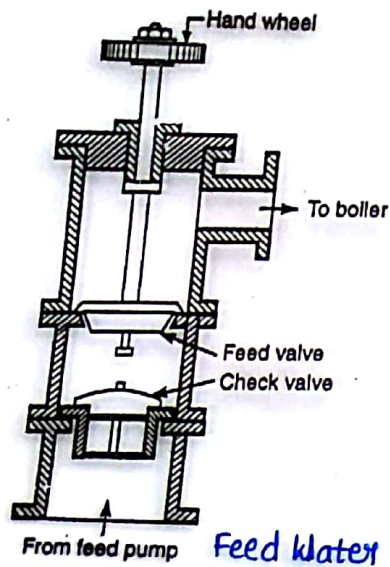


FUSIBLE PLUG

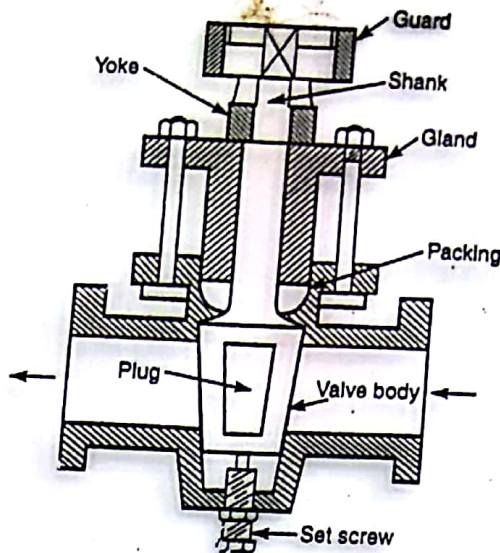
To extinguish fire in the furnace when the water level falls below the safe limit ∴ prevents boiler explosion



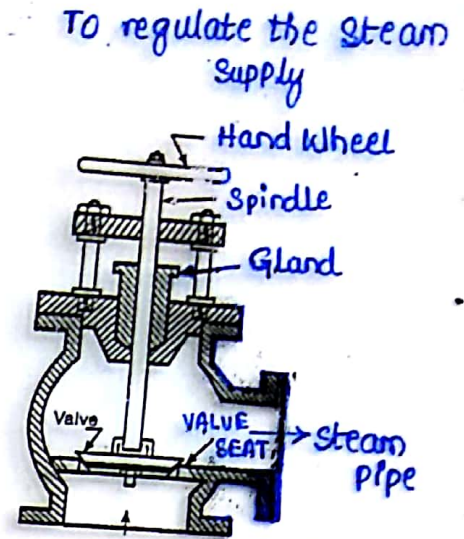
HIGH STEAM LOW WATER SAFETY VALVE



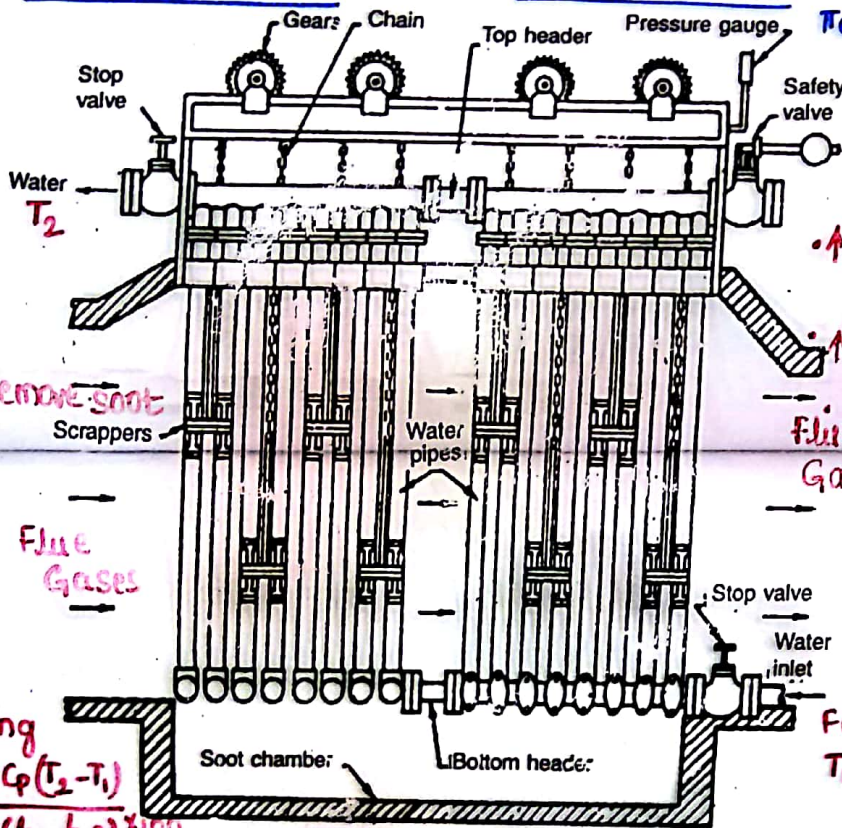
FEED CHECK VALVE



BLOW-OFF COCK

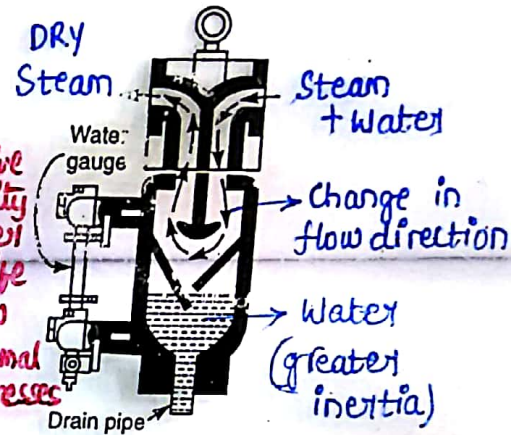


STOP VALVE / JUNCTION VALVE



WATER ECONOMISER

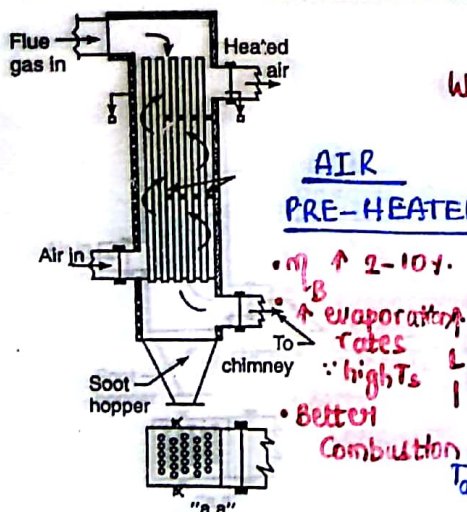
To clean the sediments periodically (mud, scale, impurities)



STEAM DRIER (OR) SEPARATOR

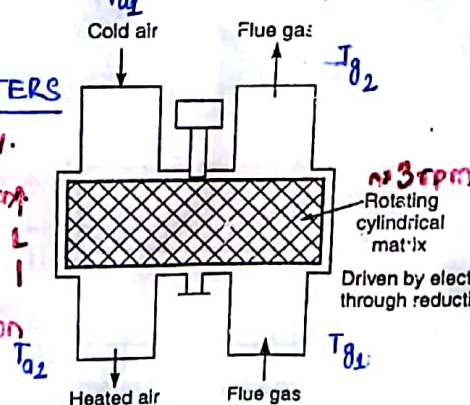
BOILER ACCESSORIES

- fittings or devices provided to increase the efficiency of boiler & for smooth working

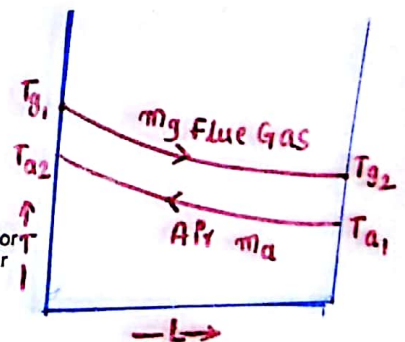


RECUPERATIVE AIR-PRE-HEATER

Waste heat recovery device



REGENERATIVE AIR-PRE-HEATER (Ljungstrom Rotary Air Pre-Heater)



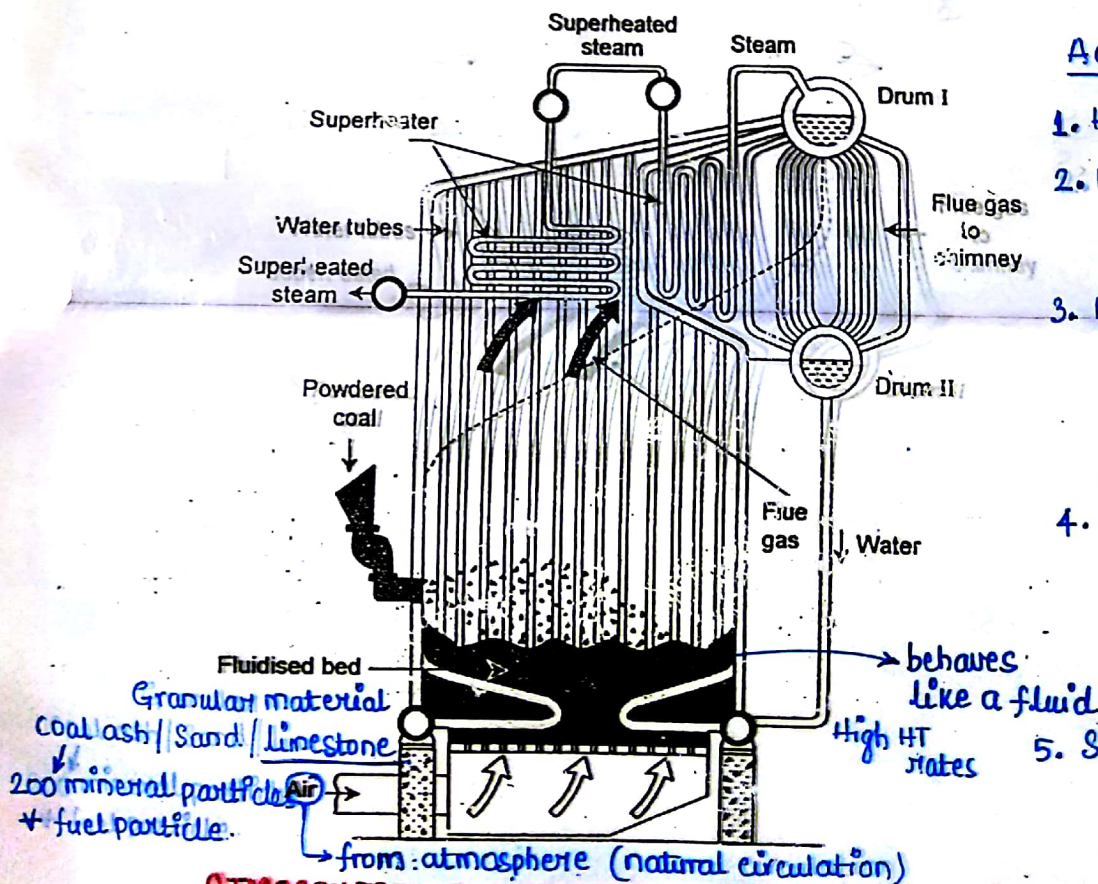
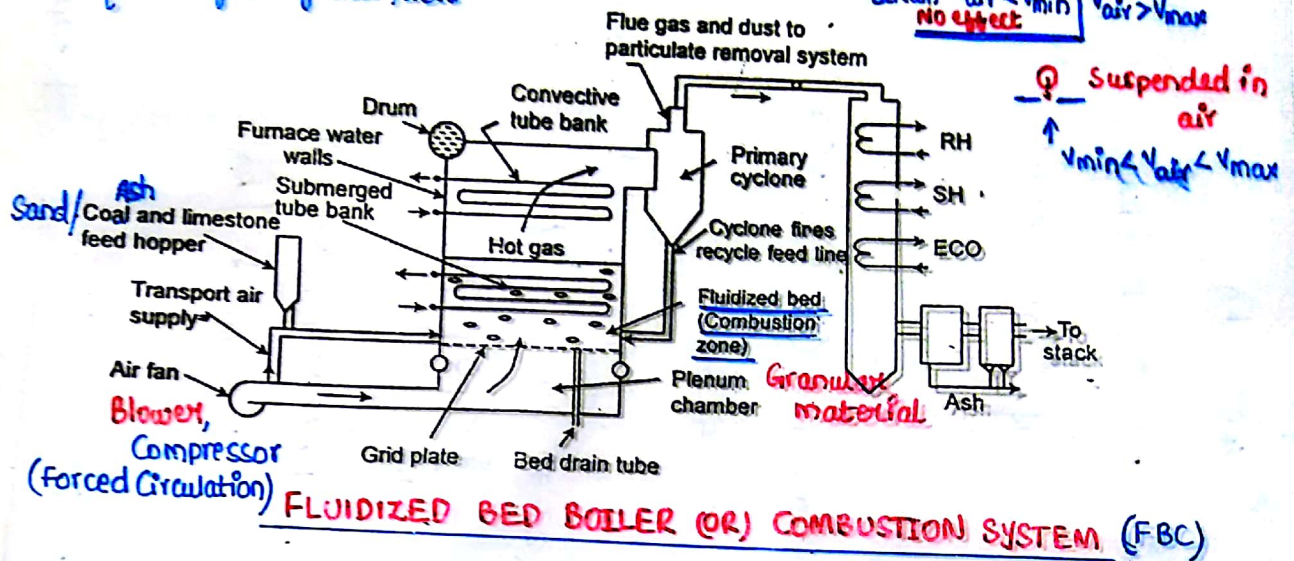
AIR PRE-HEATERS

- $\eta \uparrow 2-10\%$
- $\uparrow$ evaporator rates
- high T_s
- Better Combustion

- $\uparrow \eta_{th}$
- $1\% \rightarrow 6^\circ C$
- $\uparrow$ evaporative capacity
- Flue long boiler life
- low thermal stresses

$$\text{Saving } S = \frac{C_p(T_2 - T_1)}{(h - h_f) \times 100}$$

- * To minimise pollution resulting from Conventional plants
- * For use of low grade fuels



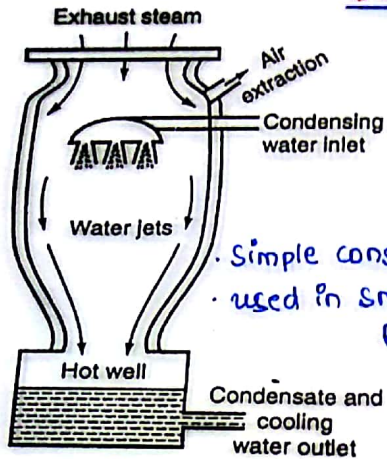
Advantages:

1. High Combustion η .
2. Flexibility in the use of fuels (Solid/Liquid/Gaseous)
3. Minimum pollution due to stabilised combustion @ lower Temperature
*No NO_x
4. Can easily or efficiently burn low grade fuels also (high ash)
behaves like a fluid
High HT rates
5. Simple & small in size. $\therefore$ ease in manufacturing.

Lime Stone — precipitates out Sulphur $\rightarrow$ comp. SO_2

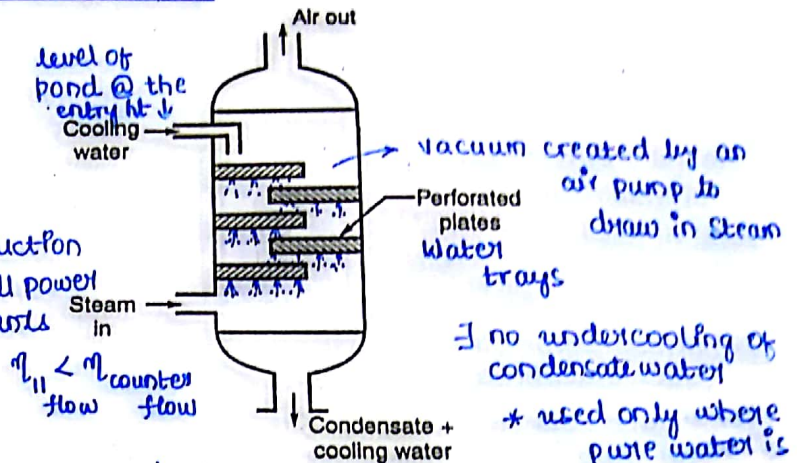
Low Temp. — Stabilised combn $\rightarrow$ $\text{NO}_x \downarrow$

JET CONDENSERS

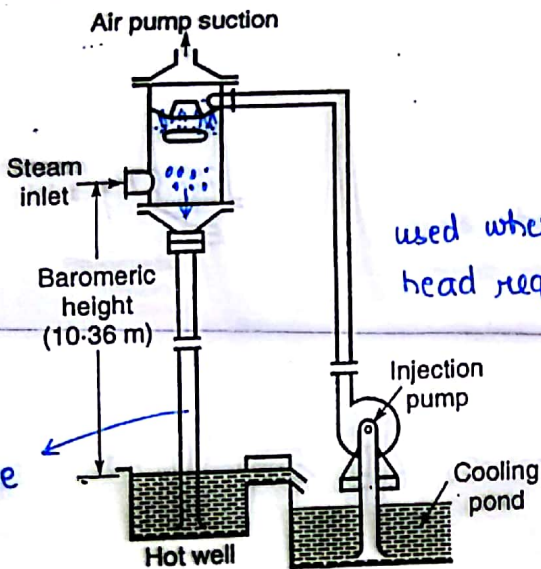


LOW LEVEL PARALLEL FLOW JET CONDENSER

Simple construction
used in small power plants

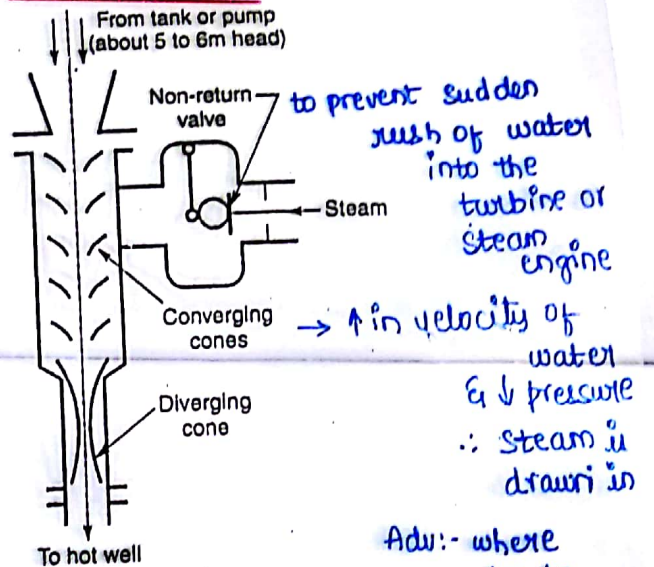


LOW LEVEL COUNTER FLOW JET CONDENSER



BAROMETRIC JET CONDENSER

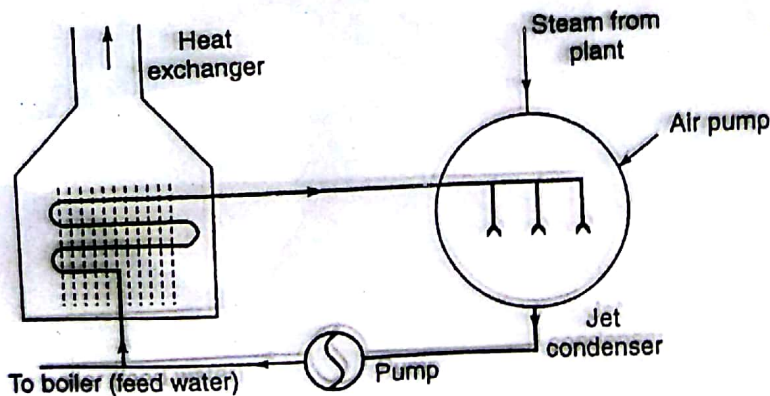
used where sufficient head required for tail pipe is available



EJECTOR CONDENSER

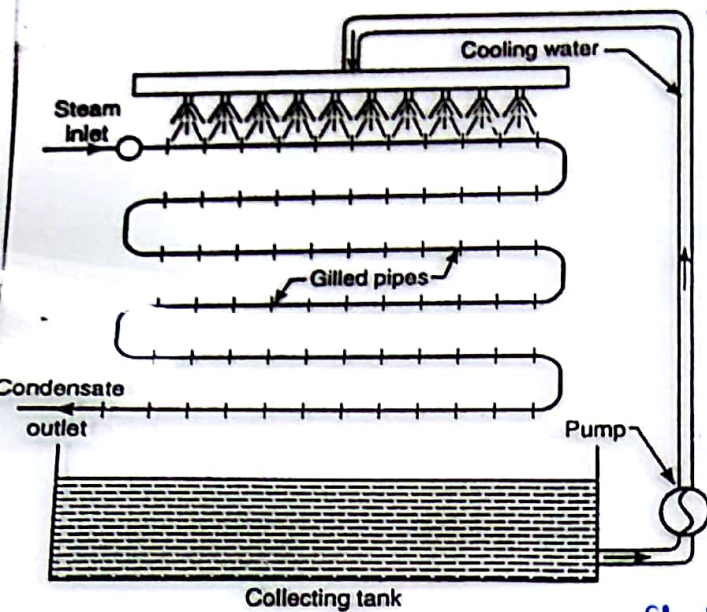
Adv:- where moderate vacuum is needed

* High Level Jet Condenser



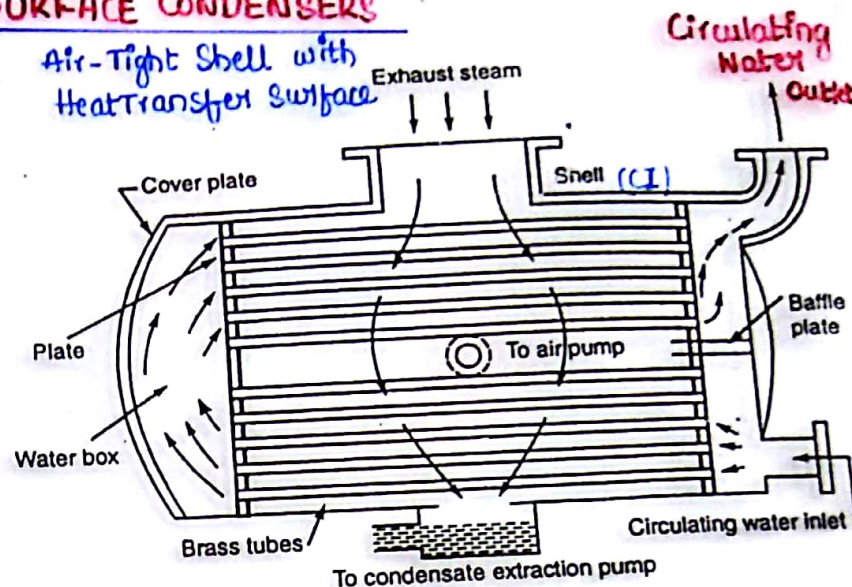
MODERN JET CONDENSER PLANT (LAY OUT)

SURFACE CONDENSERS



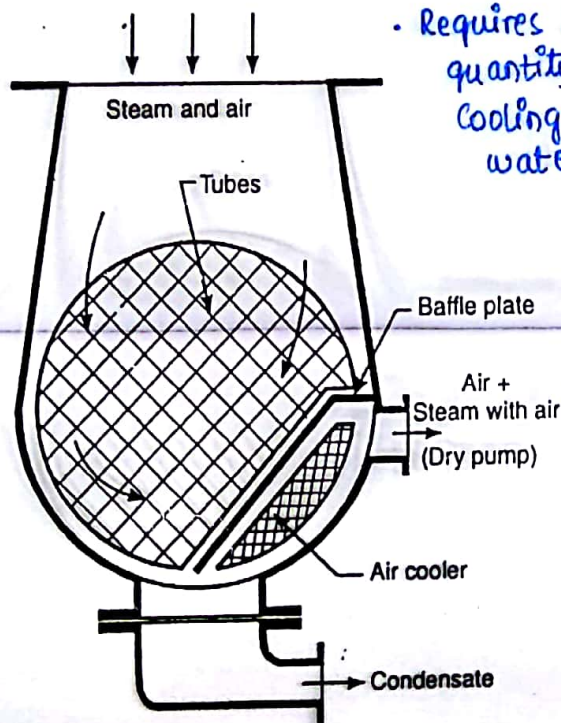
EVAPORATIVE CONDENSER

- Simple,
- Cheap
- Requires less quantity of cooling water

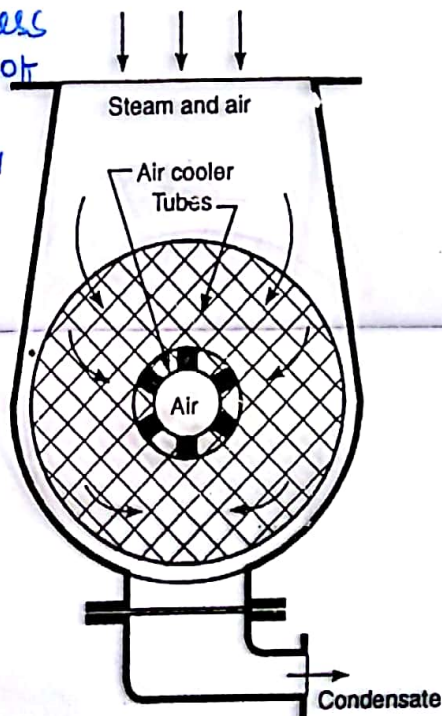


TWO-PASS SURFACE CONDENSER

Effective condensation



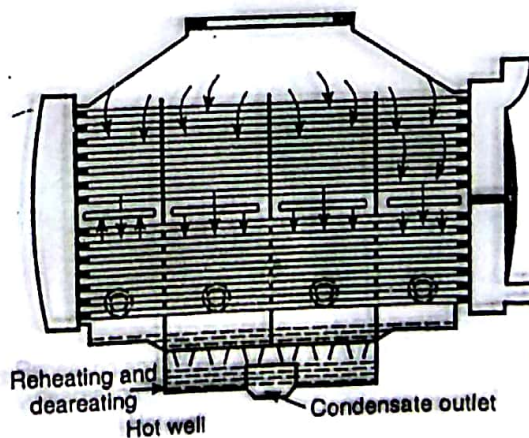
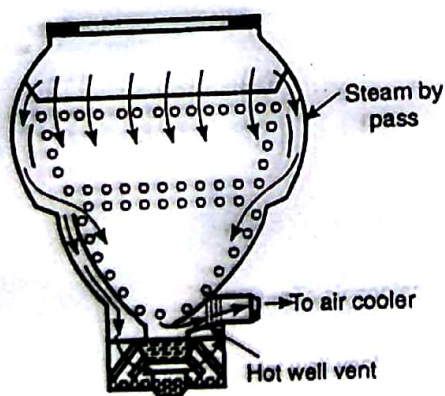
DOWNFLOW CONDENSER



CENTRAL FLOW CONDENSER

Tubes:

- Red Brass or
- Muntz Metal
- Admiralty Brass
- Stainless Steel



MODERN SURFACE CONDENSER

- (iv) Minimum erosion and corrosion,
- (v) the easy to generate peak load and
- (vi) Ease in operation

6.5. Fluidized Bed Boilers.

The efforts to minimize the pollution resulting in from conventional steam power plants and to have efficient use of low grade fuels led to the discovery of fluidized bed boiler in which steam is generated by utilising the technique of fluidized bed combustion. *The detailed analysis of fluidized bed combustion has been discussed in the chapter.*

Solid (coal) particles when subjected to a stream of air may be either (i) unaffected if the air velocity is below a certain value or (ii) air borne if the air velocity is above a certain value. Between the limits of velocity, a bed of small solid particles of fairly uniform size will not become fully air-borne but will have their reduced effective density and will move about in the bed as if bed is fluid. By proper regulation of air velocity the solid particles can be burnt in the combustion chamber almost as easily as if they are fluid. The *critical velocity* is called the *minimum fluidization velocity* at which the bed is said to be *incipiently fluidized*. With the further increase in velocity, the particles are buoyed up and imparted a violently turbulent fluid like motion. The mixing is rapid and equilibrium between gas and particles is rapidly established which is called a *fluidized bed*. There are two types of fluidized bed boilers. They are—

- (i) Bubbling fluidized bed (BFB) boilers
- (ii) Circulating fluidized bed (CFB) boilers

Both boilers may be operating at *atmospheric pressure or pressurized condition.*

6.5.1. Bubbling Fluidized Bed Boiler (BFB). Fig. 6.28 shows a schematic of a bubbling fluidized bed boiler working under atmospheric condition. *Crushed coal having the size 6 to 20 mm is injected into the fluidized bed along with limestone through the feed hopper just above an air-distribution grid at the bottom of the bed. In some cases, coal and limestone are subjected into the lower section of the furnace from bottom. The air flows upwards through the grid from the air plenum into the bed, which now becomes the furnace where combustion of the swirling mixture takes place. The products of combustion*

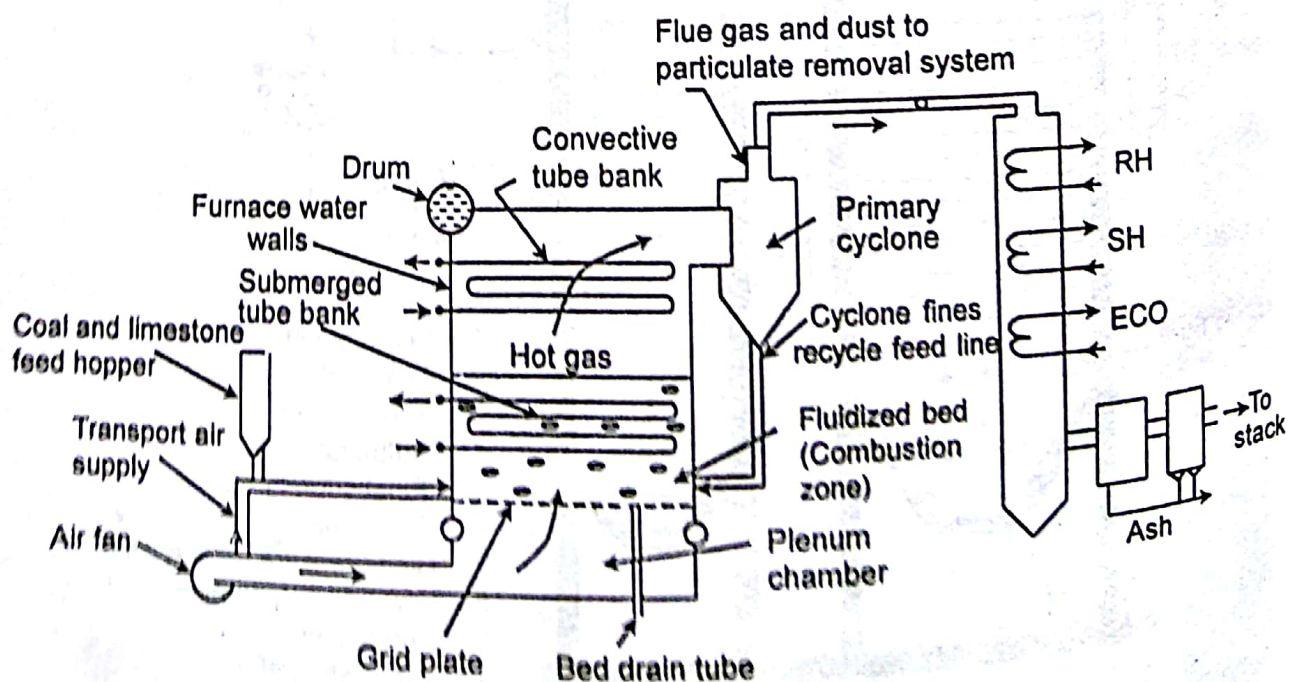


Fig. 6.28. Bubbling Fluidized Bed (BFB) Boiler.

tion leaving the bed contain a large proportion of unburned particles that are collected in a cyclone separator, which separates these particles from the gas by imparting centrifugal acceleration into the mixture. These separated unburned particles are returned back to the fluidized bed to complete their combustion. *The boiler water tubes are located in the furnace.* Finer solid residues (ash and spent sorbents) generated during combustion and desulphurization leave the furnace escaping through the cyclones, but they are collected by a bag-house or electrostatic precipitator. *The most important advantage of fluidized bed combustion lies in the concurrent removal of the sulphur dioxide that results normally from the combustion of sulphur content of the coal.* Desulphurization takes place by the addition of limestone directly to the bed together with the crushed coal. The combustion temperature is low (800- 900°C). Due to low combustion temperature, inferior grades of coal can be used *without slagging problem and there is no formation of NO_x .* The other advantage of low combustion temperature is the use of cheaper alloy material which results in economical construction. Since there is no pulverizer, so further economy is achieved. Further, a fluidized-bed combustor can be designed to incorporate the boiler within the bed, resulting in volumetric heat transfer rates that are 10 to 15 times higher and surface heat transfer rates that are 2 to 3 times higher than a conventional boiler. All these make the boiler more compact.

Economisers, superheater, reheaters, preheaters, bag-house or electrostatic precipitator, ID fan, stack, etc are used like conventional boiler.

6.5.2. Circulating Fluidized Bed (CFB) Boilers. Fig. 6.29. shows a second generation CFB boiler. The whole system is divided into two-sections namely solid fuel circulation loop and back-pass loop. The first section comprises of furnace or fast fluidized bed, cyclone (gas-solid separator), solid recycle device, (loop seal or L-valve) and external heat exchanger (optional). Similar to pulverized coal-fired (PC) boiler, the furnace enclosure is generally made of water tubes. In the a by-pass section, the remaining heat from the flue

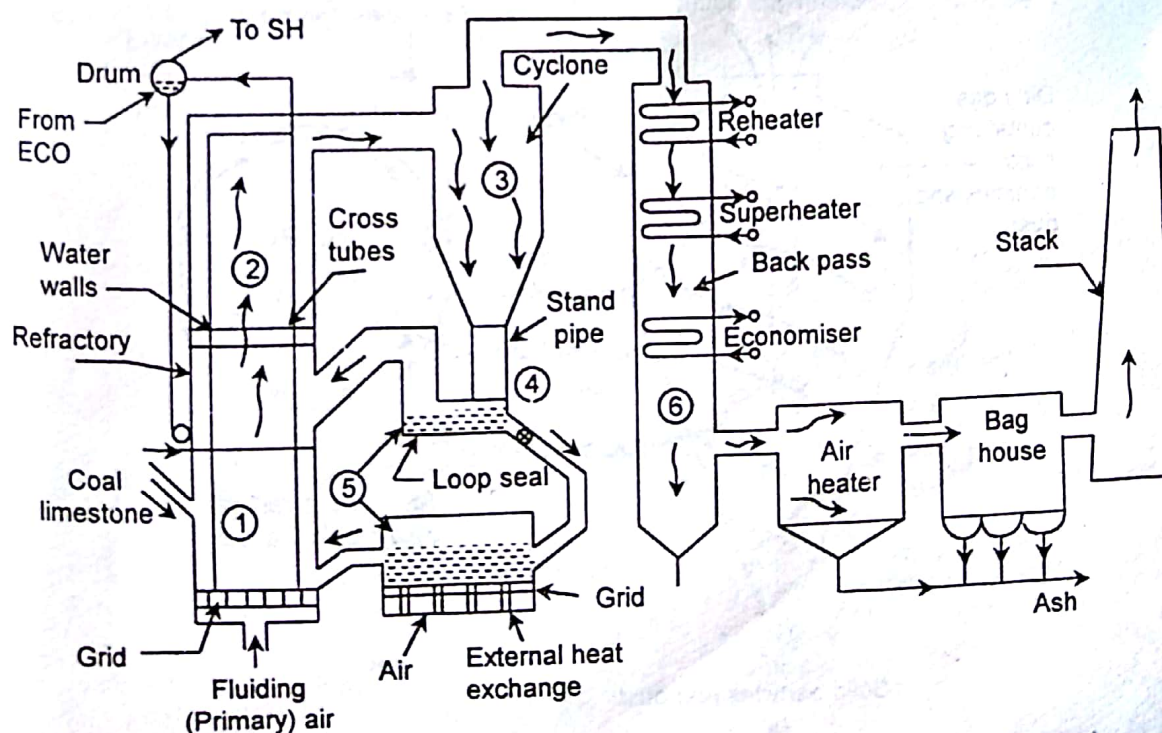
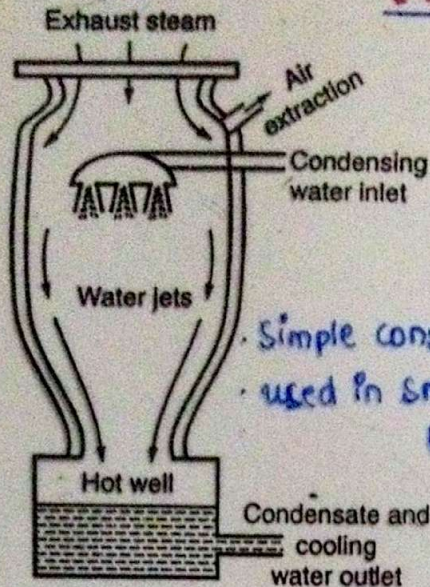


Fig. 6.29. Circulating Fluidized Bed (CFB) Boiler.

gases is absorbed by the superheaters, reheaters, economisers and air- preheater surfaces.

In the first section, the furnace walls are lined with refractory upto the level of secondary air entry and beyond this furnace walls are generally cooled by evaporative superheater or reheater surface. The cyclone dust collector and the non-mechanical valves are also lined with refractory. Coal/limestone are injected into the lower section of the furnace through hopper system. There happens to be a loop furnace in which the recycled unburned coal particles are burned. The fluidizing (primary) air is fed to the furnace through an air distributor or grate and the secondary air is admitted at some height above the grate to complete combustion. With the injection of primary and secondary air the bed is fluidized, combustion takes place resulting in lower temperature of the gas (800 to 900°C). The ash from the bed falls down. The cyclone separator, the coarse sorbent (limestone) and coal particles which are recycled back near the base of the furnace while the finer solid residues (ash and spent sorbents) after passing over superheaters, reheaters economisers, air preheaters get captured by bag house or electronic precipitator and go finally to atmosphere via stack. The flow regimes are (i) turbulent fluidized bed, (ii) fast fluidized bed, (iii) swirl flow, (iv) moving packed bed, (v) bubbling fluidized bed, (vi) pneumatic transport.

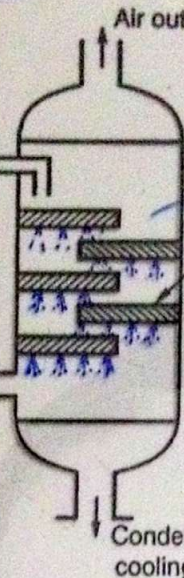
JET CONDENSERS



LOW LEVEL PARALLEL FLOW JET CONDENSER

Simple construction
used in small power plants

level of pond @ the entry ht ↓
Cooling water
Steam in
 $\rho_{H_2O} < \rho_{\text{condensate}}$
flow flow

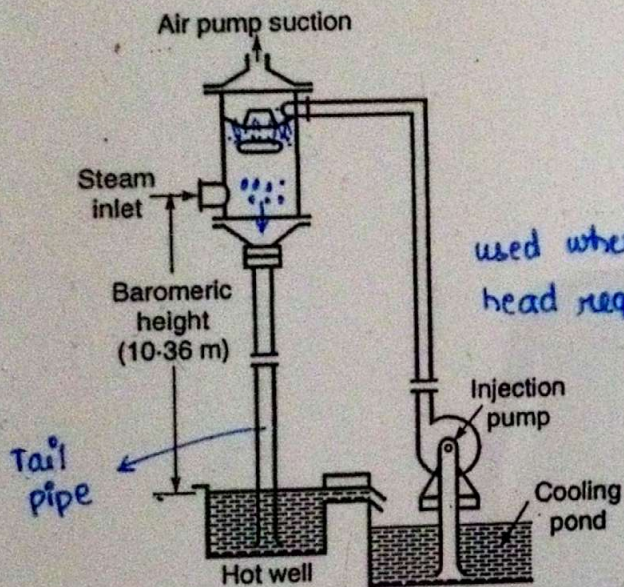


LOW LEVEL COUNTER FLOW JET CONDENSER

vacuum created by an air pump to draw in steam
Water trays

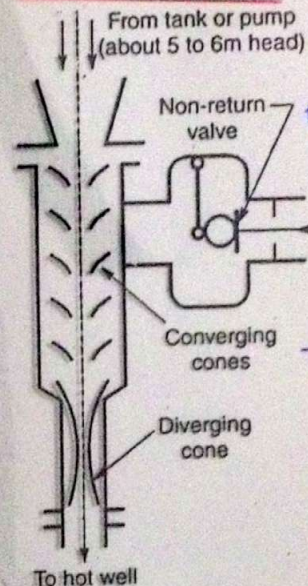
∴ no undercooling of condensate water

* used only where pure water is abundant



BAROMETRIC JET CONDENSER

used where sufficient head required for tail pipe is available

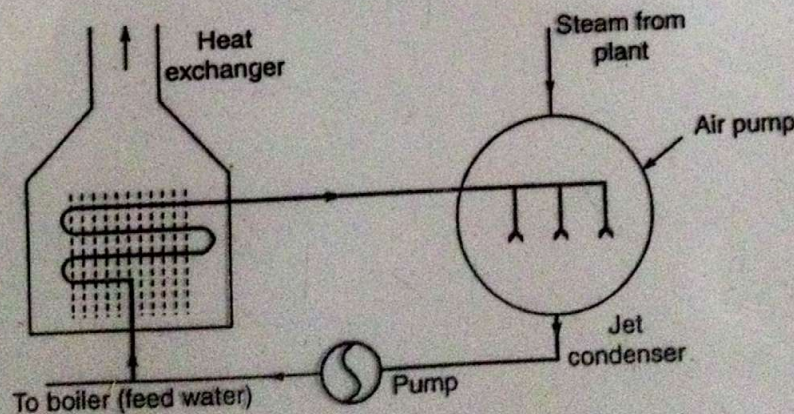


EJECTOR CONDENSER

to prevent sudden rush of water into the turbine or steam engine
→ ↑ in velocity of water & ↓ pressure
∴ steam is drawn in

Adv:- where moderate vacuum is needed

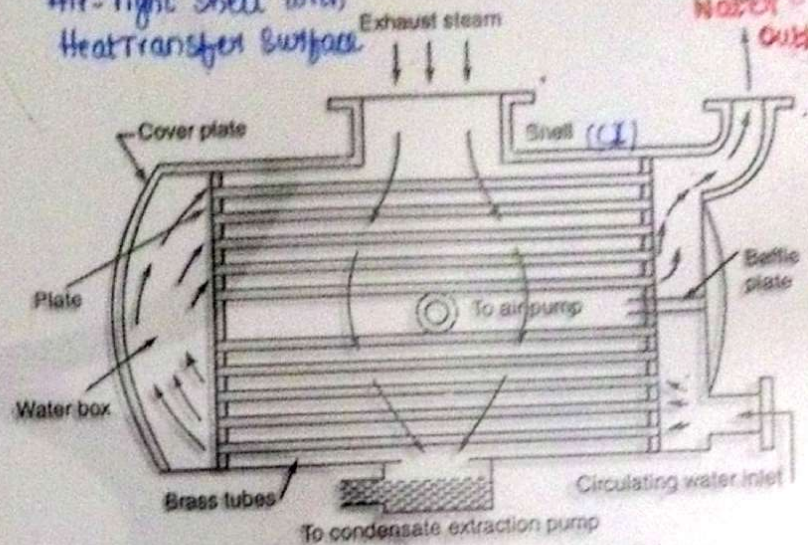
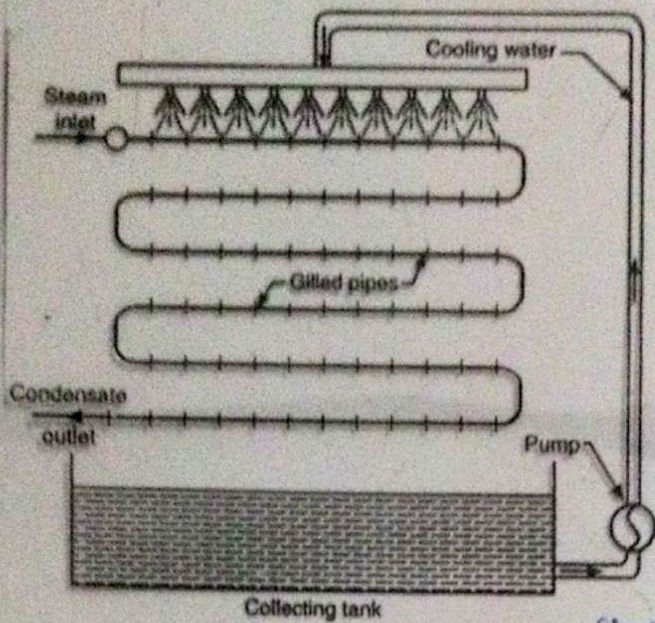
* High Level Jet Condenser



MODERN JET CONDENSER PLANT (LAY OUT)

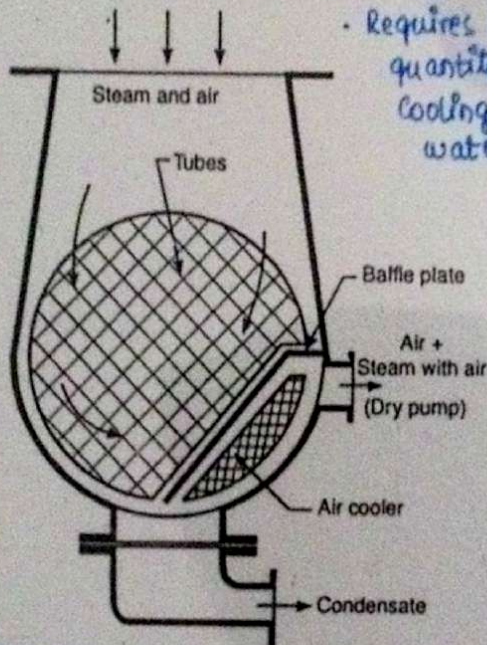
SURFACE CONDENSERS

Air-Tight Shell with Heattransfer Surface



EVAPORATIVE CONDENSER

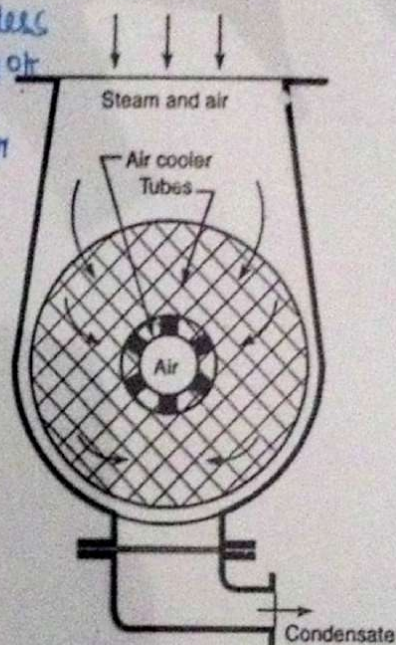
- Simple,
- Cheap
- Requires less quantity of cooling water



DOWNFLOW CONDENSER

TWO-PASS SURFACE CONDENSER

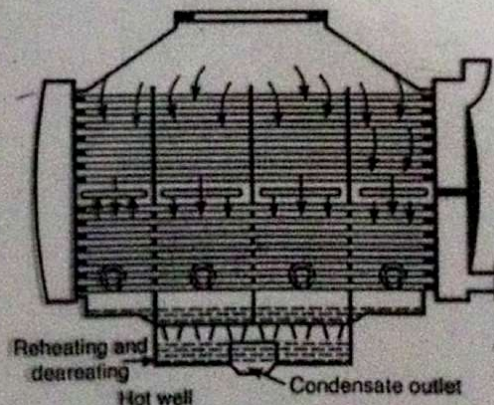
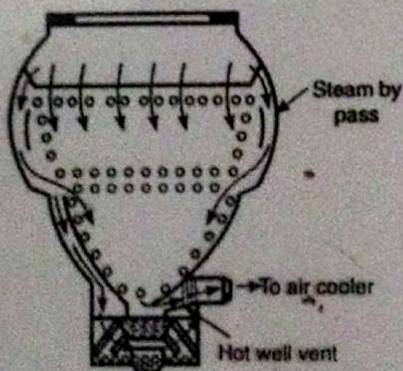
Effective condensation.



CENTRAL FLOW CONDENSER

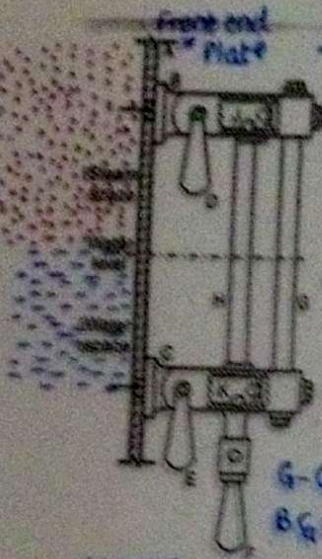
Tubes:

- Red Brass or
- Muntz Metal
- Admiralty Brass
- Stainless Steel



MODERN SURFACE CONDENSER

BOILER MOUNTINGS



- Fittings & devices which are necessary for the boiler safety and control

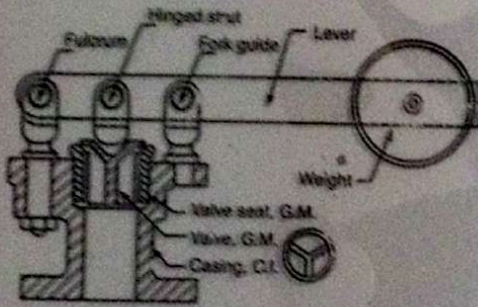
- Integral part of the boiler

- Mounted over the boiler shell

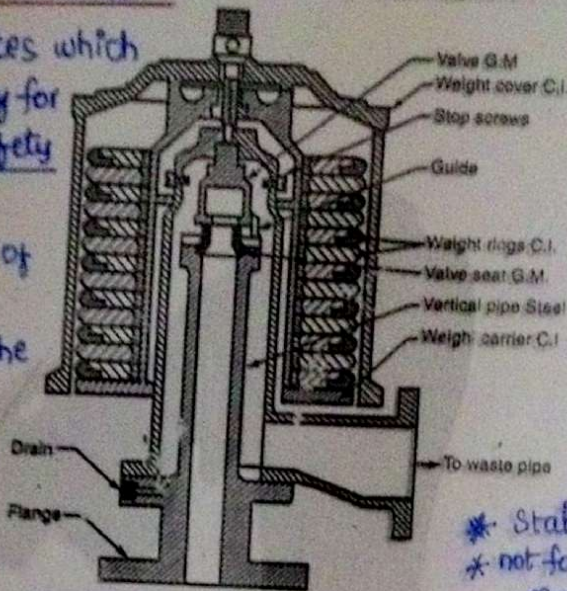
G - Gauge Glass tube
B & C: Stuffing Boxes

WATER LEVEL INDICATOR

- Shows the exact level of water at any instant

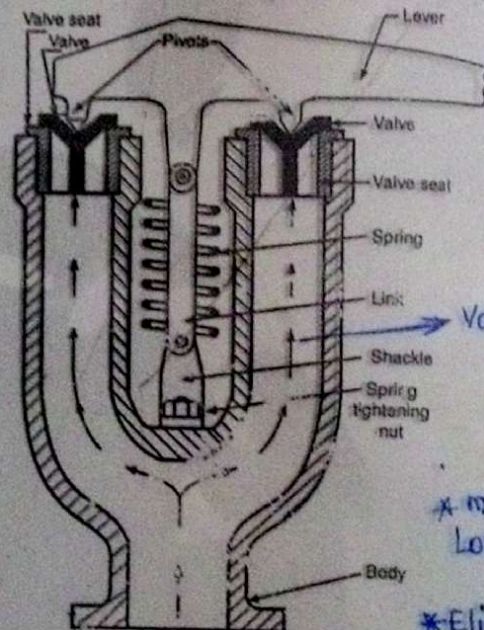


LEVER SAFETY VALVE



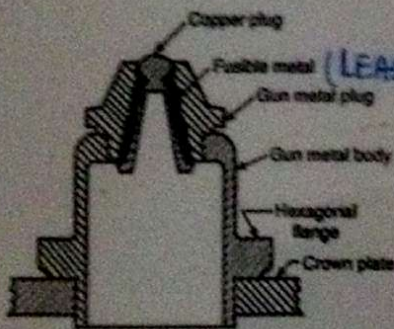
DEAD WEIGHT SAFETY VALVE

* Stationary Boilers only
* not for Locomotive & marine Boilers, need large amount of weights
* effective wth changes with gradient



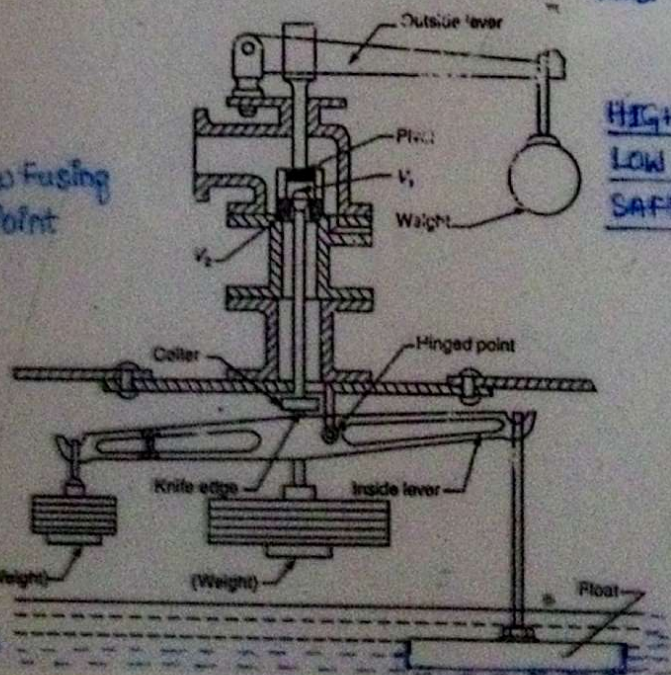
SPRING LOADED SAFETY VALVE

* most suited for Locomotive & marine Boilers
* Elimination of heavy wts.
most widespread use

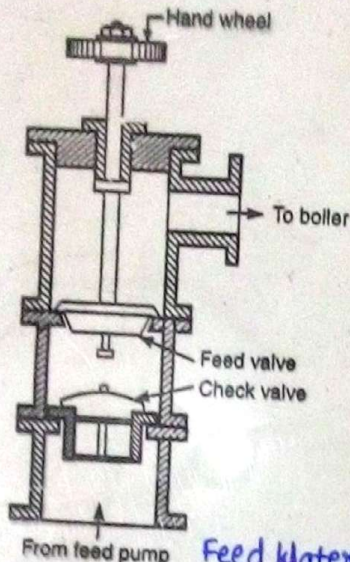


FUSIBLE PLUG

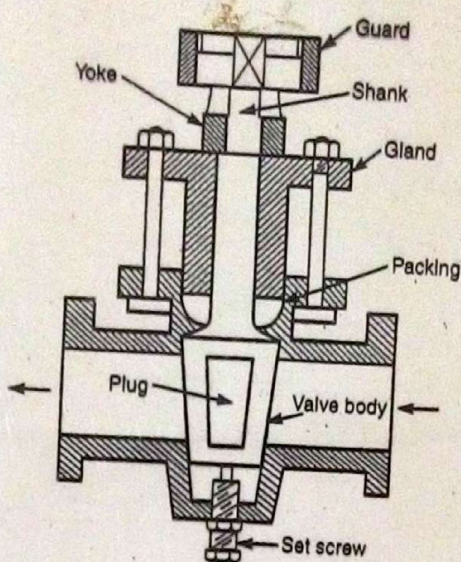
To extinguish fire in the furnace when the water level falls below the safe limit ∴ prevents boiler explosion



HIGH STEAM LOW WATER SAFETY VALVE



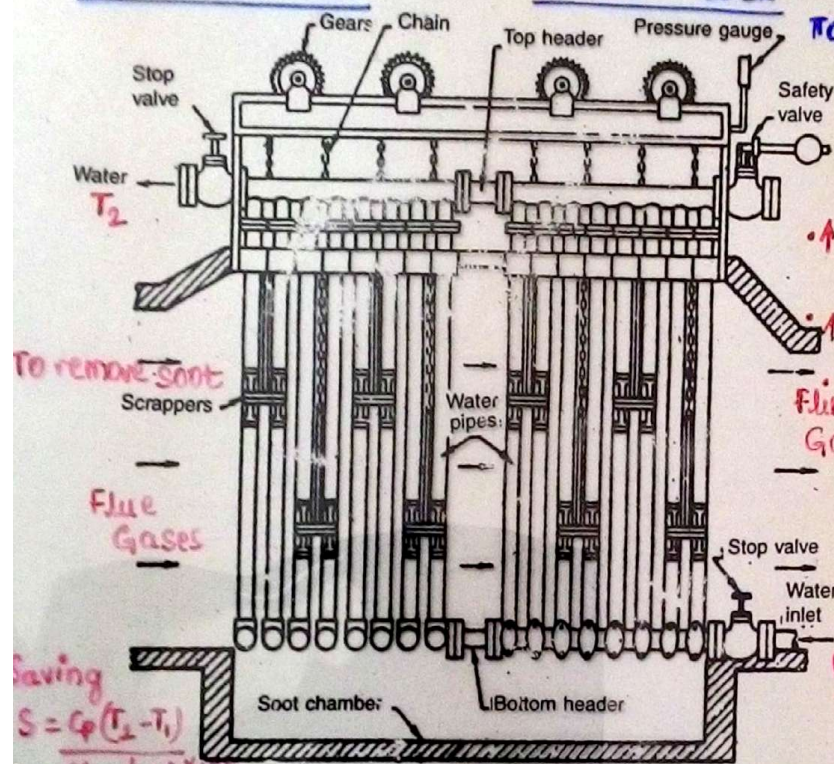
FEED CHECK VALVE



BLOW-OFF COCK



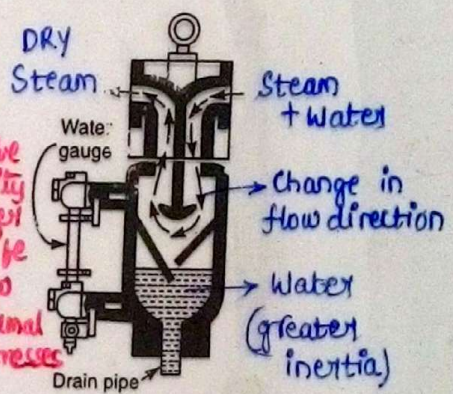
STOP VALVE / JUNCTION VALVE



WATER ECONOMISER

To clean the sediments periodically (mud, scale, impurities)

• ↑ η_{th}
 1% + 6°C
 • ↑ evaporative capacity
 • ↑ Long boiler life
 • ↑ low thermal stresses

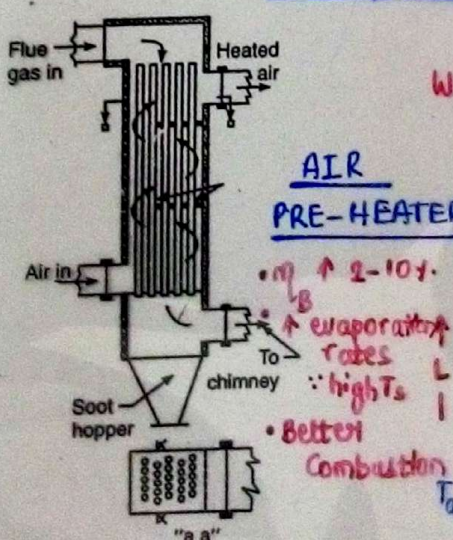


STEAM DRIER (OR) SEPARATOR

Saving $S = \frac{Q(T_2 - T_1)}{(h - h_f) \times 100}$

BOILER ACCESSORIES

- fittings or devices provided to increase the efficiency of boiler & for smooth working

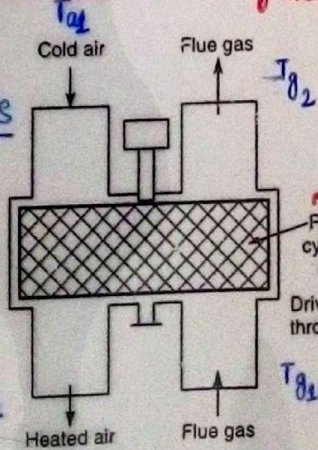


AIR PRE-HEATERS

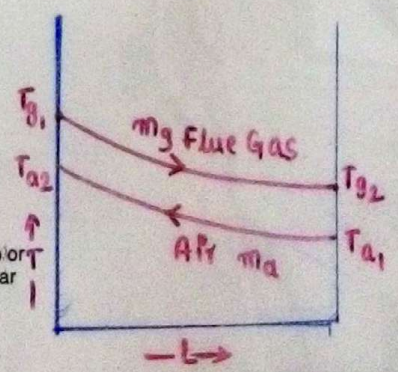
• $\eta \uparrow 2-10\%$
 • ↑ evaporative rates
 • high T_s
 • Better Combustion

RECUPERATIVE AIR-PRE-HEATER

Waste heat recovery device



REGENERATIVE AIR-PRE-HEATER (Ljungstrom Rotary Air Pre-Heater)



2.1 INTRODUCTION :

Steam is a vaporised form of water. It is obtained by boiling or evaporating the water in a boiler.

Steam is widely used as a working fluid in thermal power plant to produce power. It is also used for processing in chemical industries and sugar factories, for bleaching in cotton and paper mills and to drive locomotives and ships.

As a working fluid in power plants, the steam has the following *advantages* :

- high heat capacity
- can easily be condensed and evaporated
- work of compression is very small

In this chapter the steam boilers and their auxiliaries are described.

2.2 STEAM BOILER :

Boiler or Steam generator is a closed steel shell in which steam is generated from the water by combustion of fuel.

The water is pumped into the boiler at operating pressure. The heat of flue gas vaporise the water to form steam. The steam generated in the boiler is superheated in superheater and is then expanded in engine or turbine to produce power.

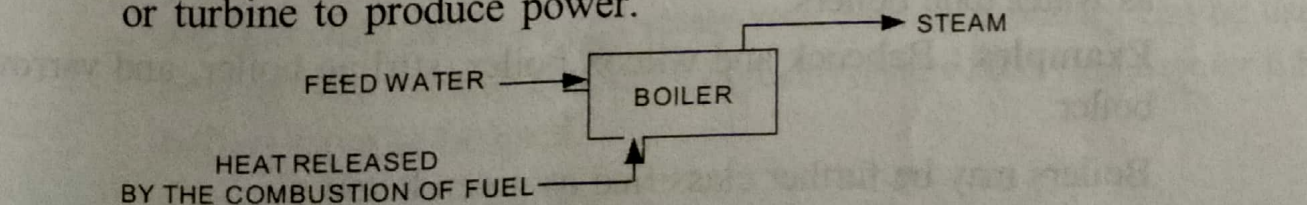


Fig. 2.1

Function of "Boiler" :

Thus the **function** of a boiler is to provide controlled release of heat in the fuel, and efficient transfer of heat to the feed water to produce steam.

Heat Power Engineering
at the desired pressure and temperature. Fig. 2.1 illustrate the essential function of a boiler. The steam can be supplied from the boiler at a constant pressure by maintaining the steam generation rate and the steam flow rate equal.

Steam generating unit :

Combination of apparatus for producing the heat and transferring the available heat to water is called *steam generating unit*. The basic elements of steam generating unit are boiler drum, superheater, economiser and air pre-heater.

Requirements of good boiler :

The primary requirements of a boiler or steam generating unit are :

- It should be capable of producing maximum amount of steam
- It should be occupy less space and should be capable of quick starting
- It should be safe in working i.e., the water must be contained safely and steam must be safely delivered in desired condition.

All parts should be easily accessible for inspection and repairs.

2.3 CLASSIFICATION OF BOILERS :

According to the content in the tubes, the boilers may be classified as

1. Fire tube boilers
2. Water tube boilers

In *fire tube boiler*, the hot gases pass through the tubes surrounded by the water from which the steam is generated.

Examples : Simple vertical boiler, locomotive boiler, cochran boiler, cornish boiler, lancashire, and scotch marine.

In *water tube boiler*, water is made to pass through a large number of tubes over which the hot gases flow. Most modern boilers are built as water tube boilers.

Examples : Babcock and wilcox boiler, stirling boiler, and yarrow boiler.

Boilers may be further classified as given below :

1. Location of furnace

- (a) Internally fired boilers
 - Furnace is located inside the boilers
- (b) Externally fired boilers
 - Furnace is located outside the boilers.

2. Axis of the shell

- (a) Vertical boiler
- (b) Horizontal boiler

3. Circulation of water

- (a) Natural circulation
 - No pump is required; circulation of water is due to difference in densities.
- (b) Forced circulation
 - Uses pump to force water.

4. Service to which they are employed

- (a) Stationary boilers
 - Boilers installed in power plants and industrial units
- (b) Mobile boilers
 - Marine boilers and locomotive boilers

2.4 SELECTION OF BOILER :

The following factors are to be considered while selecting the boiler for a particular use.

- Operating pressure
- Steam generating rate
- Quality of steam required
- Availability of floor space
- Fuel to be used
- Nature of the load on the boiler
- Initial cost.

The two important factors which dictate the selection of boilers are operating pressure and steam rate. Fire tube boilers can be used only upto 25 bar and 10 tons/hr capacity. Above this range water tube boilers have to be used.

2.5 FIRE TUBE BOILER :

In fire tube boiler the furnace is generally located inside the boiler shell or drum, and the hot flue gases are led through the flue tubes which are surrounded by water. Fig. 2.2 shows the essential features of a simple fire-tube boiler having a single large tube.

(Chapter-2)

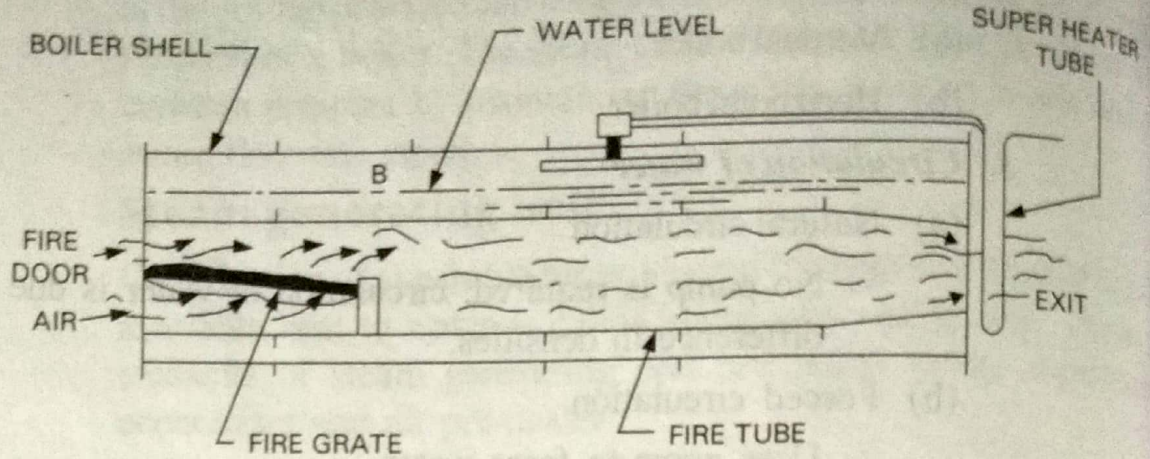


Fig. 2.2 FIRE - TUBE BOILER

The boiler shell contains water and steam under pressure. The fire tube contains the grate. The water level is at B, the space above B being filled with steam. Air for combustion of fuel flowing beneath the grate moves upwards through the fire bars. Burnt flue gases pass along the fire tube to the exit. The water surrounding the tube evaporates with the heat of flue gases. Super heater tubes are generally placed in the path of the flue gases at the exit side.

Fire tube boilers are suitable for capacity upto 10000 kg/hr and low pressure i.e. upto about 20 bar.

Advantages of fire tube boiler :

Fire tube boiler offers the following *advantages* :

- Simple design and more accessible for repairs
- Easy maintenance
- Low initial cost
- Large water content provides safety in operation
- Suitable for fluctuating loads.

However it has the following *disadvantages* :

- Rate of evaporation is low and suitable for only low capacities.
- Bigger in size.
- Lower efficiency.
- Suitable for only low pressures.

2.6 WATER TUBE BOILER :

In water tube boiler the furnace is located outside the drum, and the hot gases pass outside the small tubes containing the water. Fig. 2.3 shows the essential features of water tube boiler. It consists of a large number of parallel tubes containing water and connected to boiler drum through the headers. Feed water is supplied to the drum. From the drum water flows into the tubes via down-take header. Evaporation takes place in the tubes due to heat of flue gases. The steam formed is returned to the drum through up-take header, and steam leaves the drum through a stop-valve.

Because of smaller drum, water tube boiler is suitable for high pressure steam generation, and less time is required to raise the steam.

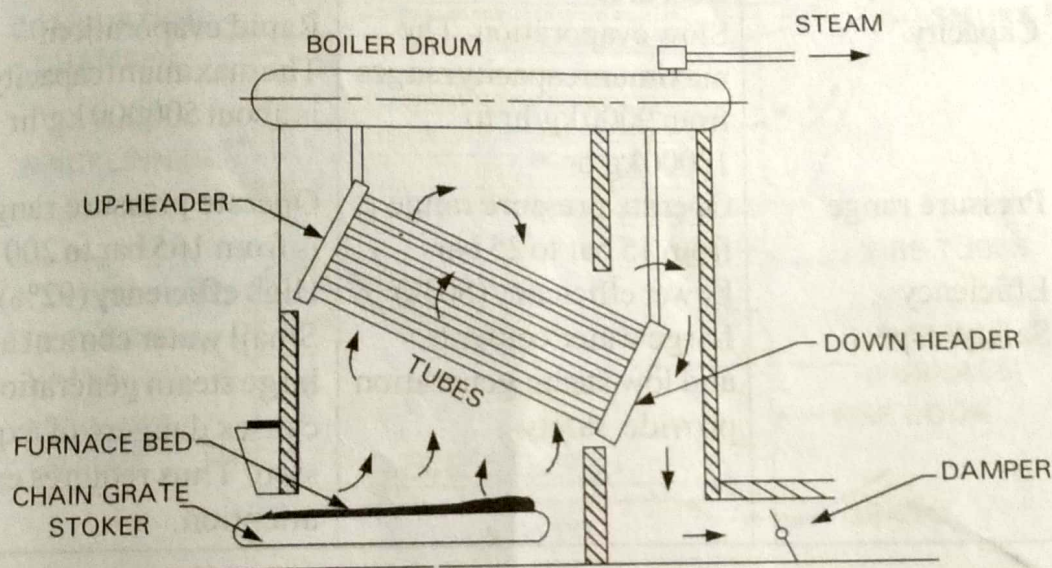


Fig. 2.3 WATER TUBE BOILER

Advantages of water tube boiler :

The *advantages* of water tube boiler are :

- Suitable for high pressure and high capacity
- Quick starting
- Smaller in size for a given output
- Higher efficiency
- Suitable for power plant.

Disadvantages of water tube boiler :

The *disadvantages* of water tube boiler are as under

- High initial cost
- Complicated design and difficult to inspect
- Impure and dirty water is not suitable as feed water
- Small water content causes the danger of explosion

Comparison of water tube and fire tube boilers

The comparison of two types of boilers is summarised below :

Table 2.1

Criterion	Fire tube boiler	Water tube boiler
Flow of hot gases and feed water	Hot gases flow inside the tubes surrounded by water	Water flows inside the tubes, hot gases flow outside the tube.
Location of furnace	Furnace will be within the boiler shell	Furnace will be situated outside the boiler.
Floor area	Bigger in size for given output; requires large floor area	Smaller in size for given output; requires smaller floor area.
Capacity	Slow evaporation. The maximum capacity ranges from 9000 kg/hr to 10000 kg/hr	Rapid evaporation; The maximum capacity is about 500000 kg/hr
Pressure range	Operate pressure range is from 15 bar to 25 bar	Operate pressure range is from 165 bar to 200 bar
Efficiency	Lower efficiency (80%)	High efficiency (92%)
Safety aspect	Large water content and low steam generation provide safety	Small water content and large steam generation causes dangers of explosion. Thus requires expert attention.

2.7 COCHRAN BOILER :

Cochran boiler is a vertical, internally fired, natural circulation, fire tube boiler. Boiler may be coal fired or oil fired. Coal fired boilers have a thermal efficiency of 70% and oil fired 75%.

Design features (construction details) :

The cochran boiler with its mountings is shown in Fig. 2.4. It has several horizontal tubes to increase the heating surface. These tubes connect the combustion chamber and smoke box, and sufficient space is provided around them for water. Combustion chamber is connected to fire box with the help of flue tube. The combustion chamber is lined with fire bricks to prevent overheating of plates. Below the firebox a fire grate is provided. Beneath the grate there is an ash pit to collect the ash. Chimney provided above the smoke box serves for the escape of gases. The man hole is provided for inspection and repair of the interior of the shell.

The top of the shell and firebox are made hemispherical to provide maximum volume and maximum strength for a boiler.

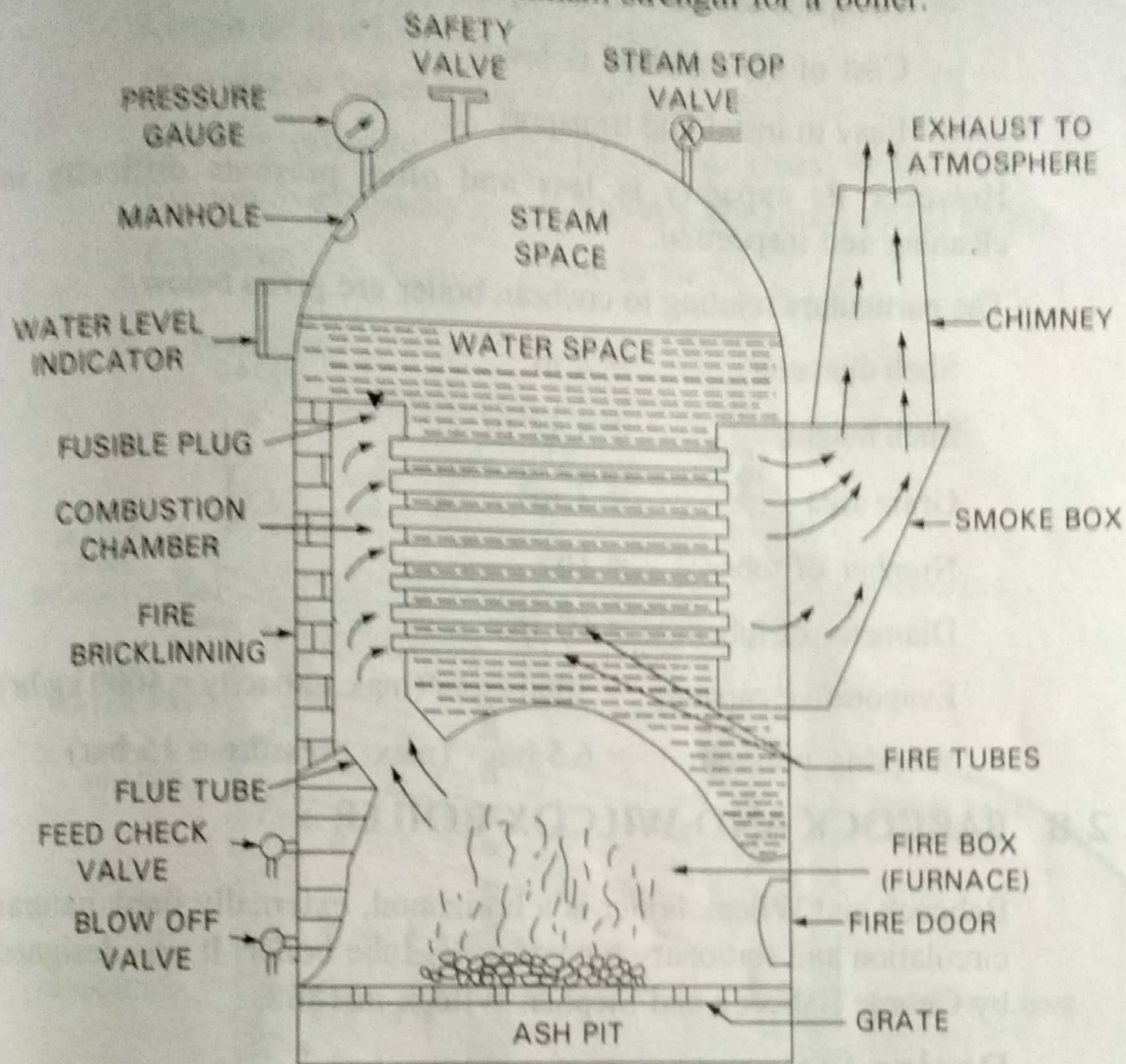


Fig. 2.4 COCHRAN BOILER

Working :

The boiler is filled with water to a specified level. The flue tubes are completely submerged in the water.

The coal is introduced through the firedoor on to the grate. In oil fired boiler, the oil is directly sprayed into the furnace. Fuel is burnt in the fire box and the hot gases from the fire box pass to the combustion chamber through a short flue pipe. From the combustion chamber they pass through the horizontal fire tubes to the smoke box and from there the gases are discharged to the atmosphere through the chimney. The hot gases while passing through the fire tubes transfer their heat to surrounding water. The water may also receive heat from the fire box (or furnace) directly. The water is evaporated to form steam. The steam formed is accumulated in the steam space. It is allowed to flow through the steam stop valve to the steam supply pipe.

The *advantages* of this boiler are :

- Compact design and occupies less space
- Cost of construction is low
- Easy to instal and transport

However its capacity is less and often presents difficulty in cleaning and inspection.

The particulars relating to cochran boiler are given below :

Shell diameter	= 2 m
Shell height	= 4.42 m
Grate area	= 2 m ²
Number of tubes	= 165
Diameter of tubes	= 60 mm
Evaporative capacity	= 3000 kg/hr (max. capacity = 4000 kg/hr)
Working pressure	= 6.5 bar (max. pressure = 15 bar)

2.8 BABCOCK AND WILCOX BOILER :

Babcock and Wilcox boiler is a horizontal, externally fired, natural circulation and stationary type of water tube boiler. It was designed by George Babcock and Stephen Wilcox in 1868.

Design features :

Babcock and Wilcox boiler consists a drum which contains water and steam. The drum is connected to the headers by short tubes. The header at the right hand end of the drum is called downtake header and the other at left handed is called uptake header. A series of inclined tubes (inclination is about 10°-15° with horizontal) are connected to these headers. A hand hole is provided in the header in front of each tube for cleaning the tubes. A mud box is provided just below the downtake header to collect the sediments and are removed from time to time through blow off pipe. Furnace is located below the uptake header. Baffles are provided to deflect the hot gases so as to flow along a longer paths. It is also provided with superheater which is placed in the path of flue gases.

Babcock and Wilcox boiler with its mountings is shown in Fig. 2.5. The particulars of the boiler are given below :

Diameter of drum	= 1 - 2 m
Length of drum	= 6 - 10 m
Diameter of water tubes	= 100 mm
Working pressure	= 10 - 20 bar (max. 40 bar)
Evaporative capacity	= 20000 kg/hr (max 40000 kg/hr)
Efficiency	= 60 to 80 %

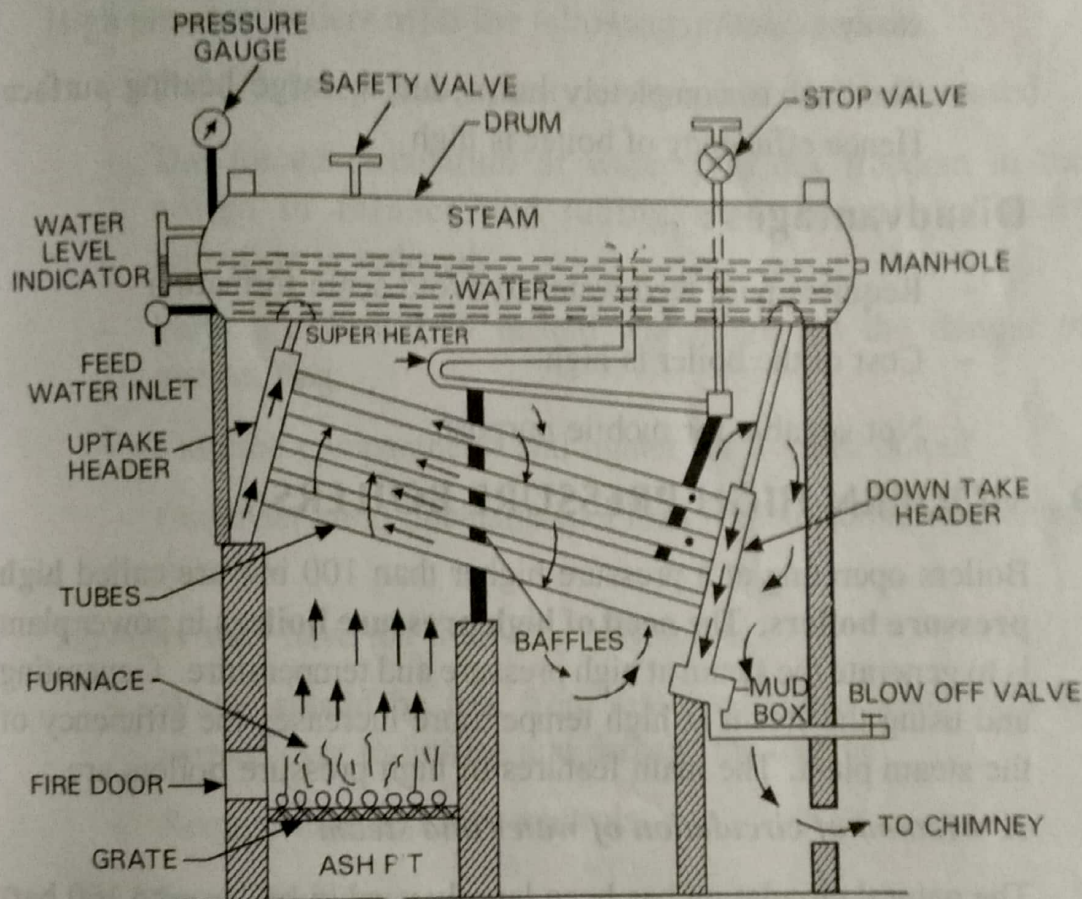


Fig. 2.5 BABCOCK AND WILCOX BOILER

Working :

Water from the drum enters the bottom header and flows through the tubes. The hot gases flowing over the tubes heat and evaporate the water flowing through the tubes. Due to low density, the mixture of water and steam enters the drum through the uptake header. Steam is separated out in the drum and collected above the water level.

The steam from the drum is led into superheater which is placed in the path of hot gases. Since the superheater is exposed to the hot gases, the steam passing in it will be superheated. Superheated steam is supplied to the prime mover through steam stop valve.

Advantages :

- Generates steam at high pressure and rate of steam generation is high
- For given output, it occupies less space
- Parts are accessible for cleaning, inspection and repair
- Parts of a boiler can be separated, transported and erected easily
- The fuel is completely burnt due to large heating surface. Hence efficiency of boiler is high

Disadvantages :

- Requires pure feed water to avoid scale formation
- Cost of the boiler is high
- Not suitable for mobile purpose.

2.9 MODERN HIGH PRESSURE BOILERS :

Boilers operating at a pressure higher than 100 bar are called high pressure boilers. The need of high pressure boilers in power plant is to generate the steam at high pressure and temperature. Generating and using the steam at high temperature increases the efficiency of the steam plant. The main features of high pressure boilers are

1. Method of circulation of water and steam

The natural circulation has been largely used in boiler upto 160 bar. Above that forced circulation is employed. Forced circulation is caused by a centrifugal pump driven by some external source.

2. Arrangement of drum and tubing

The size of water drum is very small and in certain cases it is completely eliminated. Water flowing through pipe circuit is evaporated and converted into steam continuously. To minimise the resistance to flow, the water is passed through a parallel system of tubing.

3. Improved method of heating

The improved method of heating can be achieved by

- Providing large heating devices such as superheater, economiser and air preheater.

- Increasing the heat flow through the tubes by using hot gases travelling with a supersonic speed
- Saving of latent heat by evaporation of water at pressure above the critical pressure
- Using the solid fuels in the pulverised form

Advantages and limitations of high pressure boiler :

High pressure boilers offer the following *advantages*

- The efficiency and capacity of the plant can be increased
- The forced circulation of water provides freedom in the design of furnace and tubing, and tendency of scale formation is reduced
- Parts are uniformly heated which prevent the danger of overheating
- Extreme compactness, and lighter for a given output
- Can meet rapid fluctuations of load without additional control devices.

However they have the following *limitations*.

- Water should be free from dissolved solids which will be carried over to turbine and damage the turbine
- Require high sensitive controls

2.9.1 La Mont Boiler :

This boiler was introduced by La Mont in 1925. It is a water tube, high pressure, forced circulation boiler. Forced circulation is achieved by centrifugal pump which is driven by a steam turbine. It deals with smaller quantity of water at high pressure and the high rate of heat transmission can be achieved by increasing the velocity of water.

The feed water passes through economiser to a steam storage and separating drum from which it is drawn to the circulation pump. The pump delivers the feed water to the evaporator tubes. Water circulation is about ten times the steam generated. The large circulation of water prevents over heating the tubes. The water is rapidly evaporated in the tubes and the mixture of water and steam (wet steam) enters the drum. The steam is separated in the drum and is then drawn through

the superheater which is of the convective type. Superheater is placed in the path of flue gases, and steam flowing through the superheater tube is superheated by absorbing heat of flue gases. Finally, the super heated steam at high temperature is delivered to prime mover.

Fig. 2.6 shows various components of La Mont boiler.

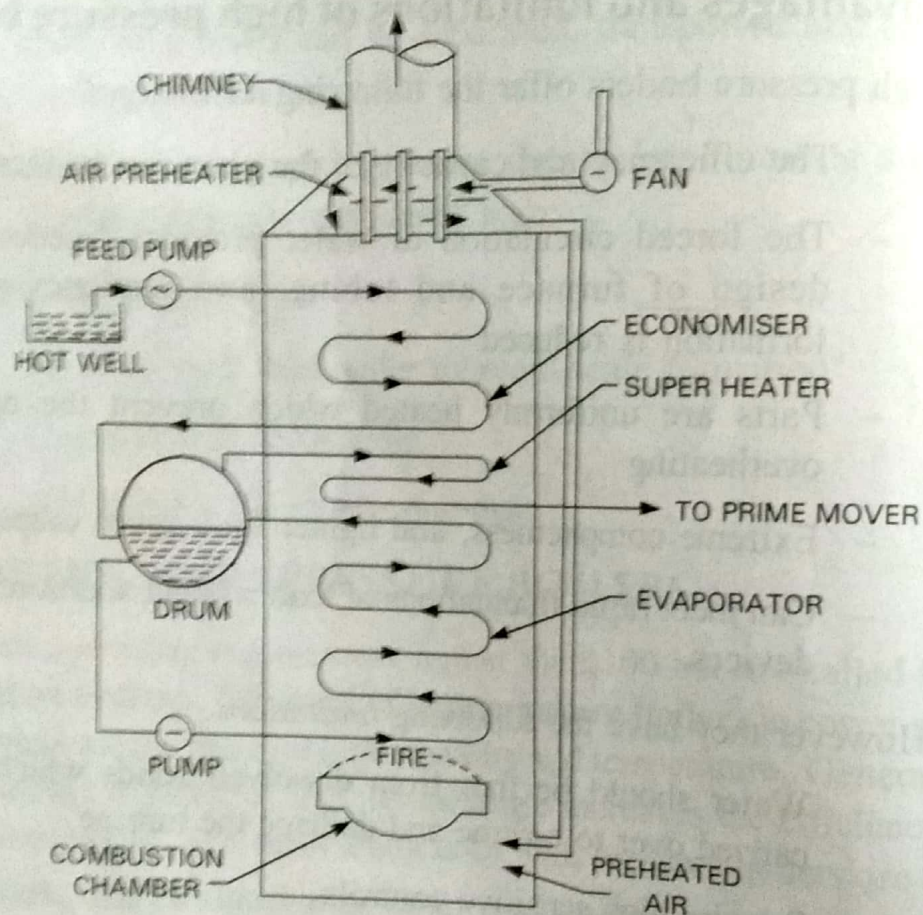


Fig. 2.6 LA MONT BOILER

The level of water in the drum is maintained constant by supplying feed water equal to the steam drawn off. The air is preheated by the flue gases before entering into the combustion chamber. The preheated air improves the combustion efficiency.

The La Mont boiler is designed to produce 45000 to 50000 kg of superheated steam per hour at a pressure of 125 bar and at a temperature of 500°C.

La Mont boiler has the following *advantages*

- Lighter in weight for a given output
- Smaller drum makes boiler compact
- No overheating of the tubes
- Absence of scale formation in the tubes
- Easy starting of the boiler

2.9.2 Benson Boiler :

The idea of generating steam at critical pressure to avoid formation of bubbles was suggested by Mark Benson in 1922. Benson boiler shown in Fig. 2.7 can operate at critical pressure (225 bar) where the steam and water have the same density, and therefore, the danger of bubble formation can be easily eliminated and water transforms into steam without boiling (i.e., latent heat is zero). The Benson boiler was first operated in 1927 in West Germany.

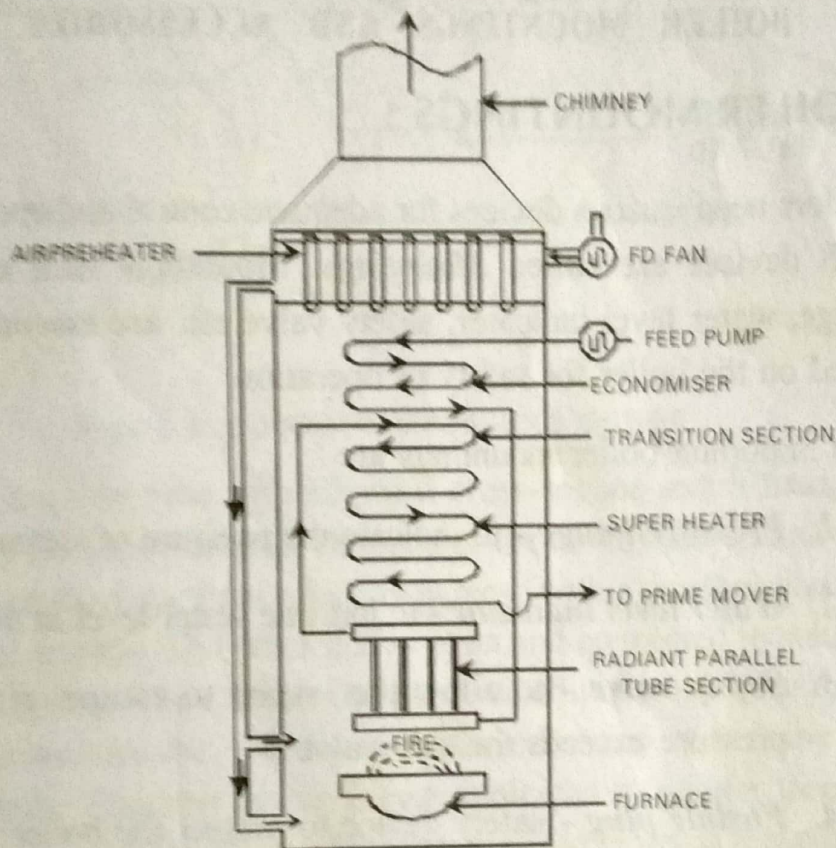


Fig. 2.7 BENSON BOILER

In Benson boiler the drum is completely eliminated. It employs a forced circulation and steam is generated in a single pass. Thus, it is a drumless *once-through boiler* takes boiler feed water in at one end and discharges it as superheated steam at the other end. The feed water at high pressure is circulated through economiser. The water is then pumped through radiant parallel tube section to evaporate partly. The water and steam then move to the transit section where it is converted into steam. The steam is then passed through the superheater where the steam is superheated. Finally, the superheated steam is supplied to prime mover.

In Benson boiler steam can be produced at about 225 bar pressure, but it requires large energy to operate the pump. For this reason, it is operated at about 210 bar. The evaporative capacity is about 135000 kg/hr and its thermal efficiency is about 90%.

It has the following *advantages*.

- Light in weight and occupies smaller floor area
- Can be started very quickly
- Elimination of drum reduces the cost of boiler
- Steam at supercritical pressure can be generated
- Explosion hazards are not severe

BOILER MOUNTINGS AND ACCESSORIES

2.10 BOILER MOUNTINGS :

Boilers need various devices for adequate control and operation, and such devices are called *Mountings*. Mountings such as pressure gauge, water level indicator, safety valve etc. are *essential* devices fitted on the boiler for safety of operation.

The important boiler mountings are

1. *Pressure gauge* - to indicate the pressure of steam generated
2. *Water level indicator* - to indicate water level in the boiler
3. *Safety valve* - to allow the steam to escape if the steam pressure exceeds the safe valve
4. *Fusible plug* - safety device to protect the boiler from over heating
5. *Feed check valve* - non return valve which allows the feed water to enter the boiler
6. *Blow off cock* - for removal of sediments
7. *Stop valve* - to regulate the steam flowing out of the boiler
8. *Man hole* - door to enter the boiler for inspection

2.10.1 Pressure Gauge :

The function of pressure gauge is to indicate the steam pressure inside the boiler. It is usually mounted on the front top of the shell or drum. The Bourdon pressure gauge (Fig. 2.8) is most commonly used.

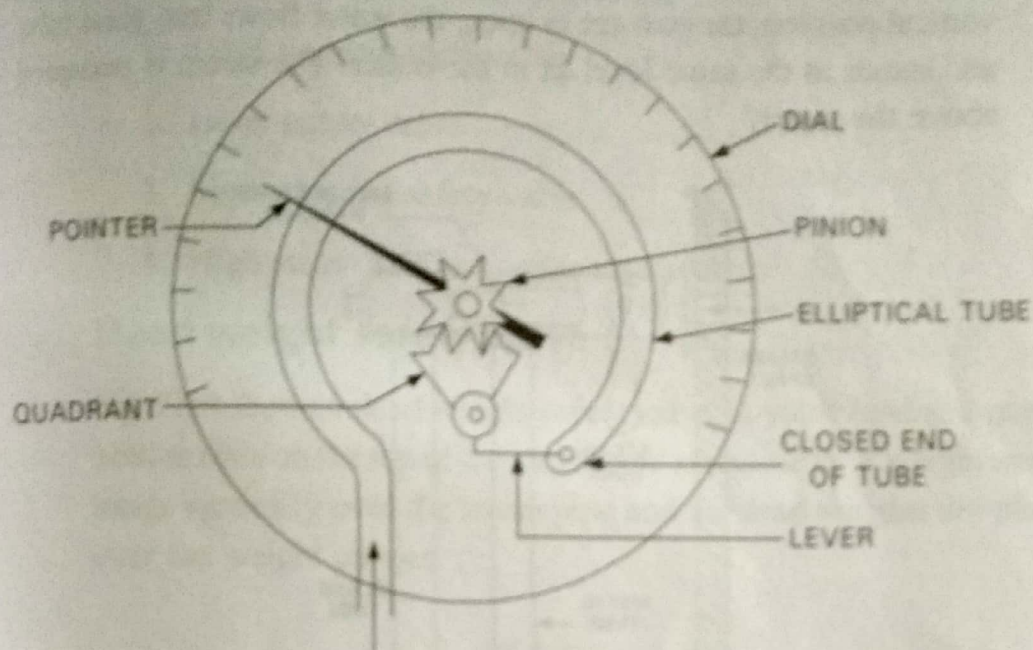


Fig. 2.8 BOURDON PRESSURE GAUGE

It has a spring tube with elliptical cross-section and is made of high quality phosphor bronze. One end of the tube is closed and attached to the toothed quadrant which is in mesh with a small pinion fitted on central spindle. The other end is open and connected to steam space of the boiler. When the steam enters the elliptical tube, the tube tends to become circular which causes the other end of the tube to move outwards. This movement is communicated to pointer through link and sector. The pressure of the steam in the boiler is recorded on the dial which is graduated to read the pressure above atmospheric pressure.

The pressure gauge is fitted to the boiler through U-tube siphon which is connected to the steam space of the boiler.

2.10.2 Water Level Indicator :

The water level indicator is a safety device which is used to indicate the water level in the boiler. Usually two water level indicators are fitted in front of the boiler.

A common form of glass tube water level indicator (water gauge) is shown in Fig. 2.9. It has two horizontal tubes which are bolted to the boiler. Bottom one is connected to the water in the boiler and top one to the steam space. Two cocks (steam and water cocks) control the

passage between the boiler and the glass tube. When the cocks are in vertical position, the passage is open, the water flows into glass tube and stands at the same level as in the boiler. The steam is occupied above the water.

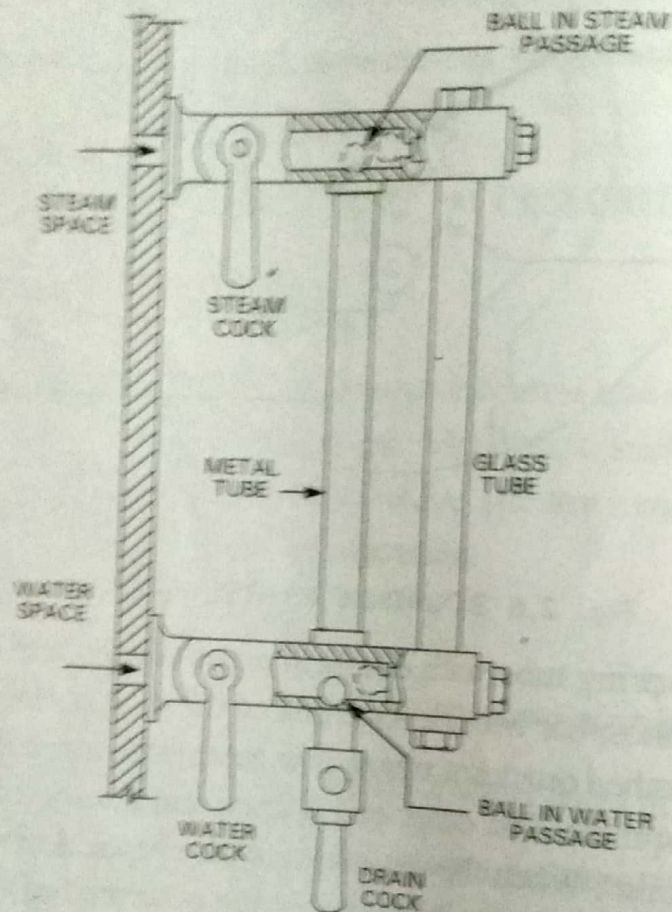


Fig. 2.9 WATER LEVEL INDICATOR

There are two balls in the steam and water passages which rest in their position during working. In case the glass tube is broken, the balls are carried (by flow of water and steam) along their passages to the ends of the glass tube. Thus the steam and water will not escape out. The glass tube can easily be replaced by closing the steam and water cocks. Drain cock is used to ascertain that the steam and water cocks are clear.

2.10.3 Safety Valve :

The function of safety valve is to maintain safe pressure inside the boiler. It is located above the steam space. If the pressure of steam exceeds the design pressure, the valve is lifted automatically and steam escapes out reducing the pressure inside the boiler. Escaping steam produces a whistling sound. Thus raising alarm to the boiler attendant.

The common types of safety valves are

1. Dead weight safety valve
2. Lever safety valve
3. Spring loaded safety valve
4. High steam and low water safety valve

Dead weight safety valve :

Fig. 2.10 shows a dead weight safety valve. A valve having a conical bottom rests on the top of the steam pipe and close it. A weight carrier hangs vertically over the steam pipe and the dead weights are placed over the weight carrier.

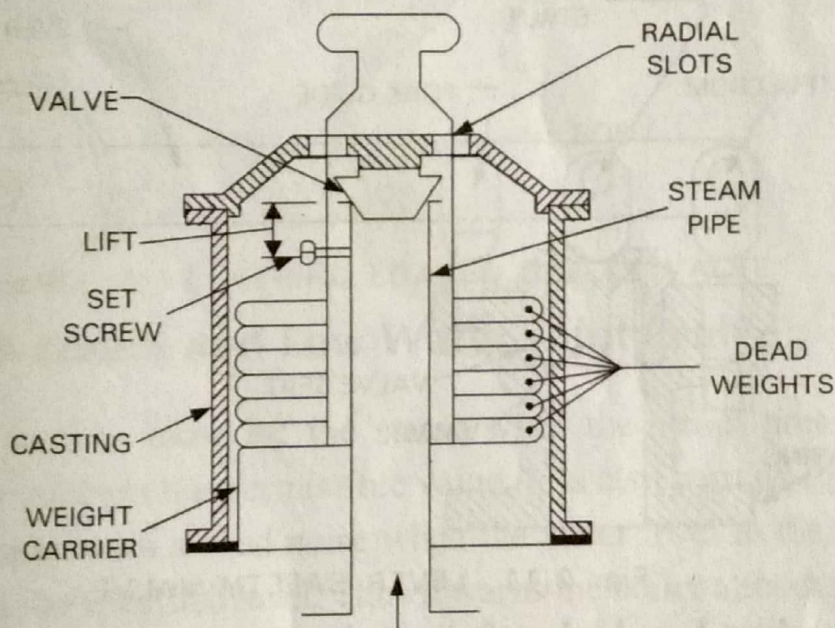


Fig. 2.10 DEAD WEIGHT SAFETY VALVE

When the steam pressure in the boiler exceeds the safe working pressure, the valve along with carrier lifted up from the seat against downward thrust of the dead weights and allows the excess steam to escape through radial slots. The lift of the valve is regulated by the set screw.

The dead weight safety valve is suitable for stationary boilers because it is affected by vibrations. It is not suitable for high pressure boilers as it requires large amount of dead weight.

Lever safety valve :

A lever safety valve is shown in Fig. 2.11. It maintains constant safe pressure inside the boiler. The valve is loaded by means of a lever which is hinged at one end and at the other end it carries a weight. The valve rests in seat which is fixed in the body. The lever with its load exert downward thrust on the valve. When the pressure of steam exceeds the safe limit, the upward thrust of steam raises the valve from its seat. This allows the steam to escape till the pressure falls back to its normal value. The valve then returns back to its original position.

The *advantage* of lever safety valve is that it does not requires heavy weight. But it is suitable for stationary boilers only as it is affected by vibration.

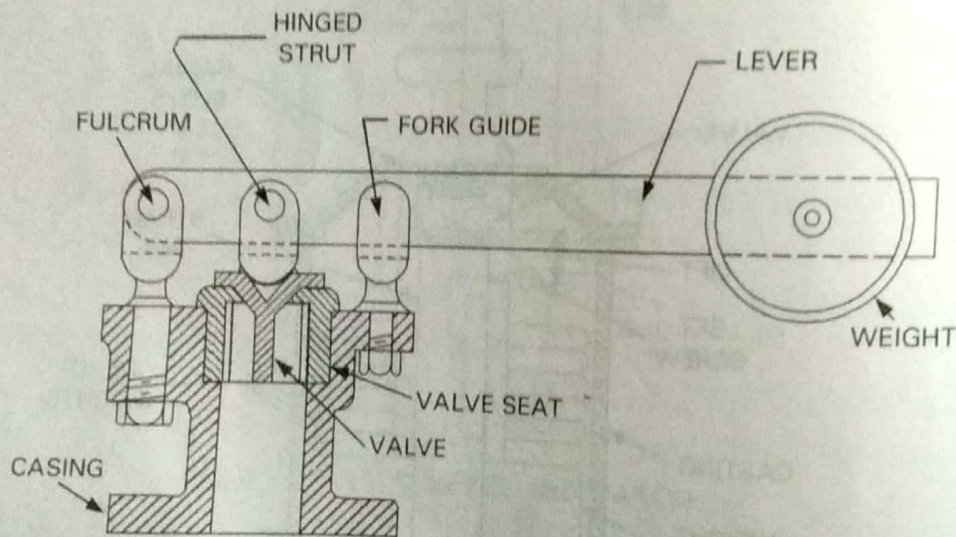


Fig. 2.11 LEVER SAFETY VALVE

Spring loaded safety valve :

The spring loaded safety valve is shown in Fig. 2.12. It consists two separate valves and one lever which acts on the two valves. The valves are closed by the tension of helical spring connected between the lever and the shackle. When the steam pressure exceeds the safe working pressure, both valves are lifted up against the action of the spring and excess steam escape out, after which the valves are again closed due to spring tension.

In the event of spring failure, both valves and lever are held in position by a link which connects the link and the valve.

Spring loaded (Rams bottom) safety valve is suitable for both mobile and stationary boilers.

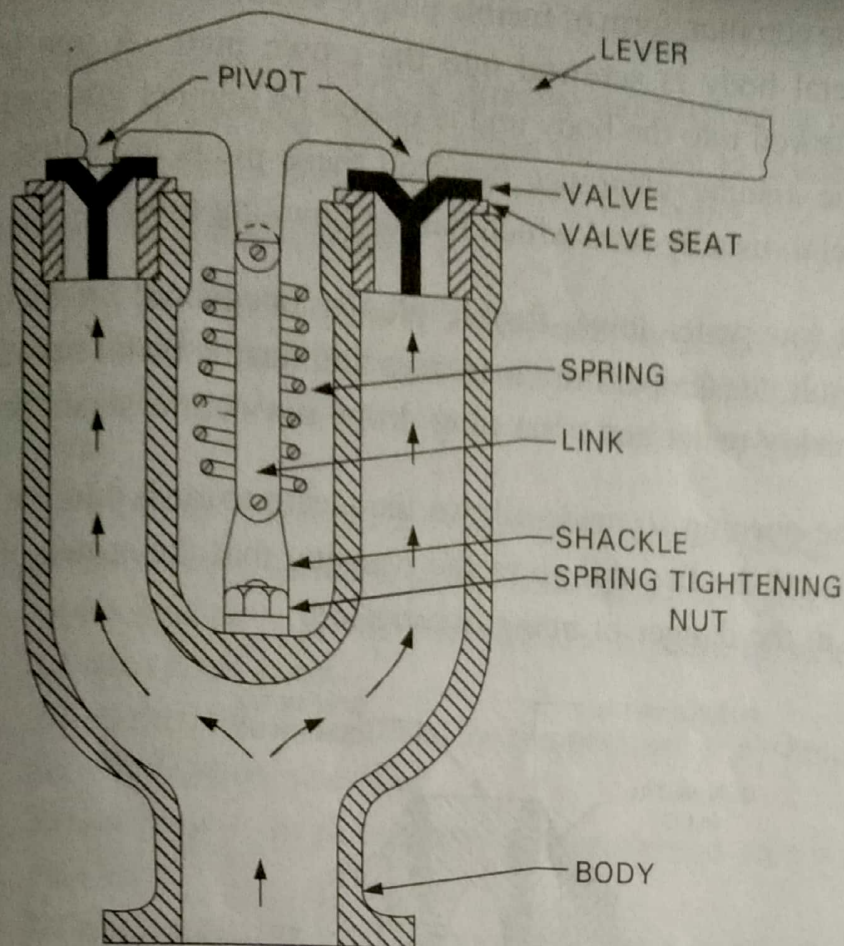


Fig. 2.12 SPRING LOADED SAFETY VALVE

High steam and Low Water safety valve :

It is used to blow off the steam when the steam pressure in the boiler exceeds the permissible value. It also permits the escape of the steam with a loud noise when the water level in the boiler falls below the specified level. Thus it warns the boiler attendant about the fall in the water level in the boiler. Since it serves as a safety device against high pressure and low water level, it is called as high steam low water safety valve alarm. It is generally mounted on fire tube boilers.

2.10.4 Fusible Plug :

It is a safety device fitted over the crown of the furnace or combustion chamber. The *main function* of fusible plug is to put off the fire in the furnace when the water level in the boiler falls below the specified level. It avoids the explosion due to overheating of the tubes and shell.

The common form of fusible plug is shown in Fig. 2.13. A hollow gun metal body is screwed into the crown plate. A gun metal plug is screwed into the body and is closed by another gun metal cone plug. The annular clearance between these plugs are filled with fusible metal, usually lead which has low melting point.

At low water level, fusible plug is uncovered by the water. As a result, the temperature increases and fusible metal between the plugs quickly melts and cone plug drops out due to steam pressure.

The opening so made allows the steam to rush into the furnace and put off the fire and gives the warning that the crown of the furnace is in the danger of being overheated.

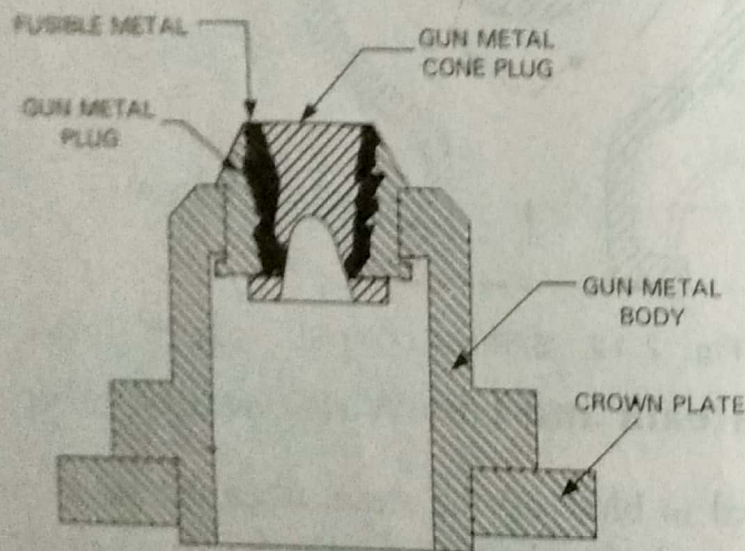


Fig. 2.13 FUSIBLE PLUG

2.10.5 Feed Check Valve :

Feed water at high pressure is supplied to the boiler through feed check valve. It regulate the rate of flow of feed water into the boiler. In the event of failure of feed pump, the boiler pressure becomes more than the feed pump side, this closes the check valve and prevent the escape of the water from the boiler.

2.10.6 Blow off Cock :

The blow off cock is fitted to the lowest part of the boiler. The functions of blow of cock are to empty the boiler when ever required, and to discharge the mud and the sediments which are collected at the bottom.

2.10.7 Stop Valve :

Stop valve or junction valve is directly mounted on a boiler and connected to steam pipe which carries the steam to primemover. It is used to control the flow of steam from the boiler.

2.11 BOILER ACCESSORIES :

The auxillary devices installed with the boiler for proper operation and to increase the efficiency of the plant are known as *boiler accessories*. The following accessories are more commonly employed in steam power plant.

1. **Feed pump** - to force the feed water into the boiler
2. **Economiser** - to increase the temperature of feed water
3. **Super heater** - to increase the temperature of steam above the saturation point
4. **Air preheater** - to increase the temperature of air entering into the combustion chamber
5. **Steam trap** - to drain away the condensed steam from the pipe line
6. **Steam separator** - to remove the water particles from the steam flowing to engine
7. **Steam injector** - to supply the feed water to the boiler at high pressure

Fig. 2.14 represents the relative positions of important accessories indicating their functions.

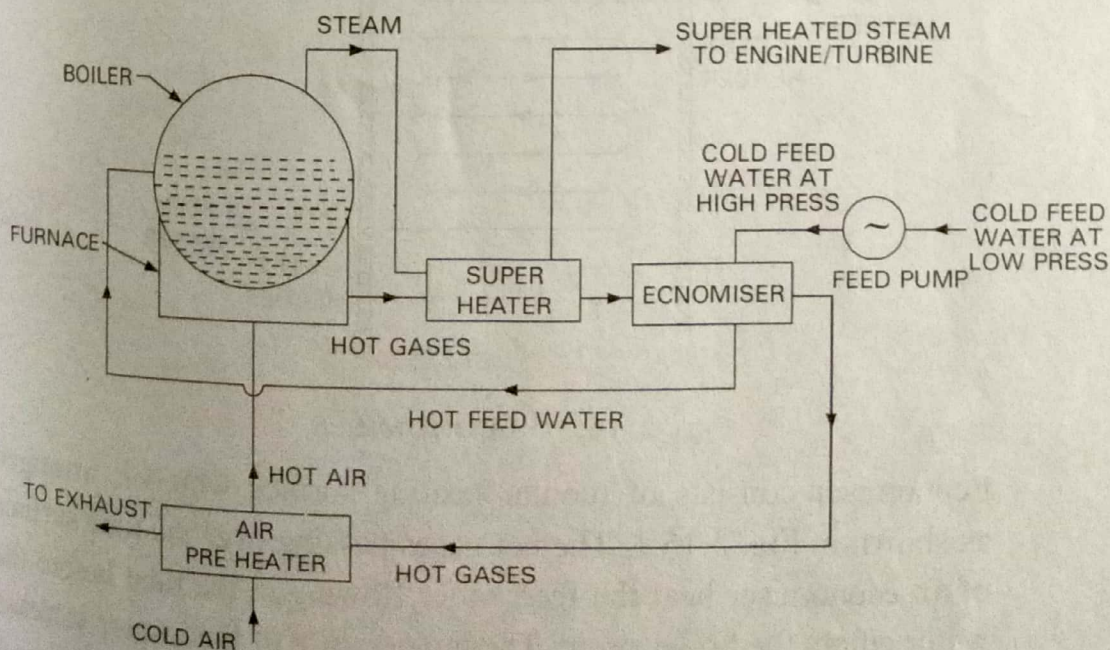


Fig. 2.14 BOILER ACCESSORIES

2.11.1 Feed Pump :

The function of feed pump is to force the feed water into the boiler at high pressure.

The following types are commonly used

1. Reciprocating pump, and
2. Rotary pump

Reciprocating pumps are driven by directly coupling them to steam engine. Rotary pumps (centrifugal pumps) are driven by electric motor or steam turbine.

The rotary pumps are generally used when a large quantity of feed water is to be supplied to the boiler.

2.11.2 Economiser :

Flue gases leaving the boiler contain large amount of heat energy at a temperature higher than the steam saturation temperature. Part of this energy can be recovered in an economiser and is used to heat the feed water. It is usually placed between boiler furnace and air preheater.

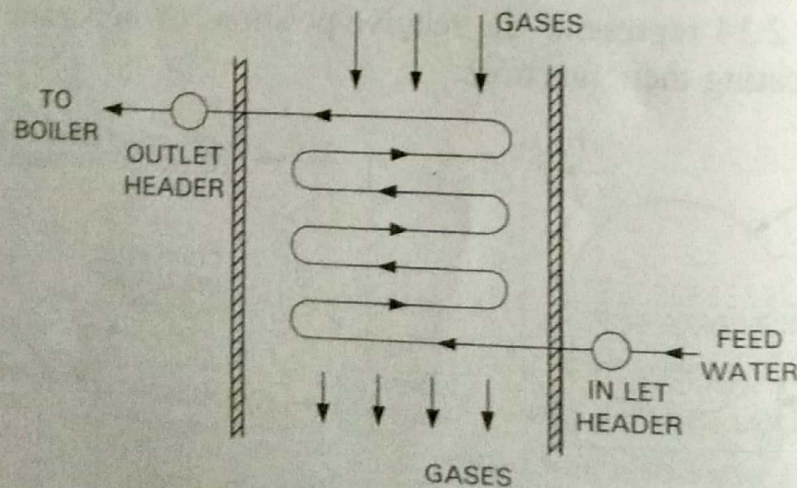


Fig. 2.15.1 ECONOMISER

Economiser consists of tubular heating surface which is arranged as shown in Fig. 2.15.1. The hot gases passing over the tube surface of an economiser heat the feed water flowing in the tube before the water enters the boiler drum. The temperature of feed water increases due to heat transfer from hot gases to water. Thus the function of

economiser is to heat the feed water by utilising the energy from hot gases. In economiser only feed water is heated to near saturation temperature, but no steam generation; also, there is no recirculation path.

Green's Economiser :

Green's economiser is most commonly used in steam power plants. It consists of transverse section of vertical tubes which are connected to horizontal pipes (boxes) at top and bottom. The tubes are placed such that hot gases can freely pass over them. The top boxes are connected to pipe C and the bottom boxes are connected to the pipe D. The feed water is pumped into economiser at E. Then it passes into the bottom box through the pipe D. It again goes up through the tubes into top box. From there it is led to the pipe C and then to the boiler. At the end of the pipe C, a safety valve is provided to blow off if the pressure is high.

To remove the soot which is formed on the surface of the tubes, scrapers are provided in groups. The adjacent groups of scrapers are connected by chain which pass over the pulley. The pulleys are driven by worm and worm wheel as shown in Fig. 2.15.2.

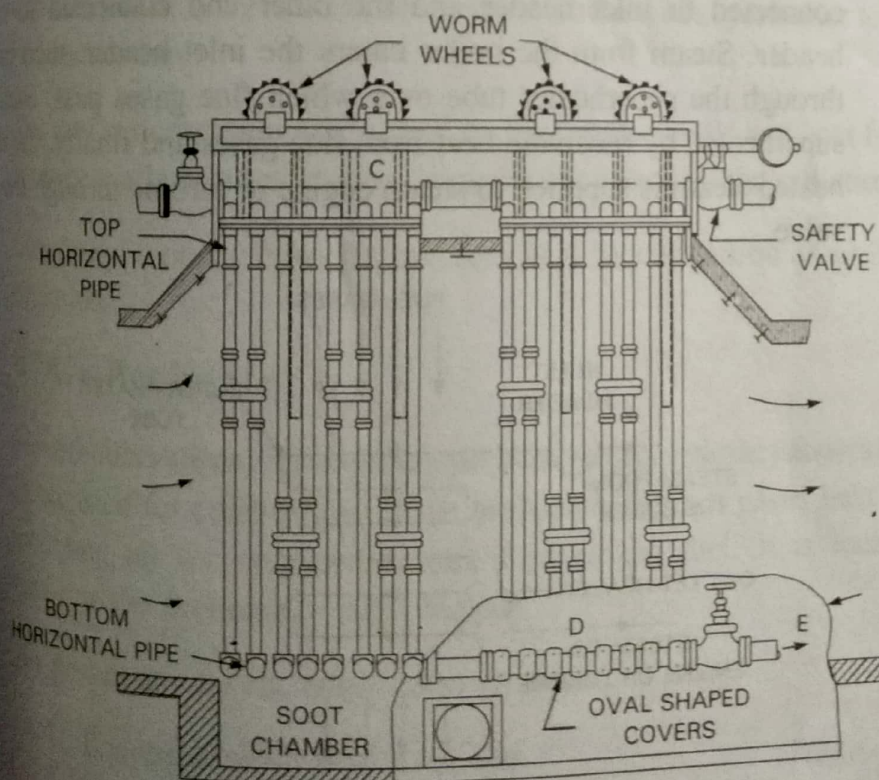


Fig. 2.15.2 GREEN'S ECONOMISER

(Chapter-2)

Advantages and disadvantages :

The *advantages* derived by providing economiser are

1. Fuel economy - About 15 to 20% fuel may be saved for same output.
2. Increase in steam generation rate of the boiler.
3. Increase in thermal efficiency of the boiler plant.
4. The life of boiler is increased due to reduction of thermal stresses.

However, economiser increases the draught requirement, and the surfaces are coated with dirt which reduce the heat transfer.

2.11.3 Superheater :

Super heater is placed in the path of flue gases. It consists of a set of tubes through which steam passes and hot flue gases pass over the tubes. The steam is further heated in the tubes by utilising the heat of flue gases, and due to this the temperature of steam is increased above the saturations temperature. Thus the *function* of a superheater is to increase the temperature of steam by utilising the heat of flue gases. Super heating raises overall cycle efficiency, and also reduce the initial-condensation losses in steam engine.

Fig. 2.16 shows a simple type of superheater. One end of the tube is connected to inlet header and the other end connected to outlet header. Steam from the boiler enters the inlet header, then passes through the superheater tube over which flue gases pass. Steam is superheated by receiving heat from flue gases and finally, the super heated steam is supplied to steam engine or turbine through the stop valve.

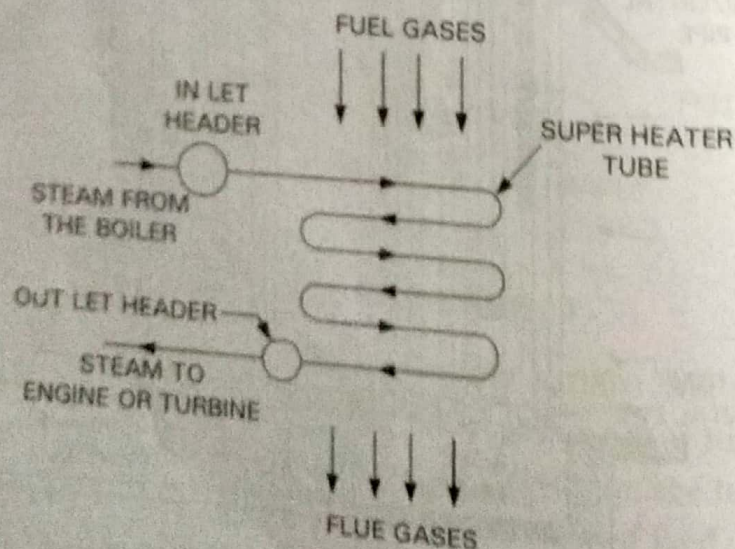


Fig. 2.16 SUPERHEATER

Advantages :

The *advantages* derived by using superheater are

- Reduce the steam consumption per given output
- Reduce the condensation losses
- Eliminates the erosion of steam turbine blades due to absence of moisture.
- Increase overall cycle efficiency

Types of superheaters :

Superheaters are classified as

1. Convection superheater
2. Radiant superheater, and
3. Combined type superheater.

Convection superheaters are located away from the furnace and exposed only to the hot gases. They are not in contact with the flame and the heat of hot gases is transferred to surface of superheater tubes by convection.

Radiant superheaters are located on the furnace walls and heat from furnace walls is transferred to superheater tubes by the radiation.

Combined type superheater receives heat by convection as well as radiation.

2.11.4 Air Preheater :

Steamboilers are provided with air preheater to increase the temperature of air used for combustion. In air preheater, the exit gases heat the incoming air for combustion and it saves the fuel. It is located between the economiser and chimney.

Air preheaters are classified into two types

1. Recuperative air preheater, and
2. Regenerative air preheater

2.11.5 Steam Trap :

The function of steam trap is to drain away the condensed steam from steam pipes. It does not allow the steam to escape with the condensate water. The bucket type steam trap is most common (Fig. 2.18).

It consists a outer vessel with sufficient water to float the bucket which carries valve rod. The water formed due to condensation of steam in the pipe line flows into the trap due to gravity.

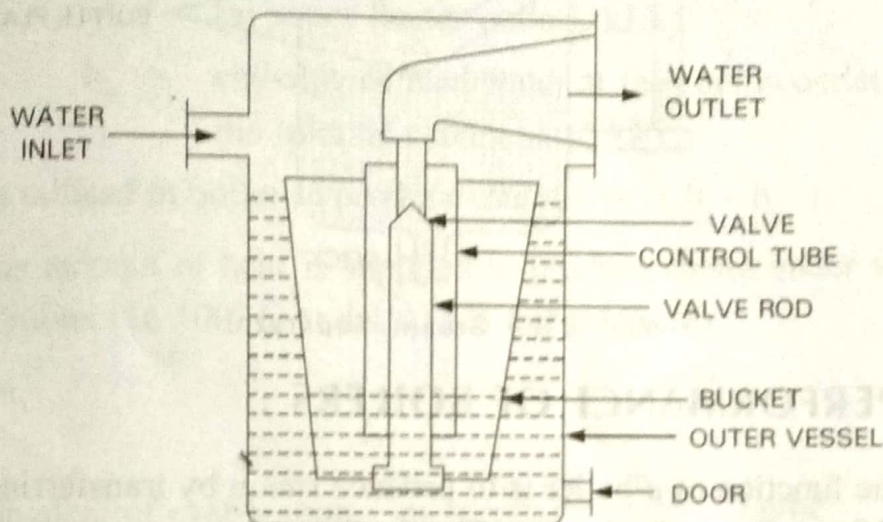


Fig. 2.18 Steam Trap

At normal condition valve is closed. When sufficient water is collected, the water level in the vessel rises and over flows into the bucket. Due to increased weight bucket sinks down the valve is open. The water is forced up through the control tube. The door provided at the bottom of the trap is used to remove sediments and dust deposited periodically.

2.11.6 Steam Separator :

The use of wet steam in steam engine and steam turbine produces many disadvantages. A steam separator is placed near to the engine on the main steam pipe line leading from the boiler to engine. The function of steam separator is to remove the water particles carried along with the steam on its way from the boiler to engine.

A steam separator is shown in Fig. 2.19. The steam enters at inlet and suddenly strikes the baffle plates and gets deflected upwards and passing in the direction of arrows, steam leaves the separator at outlet. Due to heavier weight water particles thrown down into the bottom and water collected at the bottom is discharged through cock. A glass tube is fitted to show the level of water in the separator.

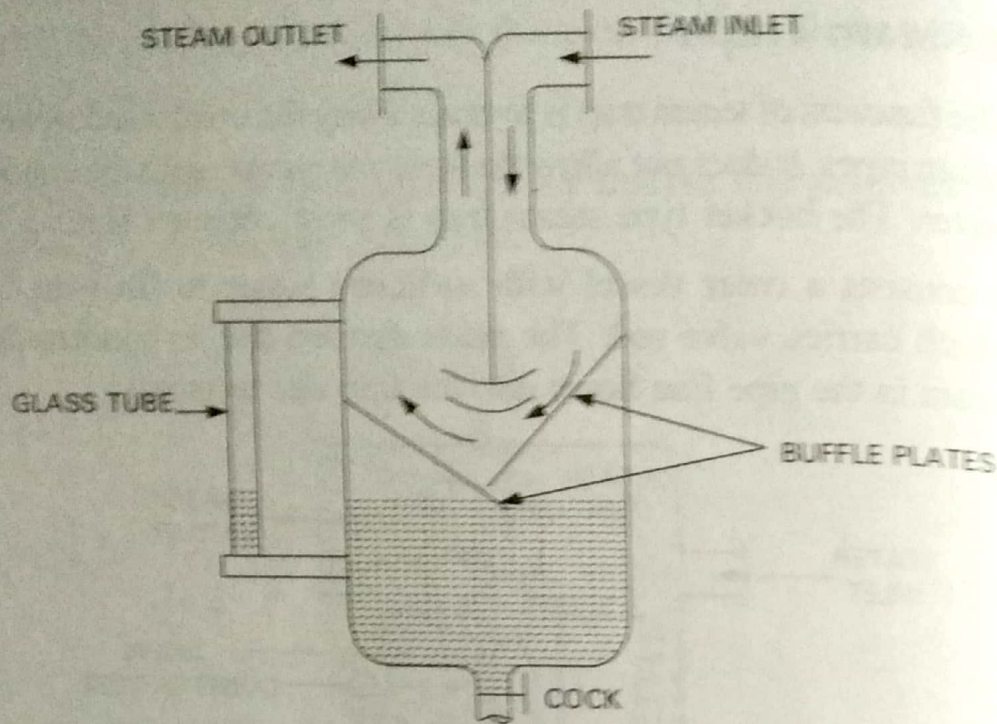


Fig. 2.19 Steam Separator

2.12 PERFORMANCE OF BOILERS :

The function of a boiler is to produce steam by transferring the heat of flue gases to feed water. The quantity of steam generated per hour is known as *evaporating capacity* of a boiler. The evaporating capacity and amount of heat supplied are considered in dealing with the performance of a steam boiler. The heat liberated may also be utilised in economiser, air preheater and superheater and some of the energy tends to leave with the flue gases through the chimney. Therefore it is necessary to draw up heat balance sheet for a boiler to know the heat distribution in boiler.

The two terms, the equivalent of evaporation and efficiency are used to measure the performance of a boiler. The analysis of data obtained on boiler trial will be discussed below.

2.13 EQUIVALENT EVAPORATION :

Different boilers generate steam under different conditions. For comparison of boilers it is necessary to assume that the feed water is evaporated in all boilers under the following conditions.

- (i) feed water supplied at 100°C and atmospheric pressure (1.01325 bar)
- (ii) steam is evaporated from and at 100°C and is just dry and saturated.

The mass of steam generated under above conditions is referred as equivalent evaporation.

Thus *equivalent evaporation* may be defined as the mass of water that would be evaporated from and at 100°C and atmospheric pressure (1.01325 bar) to form dry saturated steam by the same amount of heat which produces the steam under actual conditions.

let m_s = mass of steam produced at given conditions, kg/h

h_2 = enthalpy of steam produced at the given condition [at super heater outlet], kJ/kg

h_{f1} = enthalpy of feed water at feed pump outlet (i.e., at the inlet of economiser) kJ/kg

Heat utilised in boiler to produce steam = $m_s (h_2 - h_{f1})$

If this amount of heat is supplied to produce steam under standard conditions (i.e 100°C and 1.01325 bar pressure)

Then,

$$\text{Equivalent of evaporation, } m_e = \frac{m_s (h_2 - h_{f1})}{h_{fg,100}} \text{ kg/hr} \quad \dots (i)$$

where $h_{fg,100}$ = Latent heat at 100°C

$$= 2256.9 \text{ kJ/kg}$$

$$\therefore m_e = \frac{m_s (h_2 - h_{f1})}{2256.9} \text{ kJ/hr}$$

Some times equivalent evaporation is expressed in kg/kg of fuel.

If m = mass of steam generated per kg of fuel

$$\text{Then, } m_e = \frac{m(h_2 - h_{f1})}{2256.9} \text{ kg/kg of fuel}$$

The term $\left(\frac{h_2 - h_{f1}}{h_{fg,100}} \right)$ is called *factor of evaporation*.

Therefore,

$$\text{Equivalent evaporation} = m_s \times \text{factory of evaporation.}$$

$$\text{Then } \eta_{\text{boiler}} = \frac{m(h_2 - h_{f1})}{CV}$$

The percentage of heat utilised by feed water to form into steam is given by the equation

$$\text{Percentage heat utilised} = \frac{m(h_2 - h_{f1})}{CV} \times 100$$

From equations (i) and (ii)

$$m_e = \eta_{\text{boiler}} \left[\frac{m_f \times CV}{h_{fg,100}} \right]$$

Thus, equivalent evaporation is just an other way of expressing boiler efficiency. For boilers using the same fuel, a higher boiler efficiency means higher equivalent evaporation.

Factors influencing Boiler efficiency :

Boiler efficiency depends on the following factors

1. Fixed factors

- (i) Boiler design
- (ii) Heat recovery devices
- (iii) Built in losses
- (iv) Grate area
- (v) Characteristic of fuel burnt

2. Variable factors

- (i) Actual rate of fire and fuel characteristics as it is fired
- (ii) Excess air fluctuations
- (iii) Incomplete combustion
- (iv) Changes in draught due to atmospheric conditions
- (v) Humidity and temperature of the combustion air

Losses in Boilers :

The heat energy of the flue gas is lost as it leaves the chimney. This loss is known as chimney or stack loss (heat in the exhaust gases). Other minor losses in the boiler are due to

- incomplete combustion of fuel
- pressure of moisture in the fuel
- heat in the ash
- radiation and convection etc.

2.15 BOILER H.P. :

The capacity to evaporate 15.653 kg of water per hour from and at 100° C is called one boiler H.P.

$$\text{Boiler HP} = \frac{m_s (h_2 - h_{f1})}{2256.9 \times 15.653}$$

Example 2.1 :

A boiler producing steam which is 0.95 dry at the rate of 5 kg/s. The water entering the boiler has a temperature of 50°C and specific heat capacity 4.2 kJ/kg K. The steam leaves the boiler with a pressure of 20 bar. Assuming there are no losses, determine the energy required from the fuel.

Solution :

Rate of steam generated, $m_s = 5 \text{ kg/s.}$

Feed water temperature, $= 50^\circ \text{C}$

The energy required from fuel $= m_s (h_2 - h_{f1})$

Where $h_2 =$ enthalpy of steam generated

$h_{f1} =$ enthalpy of feed water

Enthalpy of steam,

$$\begin{aligned} h_2 &= h_{f2} + x_2 h_{fg2}, \text{ at 20 bar, } h_f = 908.6 \text{ kJ/kg,} \\ & \quad h_{fg} = 1888.7 \text{ kJ/kg} \\ &= 908.6 + 0.95 (1888.7) = 2702.86 \text{ kJ/kg} \end{aligned}$$

Enthalpy of feed water, $h_{f1} = m.c. dt = 1 \times 4.2 (50 - 0)$
 $= 210 \text{ kJ/kg}$

$$\begin{aligned} \therefore \text{Energy required from fuel} &= 5 (2702.86 - 210) \\ &= 12464.3 \text{ kJ/kg Ans.} \end{aligned}$$

Example 2.2 :

A boiler working at a pressure of 1.4 MN/m^2 evaporates 9 kg of water per kg of fuel burnt from feed water supplied at 45°C . The steam generated is 0.95 dry. Find the equivalent evaporation from and at 100°C .

Solution :

Mass of steam generated, $m = 9 \text{ kg/kg fuel}$

Pressure, $p = 1.4 \text{ MN/m}^2 = 14 \text{ bar}$

Feed water temperature, $= 45^\circ\text{C}$

At 14 bar, $h_{f2} = 830.1 \text{ kJ/kg}$, $h_{fg2} = 1957.7 \text{ kJ/kg}$

$$\therefore h_2 = h_{f2} + x_2 h_{fg2} = 830.1 + 0.95 (1957.7) = 2689.9 \text{ kJ/kg}$$

At 45°C , $h_{f1} = 188.36 \text{ kJ/kg}$

Equivalent evaporation

$$m_e = \frac{m(h_2 - h_{f1})}{h_{fg,100}} = \frac{9(2689.9 - 188.36)}{2256.9}$$

$$= 9.975 \text{ kg/kg of fuel Ans.}$$

Example 2.3 :

A generating station in a 3 hours have the following results.

Power generated $= 80,000 \text{ kWh}$

Coal used $= 40 \text{ tonnes}$

CV of coal $= 33000 \text{ kJ/kg}$

Calculate overall efficiency of the boiler

Solution :

Output in 3 hours $= 80,000 \text{ kWh} = 80,000 \times 3600 \text{ kJ}$

Input $= m_f \times \text{CV} = (40 \times 1000) 33,000$

$$\text{Overall } \eta = \left[\frac{\text{output}}{\text{input}} \right] \times 100$$

$$= \left[\frac{80,000 \times 3600}{(40 \times 1000) \times 33000} \right] \times 100$$

$$= 21.82 \% \text{ Ans.}$$

Example 2.4 :

A boiler plant supplies 5400 kg of steam per hour at 7.5 bar abs and 0.98 dry from feed water at 41°C . The boiler uses 670 kg of coal per hour having calorific value of 31000 kJ/kg

- Determine (i) The thermal efficiency of boiler
(ii) The equivalent evaporation.

Solution :

Mass of steam, $m_s = 5400 \text{ kg/hr}$

Pressure, $p = 7.5 \text{ bar}$

Feed water temperature = 41°C

Mass of coal, $m_f = 670 \text{ kg/hr}$;

Calorific value = 31000 kJ/kg

From steam tables,

At 7.5 bar, $h_{f2} = 709.3 \text{ kJ/kg}$ $h_{g2} = 2055.55 \text{ kJ/kg}$

at 41°C , $h_{f1} = 173.9 \text{ kJ/kg}$

(i) Boiler thermal efficiency :

$$\eta_{\text{boiler}} = \frac{m_s (h_2 - h_{f1})}{m_f \times \text{CV}}$$

Where $h_2 = h_{f2} + x_2 h_{g2}$

$$= 709.3 + 0.98(2055.55) = 2723.74 \text{ kJ/kg.}$$

$$\therefore \eta_{\text{boiler}} = \frac{5400 (2723.74 - 173.9)}{670 \times 31000}$$

$$= 0.663 = 66.3 \% \text{ Ans.}$$

(ii) Equivalent evaporation :

$$m_e = \frac{m_s (h_2 - h_{f1})}{h_{fg,100}}$$

$$= \frac{5400 (2723.74 - 173.9)}{2256.9}$$

$$= 6100.9 \text{ kg/hr from and at } 100^{\circ}\text{C} \text{ Ans.}$$

Note : Equivalent evaporation can also be expressed kJ/kg of coal

$$\text{If } m = \frac{5400}{670} = 8.0597 \text{ kg/kg of coal}$$

$$\begin{aligned} \therefore m_e &= \frac{8.0597(2723.74 - 173.9)}{2256.9} \\ &= 9.106 \text{ kg/kg. of coal Ans.} \end{aligned}$$

Example 2.5 :

A boiler produces steam at 10 bar and 300°C at the rate of 5 tons/hour from water at 30°C . Calculate the boiler h.p and equivalent evaporation.

Solution :

$$\begin{aligned} \text{Mass of steam, } m_s &= 5 \text{ tons/hr} \\ &= 5000 \text{ kg/hr} \end{aligned}$$

$$\text{Pressure } p = 10 \text{ bar}$$

$$\text{Temperature of steam } t = 300^\circ \text{C}$$

$$\text{Feed water temperature, } = 30^\circ \text{C}$$

From steam tables,

$$\text{At } 10 \text{ bar } 300^\circ \text{C, } H_2 = 3052.1 \text{ kJ/kg}$$

$$\text{at } 30^\circ \text{C } h_{f1} = 125.7 \text{ kJ/kg}$$

$$\text{Boiler H.P} = \frac{m_s (h_2 - h_{f1})}{h_{fg,100} \times 15.563}$$

$$= \frac{5000 (3052.1 - 125.7)}{2256.9 \times 15.563}$$

$$= 416.57 \text{ Ans.}$$

Equivalent evaporation,

$$m_e = \frac{5000 (3052.1 - 125.7)}{2256.9}$$

$$= 6843.2 \text{ kg/hour from and at } 100^\circ \text{C Ans.}$$

Example 2.6 :

During a test on a boiler the following data were collected.

Steam pressure : 9.487 bar gauge

Steam condition : Dry and saturated

Feed water temperature : 35°C

Rate of evaporation : 12 kg/kg of coal

Calorific value : 37.5×10^3 kJ/kg

Determine :

(a) Thermal efficiency

(b) Factor of evaporation

(c) Equivalent evaporation

Solution :

Pressure, $p = 9.487$ bar gauge

$$= 9.487 + 1.013 = 10.5 \text{ bar abs}$$

At 10.5 bar the enthalpy of dry and saturated steam,

$$h_2 = 2776.2 \text{ kJ/kg}$$

and at 35°C, $h_{f1} = 148$ kJ/kg

(a) Thermal efficiency :

$$\eta_{\text{boiler}} = \frac{m(h_2 - h_{f1})}{CV} = \frac{12(2776.2 - 148)}{37.5 \times 10^3} = 0.841$$

$$= 84.1\% \text{ Ans.}$$

(b) Factor of evaporation :

$$\text{Factor of evaporation} = \frac{(h_2 - h_{f1})}{h_{fg,100}} = \frac{2776.2 - 148}{2256.9}$$

$$= 1.164 \text{ Ans.}$$

(c) Equivalent evaporation :

$$m_e = \frac{m(h_2 - h_{f1})}{h_{fg,100}} = \frac{12(2776.2 - 148)}{2256.9}$$

$$= 13.97 \text{ kg/kg of coal Ans.}$$

(Chapter-2)

Example 2.7 :

A boiler generates 17250 kg/hr of steam at 20 bar and 0.95 dry from feed water at 44°C. The steam leaves the superheater at 325°C. If the efficiency of the unit is 75%, determine the fuel consumption (CV = 4.25×10^3 kJ/kg) and heat transfer in the superheater per kg of steam.

Solution :

Mass of steam, $m_s = 17250$ kg/hr

Temperature of feed water = 44°C

At 20 bar, $h_f = 908.6$ kJ/kg;

$h_{fg} = 1888.7$ kJ/kg

and at 20 bar and 325°C, $h_{\text{sup}} = 3081.8$ kJ/kg

Enthalpy of water corresponding to 44°C = 184.2 kJ/kg

$$\text{Efficiency} = \frac{m_s(h_2 - h_{f1})}{m_f \times CV}$$

$$\therefore \text{Mass of fuel, } m_f = \frac{17250(3081.8 - 184.2)}{0.75 \times 42.5 \times 10^3}$$

$$= 1568.1 \text{ kg/hr Ans.}$$

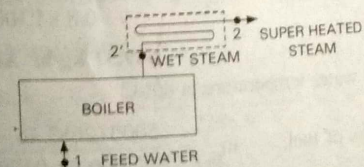


Fig. 2.21

Heat transferred in the super heater per kg, of steam

$$= (h_2 - h_{\text{wet}_2})$$

$$= [3081.8 - (908.6 + 0.95 \times 1888.7)]$$

$$= 378.9 \text{ kJ/hr}$$

(Chapter-2)

Example 2.8 :

A boiler produces 5 tons of steam per hour at 12 bar abs and 0.95 dry from feed water at 35°C. If the boiler efficiency is 80% and if the fuel used has a calorific value of 23000 kJ/kg, Determine

(i) The fuel required per hour

(ii) The percentage saving in fuel if feed water temperature is raised to 80°C by using economiser.

Solution :

Mass of steam, $m_s = 5000 \text{ kg/hr}$

Calorific value of fuel, CV = 23000 kJ/kg

Boiler efficiency = 80%

From steam tables

at 12 bar, $h_{f2} = 798.4 \text{ kJ/kg}$,

$h_{g2} = 1984.3 \text{ kJ/kg}$

$\therefore h_2 = 798.4 + 0.95(1984.3)$
 $= 2683.48 \text{ kJ/kg}$

at 35°C, $h_{f1} = 146.6 \text{ kJ/kg}$

80°C, $h_{f1} = 334.9 \text{ kJ/kg}$

$$\eta_{\text{boiler}} = \frac{m_s (h_2 - h_{f1})}{m_f \times \text{CV}}$$

$$\therefore \text{Mass of fuel, } m_f = \frac{5000 (2683.48 - 146.6)}{0.8 \times 23000}$$

$$= 689.37 \text{ kg/hr Ans.}$$

If feed water temperature is 80°C

$$\text{Mass of fuel, } m_f = \frac{5000 (2683.48 - 334.9)}{0.8 \times 23000}$$

$$= 638.2 \text{ kg/hr}$$

$$\text{Percentage saving in fuel} = \left(\frac{689.37 - 638.2}{689.37} \right) \times 100$$

$$= 7.423\% \text{ Ans.}$$

Example 2.9 :

The following observations were made on a boiler plant during one hour test :

Steam pressure = 20 bar

Steam temperature = 260°C

Steam generated = 37500 kg

Temperature of water entering the economiser = 15°C

Temperature of water leaving the economiser = 90°C

Fuel used = 4400 kg

Energy of combustion of fuel = 30000 kJ/kg

Calculate

(a) Equivalent evaporation per kg of fuel

(b) Thermal efficiency of the plant

(c) The percentage heat energy of the fuel utilised by the economiser.

Solution :

At 20 bar and 260°C,

$h_{\text{sup}} = 2897 \text{ kJ/kg}$

At 15°C, $h_{f1} = 62.9 \text{ kJ/kg}$ and at 90°C $h_{f1}' = 376.9 \text{ kJ/kg}$

Mass of steam generated per kg of fuel,

$$m = \frac{37500}{4400} = 8.52 \text{ kg/kg of fuel}$$

(a) Equivalent evaporation,

$$m_e = \frac{m(h_2 - h_{f1})}{2256.9} = \frac{8.52(2897 - 62.9)}{2256.9}$$

$$= 10.7 \text{ kg/kg of fuel Ans.}$$

(b) Thermal efficiency of the plant :

$$\eta = \frac{m(h_2 - h_{f1})}{\text{CV}} = \frac{8.52(2897 - 62.9)}{3000} \times 100$$

$$= 80.5\% \text{ Ans.}$$

(c) Percentage of energy utilised in economiser :

$$\begin{aligned}\text{Heat utilised in the economiser} &= m(h_f - h_n) \\ &= 8.52(376.9 - 62.9) \\ &= 2675 \text{ kJ/kg of fuel}\end{aligned}$$

% of heat utilised by the economiser

$$\begin{aligned}&= \frac{2675}{30,000} \times 100 \\ &= 8.9\% \text{ Ans.}\end{aligned}$$

2.16 HEAT BALANCE :

In order to locate the source of heat losses, boiler trials are conducted for a certain length of time and the data is recorded. Then the calculations are made to prepare the heat balance sheet.

The following calculations are made for heat balance

$$\text{Heat supplied in fuel} = m_f \times C.V$$

where m_f = mass of fuel, kg

$$C.V = \text{calorific value of fuel, kJ/kg}$$

Heat distribution :

1. Heat utilised by feed water to form steam

$$= m(h_2 - h_n) ; m = \text{mass of steam generated/ kg of fuel}$$

2. Heat lost through exhaust gases

$$= m_g \cdot C_p \cdot (t_g - t_f) ; m_g = \text{mass of exhaust gases per kg of coal}$$

t_g = temperature of flue gases.

3. Heat lost due to unburnt fuel

$$= m_{uf} \times C.V ; m_{uf} = \text{mass of unburnt fuel per kg of fuel}$$

$C.V$ = calorific value of unburnt coal

4. Heat lost to moisture = $m_1(h'_2 - h'_f)$

where h'_2 = enthalpy of steam formed, kJ/kg

h'_f = enthalpy of water

m_1 = mass of moisture per kg of coal

5. Heat lost to radiation, etc. (By difference)

(Chapter-2)

Example 2.10 :

In a boiler trial the following data were recorded :

$$\text{Steam produced per hour} = 2710 \text{ kg}$$

$$\text{Feed water temperature} = 40^\circ \text{C}$$

$$\text{Boiler pressure} = 11.5 \text{ bar}$$

$$\text{Dryness fraction} = 0.9$$

$$\text{Coal used per hour} = 330 \text{ kg}$$

$$\text{Calorific value of coal} = 29120 \text{ kJ/kg}$$

$$\text{Mass of flue gases} = 10 \text{ kg/kg of coal}$$

$$\text{Specific heat of flue gases} = 1.05 \text{ kJ/kg K}$$

$$\text{Temperature of flue gases} = 350^\circ \text{C}$$

$$\text{Boiler room temperature} = 30^\circ \text{C}$$

- Determine (i) Equivalent evaporation from and at 100°C
and (ii) Draw up heat balance per kg of coal

Solution :

$$\begin{aligned}\text{Mass of steam generated/kg of coal, } m &= \frac{2710}{330} \\ &= 8.212 \text{ kg/kg of coal}\end{aligned}$$

$$\text{At } 11.5 \text{ bar, } h_{f2} = 789.9 \text{ kJ/kg ; } h_{g2} = 1991.4 \text{ kJ/kg}$$

$$\begin{aligned}\therefore h_2 &= h_{f2} + x_2 h_{g2} \\ &= 789.9 + 0.9(1991.4) \\ &= 2582.16 \text{ kJ/kg}\end{aligned}$$

$$\text{at } 40^\circ \text{C, } h_{f1} = 167.5 \text{ kJ/kg}$$

$$\begin{aligned}\text{Equivalent evaporation } m_e &= \frac{m(h_2 - h_{f1})}{h_{fg,100}} \\ &= \frac{8.212(2582.16 - 167.5)}{2256.9} \\ &= 8.786 \text{ kg/kg of coal Ans.}\end{aligned}$$

$$\begin{aligned}\text{Heat supplied per kg of coal} &= m_f \times C.V \\ &= 1 \times 29120 \\ &= 29120 \text{ kJ/kg of coal}\end{aligned}$$

(Chapter-2)

2.42 Heat Power Engineering-II

$$\begin{aligned}\text{Heat utilised to form steam} &= m (h_2 - h_{f1}) \\ &= 8.212 (2582.16 - 167.5) \\ &= 19829.2 \text{ kJ/kg of coal} \\ \text{Heat carried away by flue gases} &= m_g \times C_p (t_g - t_r) \\ &= 10 \times 1.05 (350 - 30) \\ &= 3360 \text{ kJ/kg of coal} \\ \text{Heat lost by radiation etc.} &= 29120 - (19829.2 + 3360) \\ &= 5930.8 \text{ kJ/kg coal}\end{aligned}$$

Heat balance sheet per kg of coal :

Heat supplied	kJ/kg of coal	Percentage	Heat distribution	kJ/kg of coal	Percentage
Heat supplied	29120	100	Heat in steam	19829.2	$\frac{19829.2}{29120} \times 100 = 68.09$
			Heat to exhaust gases	3360	$\frac{3360}{29120} \times 100 = 11.54$
			Heat to radiation etc.,	5930.8	$\frac{5930.8}{29120} \times 100 = 20.37$
Total	29120	100		29120	100

Example 2.11 :

The following observations were made during a boiler trial :

$$\begin{aligned}\text{Mass of feed water per hour} &= 630 \text{ kg} \\ \text{Temperature of feed water} &= 65^\circ \text{C} \\ \text{Steam pressure} &= 9.487 \text{ (gauge)} \\ \text{Oil fired per hour} &= 50 \text{ kg} \\ \text{Calorific value} &= 45250 \text{ kJ/kg} \\ \text{Quality of steam} &= 0.97 \text{ dry}\end{aligned}$$

Percentage composition of oil by mass;

$$C = 84.75, H = 13.00, S = 1.25$$

(Chapter-2)

Steam Boilers 2.43

$$\begin{aligned}\text{Mass of dry flue gases per kg of oil} &= 17.05 \text{ kg} \\ \text{Temperature of gases leaving the boiler} &= 350^\circ \text{C} \\ \text{Specific heat of dry flue gases} &= 1.05 \text{ kJ/kg K} \\ \text{Boiler room temperature} &= 25^\circ \text{C} \\ \text{Specific heat of superheated steam} &= 2.1 \text{ kJ/kg K} \\ \text{Partial pressure of steam (vapour) in flue gases} &= 0.08 \text{ bar} \\ \text{Draw up complete heat balance sheet and calculate the boiler efficiency and equivalent evaporation per kg of oil.}\end{aligned}$$

Solution :

$$\text{Steam produced per kg of oil, } m = \frac{630}{50} = 12.6 \text{ kg}$$

$$\text{Quality of steam } x = 0.97$$

$$\begin{aligned}\text{Pressure (gauge) } p &= 9.487 \text{ gauge} \\ &= 9.487 + 1.013 \\ &= 10.5 \text{ bar abs}\end{aligned}$$

$$\text{Feed water temperature, } = 65^\circ \text{C}$$

From steam table,

$$\text{At } 10.5 \text{ bar, } h_{f2} = 772 \text{ kJ/kg ; } h_{fg2} = 2006 \text{ kJ/kg}$$

$$\begin{aligned}h_2 &= h_{f2} + x_2 h_{fg2} \\ &= 772 + 0.97 (2006) = 2717.82 \text{ kJ/kg}\end{aligned}$$

$$\text{at } 65^\circ \text{C, } h_{f1} = 272 \text{ kJ/kg}$$

$$\text{Heat supplied per kg of oil} = 45250 \text{ kJ}$$

$$\begin{aligned}\text{Moisture formed per kg of oil, } m_1 &= 9 H_2 \\ &= 9 \times 0.13 = 1.17 \text{ kg}\end{aligned}$$

$$\begin{aligned}\text{Heat to form steam} &= m (h_2 - h_{f1}) \\ &= 12.6 (2717.82 - 272) \\ &= 30817.332 \text{ kJ/kg of coal}\end{aligned}$$

(Chapter-2)

Heat carried away by dry flue gas

$$\begin{aligned}
 &= m_g C_p (t_g - t_f) \\
 &= 17.05 \times 1.05 (350 - 25) \\
 &= 5541.25 \text{ kJ/kg of coal}
 \end{aligned}$$

Heat in moisture = $m_w (h_g - h_f)$

At 0.08 bar, $h_g = 2577.5$ and $t_g = 41.54^\circ\text{C}$

at 25°C $h_f = 104.8 \text{ kJ/kg}$

$$\therefore \text{Heat in moisture} = m_w [h_g + C_p (t_{\text{sup}} - t_g) - h_f]$$

$$= 1.17 [2577.5 + 2.1 (350 - 41.54) - 104.8]$$

$$= 3650.94 \text{ kJ/kg of coal}$$

$$\begin{aligned} \text{Heat lost to radiation} &= 45250 - (30817.332 + 5541.25 + 3650.94) \\ &= 5240.48 \text{ kJ Ans.} \end{aligned}$$

Heat balance sheet :

Heat supplied	kJ/kg of oil	Percentage	Heat distributed	kJ/kg of coal	Percentage
Heat supplied per kg of oil	45250	100	(i) Heat to steam	30817.332	$\frac{30817.332}{45250} \times 100 = 68.1$
			(ii) Heat to dry flue gases	5541.25	$\frac{5541.25}{45250} \times 100 = 12.24$
			(iii) Heat in moisture	3650.94	$\frac{3650.94}{45250} \times 100 = 8.07$
			(iv) Heat radiation (etc.)	5240.48	$\frac{5145.84}{45250} \times 100 = 11.59$
Total	45250	100		45250	100

$$\begin{aligned} \text{Boiler efficiency} &= \frac{m(h_2 - h_{f1})}{\text{CV}} \\ &= \frac{12.6(2717.82 - 272)}{45252} \\ &= 0.681 = 68.1\% \text{ Ans.} \end{aligned}$$

$$\begin{aligned} \text{Equivalent evaporation per kg of oil} &= \frac{30817.332}{2256.9} \\ &= 13.65 \text{ kg/kg oil Ans.} \end{aligned}$$

(Chapter-2)

2.17 BOILER DRAUGHT :

The combustion process in a furnace can take place only when it receives steady flow of air and the combustion gases are constantly removed. This is achieved by the boiler draught system. The rate of fuel burning is also depends on the air flowing through the grate. Therefore, a pressure difference between the furnace and the surrounding should exist so that combustion air will flow into the furnace and the product of the combustion (hot gasses) will flow out of the furnace through the chimney. This pressure difference causing the air flow is called draught.

Terms related to boiler draught :

Draught - It is the difference between absolute gas pressure at any point in a gas flow passage and the ambient atmospheric pressure.

Draught is positive if $p_{\text{atm}} < p_{\text{gas}}$ and negative if $p_{\text{atm}} > p_{\text{gas}}$.

Draught differential - It is the pressure difference between two points in the gas flow path.

Draught loss - Pressure loss caused by friction between two points in the gas flow path is known as draught loss.

Functions of draught :

Functions of draught are to

- provide an adequate supply of air for complete combustion of fuel
- move the hot gases through the system
- exhaust the gases to atmosphere through the chimney.

2.18 CLASSIFICATION OF DRAUGHTS :

According to the principle or method employed, boiler draughts may be classified as shown below :

(Chapter-2)

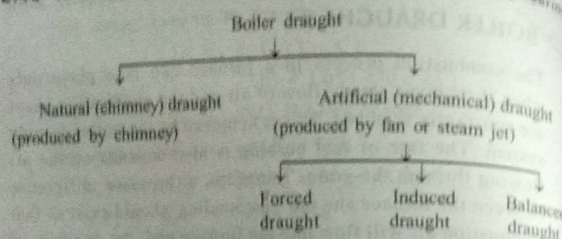


Fig. 2.22 TYPES OF DRAUGHTS

2.18.1 Natural Draught :

Natural draught is produced by the boiler chimney. A chimney creates a draught by a simple principle that at high temperature the density of gas is lower. Under this condition it may be displaced by a more dense gas.

The temperature of gases in the chimney is much higher than the temperature of the air outside. Therefore the density of hot gases inside the chimney is less than cold air outside. The pressure in the chimney is, therefore, less than the outside pressure. This pressure difference causes the flow of air into the furnace and pushes the hot gases through the chimney.

Natural draught is cheaper and simple to operate. But it has the following limitations.

1. Draught created depends on chimney height
2. Excess height of the chimney will make the cost high.
3. Sufficient draught cannot be produced; only suitable for small boilers.
4. Weather conditions will affect draught

2.18.2 Mechanical (Artificial) Draught :

Mechanical draught is produced by fan or blower. Steam jet can also be used to produce mechanical draught. The artificial draught has the following *advantages* :

- Does not depend on climatic conditions
- Draught can be easily controlled

- Higher rate of combustion
 - Inferior quality of fuel can be used.
- But it is costlier and complex to maintain.

Types of mechanical draught :

Mechanical draught produced by fan may be further classified into

1. Forced draught
2. Induced draught
3. Balanced draught

Forced draught :

In forced draught, a fan (FD fan) is located before the furnace i.e., near or at the base of the boiler. It discharges air under pressure into the furnace and on to the chimney. In forced draught furnace pressure (p_f) is higher than atmospheric pressure. This is used in small oil fired boilers. The principle of forced draught is shown in Fig. 2.23.

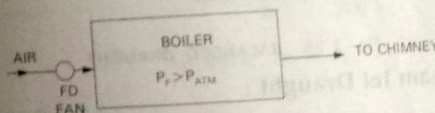


Fig. 2.23 FORCED DRAUGHT

Induced draught :

In induced draught, fan (ID fan) is located near the base of chimney i.e., after the furnace. It induces the vacuum in the chimney causing air to flow into the furnace. In induced draught furnace pressure is lower than atmospheric pressure. In this case air leakages can be high and this leads to reduction in efficiency and requires more power to drive the fan. Induced draught is used for the boilers provided with economiser and air preheater. The principle of induced draught is shown in Fig. 2.22

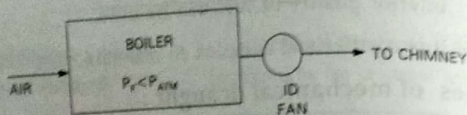


Fig. 2.24 INDUCED DRAUGHT

Balanced draught :

This is the combination of forced and induced draught. In this case furnace pressure will be nearly equal to atmospheric pressure. It uses two fans. FD fan is located before the furnace and used to supply the air for combustion. ID fan located after the furnace removes the gases from the furnace. Balanced draught is adopted in all power plants.

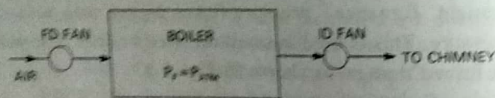


Fig. 2.25 BALANCED DRAUGHT

2.18.3 Steam Jet Draught :

In steam jet draught, the exhaust steam is used for producing draught. The forced draught is produced by placing the steam jet below the furnace grate. The air will be forced into the furnace by the steam jet.

In case of induced draught, the steam jet is placed below the chimney. It charges the surrounding air or gas creating the vacuum above the furnace which causes the air to flow into the furnace.

Board Diploma Examination Problem with Solutions**Example 2.12 :**

A boiler generates 6200 kg of steam per hour at 9 bar and 0.93 dry from feed water at 38°C, when using 825 kg of coal per hour having calorific value of 31000 kJ/kg. Determine the efficiency. [March/April 2006]

Solution :

At 9 bar and 0.93 dry,

$$h_2 = h_{f2} + x_2 h_{fg2}$$

$$= 742.6 + 0.93 (2029.5) = 2630.035 \text{ kJ/kg}$$

and at 38°C, $h_{f1} \approx 154.5 \text{ kJ/kg}$

$$\text{Efficiency of boiler} = \frac{m_s (h_2 - h_{f1})}{m_f \times CV}$$

$$= \frac{6200 (2630.035 - 154.5)}{825 \times 31000} = 0.6$$

$$= 60\% \text{ Ans.}$$

Example 2.13 :

In a boiler test, steam at a pressure of 14 bar, having a dryness fraction 0.9, is generated at the rate of 8 kg per kg of coal burnt. The calorific value of coal fired is 35,000 kJ/kg and temperature of feed water is 45°C, calculate the thermal efficiency of the boiler. [Oct./Nov. 2008]

Solution :

$$\text{At 14 bar, } h_{f2} = 830.1 \text{ kJ/kg, } h_{fg2} = 1957.7 \text{ kJ/kg}$$

$$h_2 = h_{f2} + x_2 h_{fg2}$$

$$= 830.1 + 0.9 (1957.7) = 2592.03 \text{ kJ/kg}$$

$$\text{at } 45^\circ \text{C, } h_{f1} = 189.7 \text{ kJ/kg}$$

Thermal efficiency of the boiler,

$$\eta_{\text{boiler}} = \frac{m(h_2 - h_{f1})}{CV} = \frac{8(2592.03 - 189.7)}{35000} \times 100$$

$$= 54.91\% \text{ Ans.}$$

Example 2.14 :

A boiler generates 18,000 kg/hr of steam at 12 bar with 95% quality. Feed water temperature is 110°C . Rate of coal firing is 2000 kg/hr. If H.C.V of coal is 27,500 kJ/kg, find

(a) Factor of evaporation.

(b) The equivalent evaporation.

(c) The thermal efficiency of generation. [Oct./Nov. 2007]

Solution :

$$\text{Mass of steam generated per kg of coal, } m = \frac{18,000}{2000} = 9 \text{ kg/kg of coal}$$

At 12 bar, 0.95 dry (from steam tables),

$$h_2 = h_{f2} + x_2 h_{fg}$$

$$= 798.4 + 0.95(1984.3) = 2683.48 \text{ kJ/kg}$$

h_{f1} corresponding to 110°C

$$= 460.6 \text{ kJ/kg}$$

(a) Factor of evaporation :

$$\text{Factor of evaporation} = \frac{h_2 - h_{f1}}{h_{f100}}$$

$$= \frac{2683.48 - 460.6}{2256.9} = 0.985 \text{ Ans.}$$

(b) Equivalent evaporation,

$$m_e = \frac{m(h_2 - h_{f1})}{h_{f100}} = \frac{9(2683.48 - 460.6)}{2256.9}$$

$$= 8.865 \text{ kg/kg of coal.}$$

(c) Thermal efficiency of generation :

$$\eta_{\text{boiler}} = \frac{m(h_2 - h_{f1})}{CV} = \frac{9(2683.48 - 460.6)}{27500} \times 100$$

$$= 72.7\% \text{ Ans.}$$

Example 2.15 :

The following data were obtained during a trial on a steam boiler, coal fired with natural drought :

Trial time = 90 minutes

Mass of feed water supplied = 6750 kg

Steam pressure = 10 bar (abs)

Temp. of feed water = 70°C

Quality of steam = 0.95

Coal burnt = 600 kg

Moisture of fuel = 4.25% by mass

Higher calorific value of coal = 39480 kJ/kg

Mass of flue gases = 280°C

Boiler room temp. = 30°C

Composition of coal - C = 89%, H_2 = 3%, N_2 = 4%, Ash = 4%

Heat in vapour = 3000 kJ/kg

Draw up the heat balance sheet.

[March/April 2008]

Solution :

$$\text{Steam produced per kg of coal} = \frac{6750}{600} = 11.25 \text{ kg}$$

Quality of steam, $x = 0.95$

Feed water temperature = 70°C

At 10 bar, $h_{f2} = 762.6 \text{ kJ/kg}$, $h_{fg2} = 2013.6 \text{ kJ/kg}$

$$\therefore h_2 = h_{f2} + x h_{fg2} = 762.6 + 0.95(2013.6) = 2675.52 \text{ kJ/kg}$$

At 70°C , $h_{f1} = 294 \text{ kJ/kg}$

Moisture content in 1 kg of coal = 0.0425 kg

$$\therefore \text{Heat supplied per kg of coal}$$

$$= (1 - 0.0425) 39480$$

$$= 37802 \text{ kJ/kg}$$

Heat distribution :

$$\begin{aligned}\text{Heat to form steam} &= m(h_g - h_f) \\ &= 11.25 (2675.52 - 294) \\ &= 26792.1 \text{ kJ/kg of coal}\end{aligned}$$

Heat carried away by exhaust gases

$$\begin{aligned}&= m_g C_p (t_g - t_c) ; \text{ Assume } C_p = 1.05 \text{ kJ/kg} \\ &= 18.75 \times 1.05 (280 - 30) \\ &= 4921.87 \text{ kJ/kg of coal}\end{aligned}$$

Heat in moisture = Mass of moisture $\times$ 3000

$$\begin{aligned}&= (9 H_2 + 0.0425) \times 3000 \\ &= (9 \times 0.03 + 0.0425) \times 3000 \\ &= 937.5 \text{ kJ/kg of coal}\end{aligned}$$

$$\begin{aligned}\text{Heat lost to radiation} &= 37802 - (26792.1 + 4921.87 + 937.5) \\ (\text{by difference}) &= 5150.53 \text{ kJ/kg of coal.}\end{aligned}$$

Heat balance sheet per kg of coal :

Heat supplied	kJ/kg of coal	Per-centage	Heat distribution	kJ/kg of coal	Percentage
Heat supplied in fuel	37802	100	Heat in steam	26792.1	$\frac{26792.1}{37802} \times 100 = 70.87$
			Heat to exhaust gases	4921.87	$\frac{4921.87}{37802} \times 100 = 13.02$
			Heat in moisture	937.5	$\frac{937.5}{37802} \times 100 = 2.48$
			Heat to radiation	5150.53	$\frac{5150.53}{37802} \times 100 = 13.63$
	37802	100		37802	100

SUMMARY

Boiler is a closed steel shell in which steam is generated.

➤ In *fire tube boiler*, the hot gases pass through the tubes surrounded by water.

➤ In *water tube boiler*, water is made to pass through the tubes over which the hot gases flow.

➤ The main characteristics of high pressure boilers are Forced circulation is employed when the pressure exceeds 160 bar.

Size of drum is very small and in certain cases, it is completely eliminated.

Improved method of heating by providing large heating devices.

➤ Devices fitted on boiler for safety and proper operation are called mountings.

➤ Devices installed with the boiler for proper operation and to increase the efficiency are called accessories.

➤ Equivalent Evaporation,

$$m_e = m_s \left[\frac{h_g - h_f}{h_{fg,100}} \right]$$

The term $\left[\frac{h_g - h_f}{h_{fg,100}} \right]$ is called factor of evaporation.

$$m_e = m_s \times \text{factor of evaporation.}$$

➤ Thermal efficiency of boiler

$$h_{th} = \frac{m_s (h_g - h_f)}{m_f \times CV}$$

$$\text{Boiler HP} = \frac{m_s (h_g - h_f)}{2256.9 \times 15.563}$$

where h_g = Enthalpy of steam produced kJ/kg

h_f = Enthalpy of feed water, kJ/kg

m_s = mass of steam generated, kg/hr

m_f = mass of fuel burnt, kg/hr

CV = calorific value of fuel, kJ/kg.

CONDENSERS :

In steam power plant steam is generated at high pressure and temperature in a boiler and expanded in a prime mover to obtain work. After the expansion, heat has to be rejected at lower temperature as it will lead to a higher efficiency. This is achieved by condensing the steam in a closed vessel, called *condenser*. It is a major and very important equipment in a power plant. All modern power plants are of the condensing type, and are more efficient than non-condensing ones.

Definition :

Condenser is a device in which steam is condensed to water at a pressure less than atmosphere. Condensation is effected due to removal of latent heat from exhaust steam by using circulating cooling water. Thus a condenser is basically a heat exchanger, in which heat of the exhaust steam is transferred to the circulating water.

Function of condenser :

Condensers are used in steam power plant and in refrigeration system (to convert high pressure vapour refrigerant into liquid refrigerant).

The functions of condensers in steam power plant are :

- To decrease the exhaust pressure below the atmosphere
- To provide hot feed water to the boiler.

By use of condenser (i.e., by cooling the exhaust steam in a closed vessel to a lower temperature) the exhaust pressure can

(Chapter-5)

be maintained below atmosphere. This allows a higher pressure ratio for expansion. The efficiency as well as work output per kg of steam increases substantially. The extra work available in case of condensed engines and turbines is illustrated in Fig. 5.1.

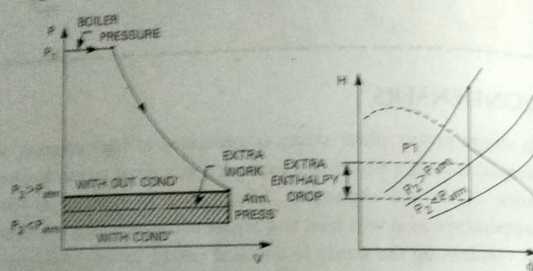


Fig. 5.1 Advantages Due to Condenser

Thus the advantages due to condensers are :

- Higher pressure ratio provides larger enthalpy drop.
- Work output per kg of steam increases. Thus the specific steam consumption decreases.
- Condensate can be reused as hot feed water to the boiler. As the feed water is hot, it reduces the time of evaporation and hence fuel economy.
- No feed water treatment is required. This reduces the cost of the plant.

5.2 ELEMENTS OF CONDENSING PLANT :

Steam condensers operate at vacuum pressure as low as 0.05 bar abs. Air leakage to condenser effect the performance. The air present in the condenser must be cooled to reduce the capacity of air pump. The following accessories are provided for effective operation of condensers.

1. An air pump to remove air and other non-condensable gases from the condenser.

2. Condensate pump to remove the condensate from the condenser.
3. Circulating water pump for circulating the cooling water.
4. A hot well to receive the condensate.
5. Cooling tower for cooling circulating water when there is only limited supply of water.

The layout of these accessories or elements are shown in Fig. 5.2.

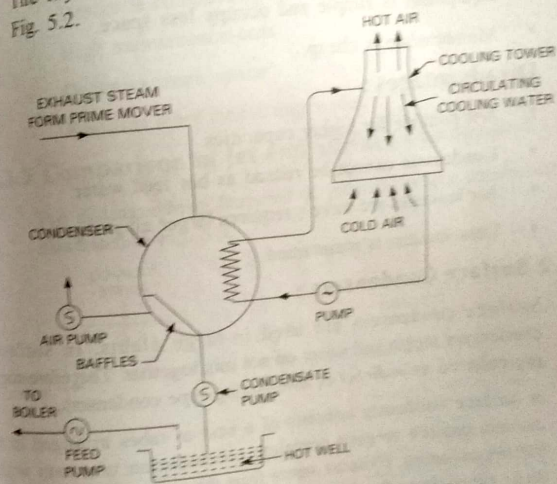


Fig. 5.2 Elements of a Steam Condensing Plant

5.3 CLASSIFICATION OF CONDENSERS :

Steam condensers may be classified into two categories.

1. Jet condensers, and
2. Surface condensers

5.3.1 Jet Condensers :

Jet condensers are used in small units where clean fresh water is available in plenty. In jet condensers water is in direct contact with exhaust steam. Therefore, jet condensers are also referred as direct contact type or mixed type condensers.

(Chapter-5)

In Jet condensers the steam is condensed by a spray of cold water injected into the condenser shell and there is a complete mixing of steam and water. The condensate (hot water) gravitates to the extraction pump and is removed from the condenser.

Advantages of Jet Condensers :

- As a result of effective mixing, it requires less circulating cooling water
- Equipment is simple and occupy less space.
- Maintenance is cheap.

Disadvantages :

- Not suitable for higher capacities
- Condensate cannot be reused as hot feed water
- Air leakages are more ; requires larger air pump
- Less vacuum is maintained.

5.3.2 Surface Condensers :

Surface condensers are used in large plants. In surface condensers steam and water do not mix together. Therefore they are referred as *indirect* or *non-mixing* type condensers.

A surface condenser consists of a nest of tubes arranged in a certain pattern to give complete condensation of steam with minimum losses. Exhaust steam from the turbine is passed over these tubes through which cooling water is circulated. The steam is condensed by giving up its latent heat to cooling water circulated through the tubes. The cooling water is then passed to a cooling tower where it is cooled by a cold air and is again recirculated through condenser. The condensate is pumped into a hot well from where it is supplied to boiler through a feed pump.

Advantages of Surface Condenser :

- Used for large capacity plants
- High vacuum can be created.

Jet Condensers

- Condensate is free from impurities, and can be reused as hot feed water to the boiler.
- Any type of water can be used as cooling medium.
- Air leakage is comparatively less, hence less power is required to operate air pump.

Disadvantages :

- Design is complicated and costly
- High maintenance cost
- Occupies more space
- Requires more circulating water.

5.3.3 Comparison of Jet and Surface Condensers :

The distinguishing features of jet and surface condensers are illustrated in Fig. 5.3.

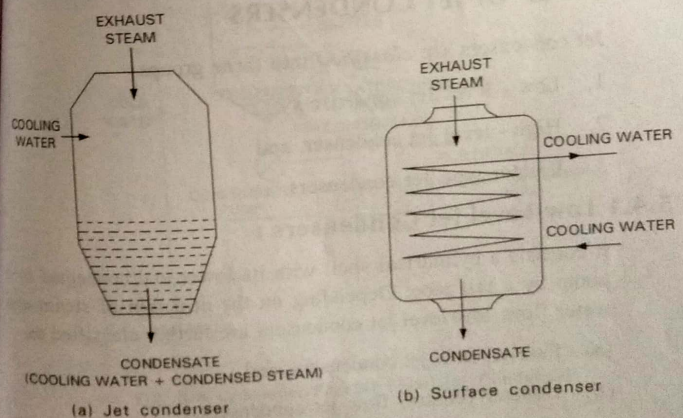


Fig. 5.3

The main differences between jet condenser and surface condenser are given below.

The main differences between jet condenser and surface condenser are given below.

Table 5.1

COMPARISON OF JET AND SURFACE CONDENSER

Jet condensers	Surface condensers
- Water and steam are in direct contact - Less quantity of cooling water is required - Low vacuum is maintained - Condensate can not be reused as hot feed water - Simple design, less maintenance cost. - Occupies less space - Suitable only for small plants.	- Water and steam are not in direct contact - Large quantity of cooling water is required. - High vacuum can be maintained - Condensate can be reused as hot feed water. - Complicated design, high maintenance cost. - Occupies more space. - Used for large plants.

5.4 TYPES OF JET CONDENSERS :

Jet condensers are classified into three groups

1. Low - level Jet condensers
2. High - level Jet condenser, and
3. Ejector type Jet condensers.

5.4.1 Low-Level Jet Condensers :

It consists a cylindrical shell with its lower end connected to a pump by a tail pipe. Depending on the direction of steam and water flow, low-level jet condensers are further classified as

- (a) Parallel flow jet condensers, and
- (b) Counter (contra) flow jet condensers.

(a) Low-level parallel flow jet condensers :

In parallel flow jet condenser cooling water and the exhaust steam flow in the same direction through the shell. Exhaust steam from the engine or turbine enters at the top. The cooling water is sprayed into the steam at the top of the shell. Both the steam

(Chapter-5)

and the cooling water are mixed which results in the condensation of the steam. The mixture (condensate and water) is extracted by a pump at the bottom and discharged to the hot-well. It may be used as boiler feed water if the cooling water is free from impurities.

A parallel flow jet condenser is shown in Fig. 5.4. It does not required pump to lift water, because it flows upwards automatically due to pressure difference. It is provided with dry air pump to extract air and baffle plates inside the shell to facilitate thorough mixing of steam and water.

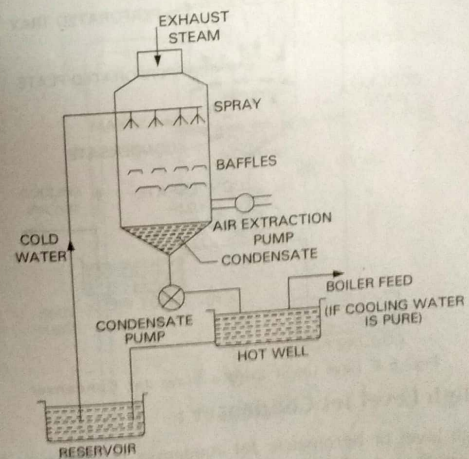


Fig. 5.4 Low-level Parallel Flow Jet Condenser

(b) Contra-flow Jet condenser :

In contra-flow jet condenser the water and steam flow in opposite direction through the shell. In this case exhaust steam from engine or turbine enters at the bottom of the condenser and rise upwards while the cooling water enters at top and flows downwards. Cooling water is sprayed in the form of jets through a series of

(Chapter-5)

perforated trays. The steam while rising up comes in direct contact with water and rapid condensation takes place. Non-condensable air is sucked by an air pump from the top. The mixture of condensate and water is collected at the bottom and discharged to hot well by means of condensate pump.

Low-level contra flow Jet Condenser is shown in Fig. 5.5.

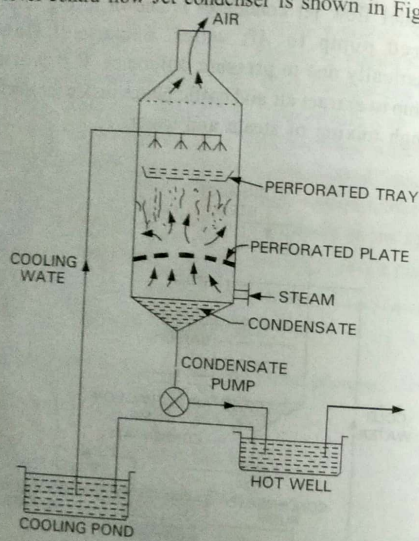


Fig. 5.5 Low Level Contra Flow Jet Condenser

5.4.2 High Level Jet Condenser :

High-level or barometric Jet condenser is similar to low-level contra flow Jet condenser, but condenser shell is situated at an height of 10.36 m from the level of hot well and accumulated condensate will drain by itself without the aid of condensate pump. High-level jet condenser is shown in Fig. 5.6. The exhaust steam enters at the bottom while the cooling water is pumped in at the top. Thus the counter flow is maintained and steam mixed with water gets condensed. The mixture of condensate and cooling water flows down to the hot well. The air is extracted from top with the help of dry air pump.

(Chapter-5)

Advantages & Disadvantages :

Advantages :

1. Does not require condensate pump.
2. Condensate can be reused as feed water if cooling water is free from impurities.

Disadvantages :

1. Requires circulating pump to lift cooling water to the top of the condenser.

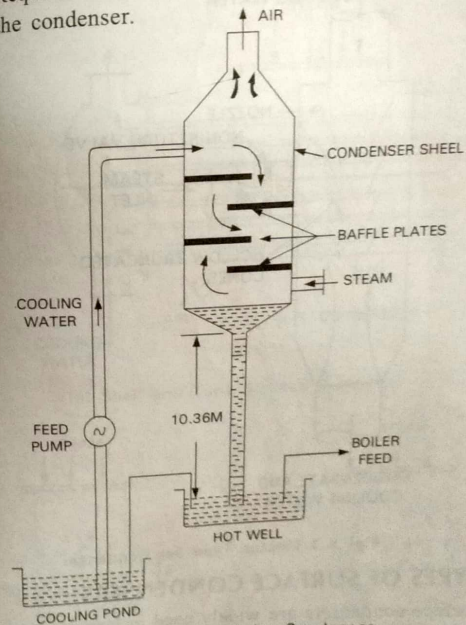


Fig. 5.6 High-level Jet Condenser

5.4.3 Ejector Type Condenser :

Fig. 5.7 shows an ejector type jet condenser. It is suitable for moderate vacuum. In this condensers, cooling water is discharged through a series of convergent nozzles which creates partial vacuum and a mixture of air and exhaust steam is drawn into the condenser through a non-return valve. The steam is thoroughly

(Chapter-5)

5.10 Heat Power Engineering-II

mixed with cooling water in hollow truncated cones and gets condensed. The vacuum is further increased by condensation of steam. The mixture of condensate, air and cooling water is discharged through diverging cone into hot well. In the absence of cooling water supply, water from hot well may rush into the engine or turbine. This is prevented by providing a non-return valve in the steam pipe line.

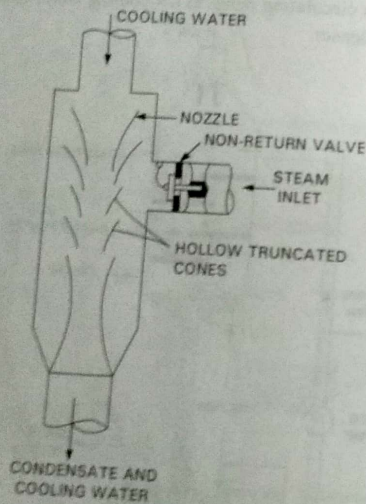


Fig. 5.7 Ejector Type Jet Condenser

5.5 TYPES OF SURFACE CONDENSERS :

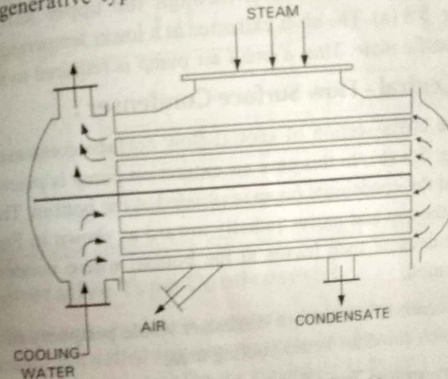
Surface condensers are widely used in large power plants. The details of shell and tube surface condenser are shown in Fig. 5.8 (a). It consists a large number of tubes through which cooling water flows. The steam to be condensed is made to flow around the tubes. The condensate is free from impurities and can be reused as hot feed water to a boiler. Air is removed with the help of air pump in order to reduce the back pressure and improve the efficiency.

(Chapter-5)

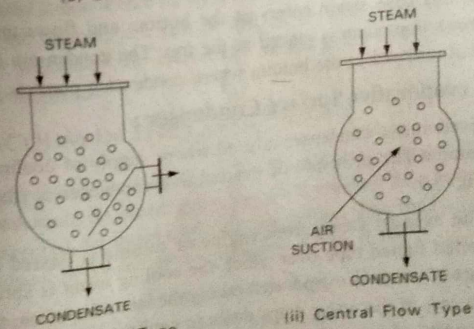
5.11 Condensers

The common forms of surface condensers are :

1. Down flow surface condenser
2. Central flow surface condenser
3. Inverted surface condenser
4. Evaporative condenser
5. Regenerative type condenser.



(a) Shell and Tube Surface Condenser



(i) Down Flow Type
(Cross Section)

(ii) Central Flow Type
(Cross Section)

Fig. 5.8 Types of Surface Condensers

(Chapter-5)

5.12 Heat Power Engineering-II

5.5.1 Down Flow Surface Condenser :

Fig. 5.8 (i) shows a sectional view of a down flow surface condenser. The exhaust steam enters at the top and flows downwards over the tubes through which the cooling water is circulated. The steam is condensed and condensate is extracted at the bottom by condensate pump. The cooling water flows in one direction through the lower half of the tubes and returns in the opposite direction through the upper half as shown in Fig. 5.8 (a). The air is extracted at a lower temperature by providing a baffle plate. Thus a small air pump is required to handle the air.

5.5.2 Central - Flow Surface Condenser :

The cross-section of central-flow surface condenser is shown in Fig. 5.8 (b). In this case air extraction pump is placed at the centre and the condensate pump is situated at the bottom. The steam enters at the top and moves radially inward as shown in Fig. 5.8 (b). The condensate then leaves at the bottom where condensate pump is located.

In central-flow surface condenser whole periphery of tubes is used, which provides better cooling area.

5.5.3 Inverted Type Surface Condenser :

In this case steam enters at the bottom and flows upwards. Air extraction pump is placed at the top. The condensate flows down and removed at the bottom where condensate pump is located.

5.5.4 Evaporative Surface Condenser :

Evaporative condenser is used where supply of cooling water is limited. The principle of evaporative condenser is illustrated in Fig. 5.9.

The exhaust steam from engine or turbine is passed through a coiled finned tube over which the cooling water is sprayed. The part of water is evaporated taking the latent heat from steam and the rest of the water drips down to cooling pond. The evaporated quantity of water is compensated with fresh cooling water. The condensate is recovered from the other end of the coil.

5.13

5.5.5 Regenerative Type Condensers :

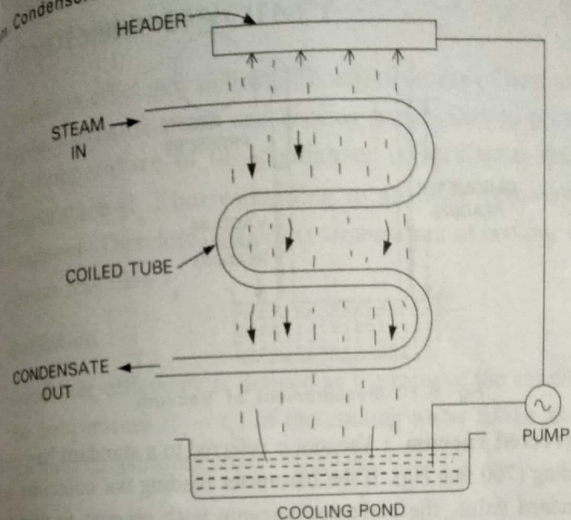


Fig. 5.9 Evaporative Condenser

The evaporated condensers are used in small plants.

5.5.5 Regenerative Type Condensers :

The condensers used in a regenerative method of heating the condensate are called regenerative type condensers. In this case, the condensate after leaving the condenser is passed through the exhaust steam where its temperature is increased. The condensate at high temperature can be reused as feed water to the boiler. This increases the efficiency of the plant and minimise fuel consumption.

5.6 VACUUM IN THE CONDENSER :

The vacuum in a condenser is produced by removing air and condensate from the condenser. The vacuum in a condenser is measured by means of vacuum gauge. The measurement of vacuum in condenser is illustrated in Fig. 5.10.

$$\text{Absolute pressure} = \text{Barometer reading} - \text{Vacuum gauge reading} \\ (\text{Atm. pressure})$$

$$\text{or Vacuum in the condenser} = \text{Barometer reading} - \text{Absolute pressure}$$

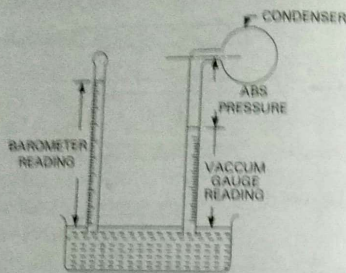


Fig. 5.10 Measurement of Vacuum

Corrected vacuum : Vacuum is referred to a standard barometer reading (760 mm Hg). If the barometer reading not coincides with standard value, the corrected vacuum with respect to standard barometer may be calculated by using the relation.

5.7 VACUUM EFFICIENCY :

Due to presence of air inside the condenser, the actual pressure in the condenser is higher than the saturated pressure corresponding to steam temperature inside the condenser. The presence of air reduces the vacuum due to the increased absolute pressure. The amount of air leakage to condenser is indicated by volumetric efficiency.

Definition : Vacuum efficiency is defined as the ratio of actual vacuum in the condenser to the ideal vacuum (max. obtainable vacuum).

$$\text{Vacuum efficiency} = \frac{\text{Actual vacuum}}{\text{Maximum obtainable vacuum}}$$

$$= \frac{\text{Barometer reading} - \text{abs. pressure in condenser}}{\text{Barometer reading} - \text{pressure at condensate temperature}}$$

$$\text{Absolute pressure in the condenser}$$

$$p = (p_s + p_a)$$

$$\therefore \text{Vacuum } \eta = \frac{p_{bar} - p}{p_{bar} - p_s}$$

If there is no air, then $p_a = 0$ and vacuum η is 100%.

(Chapter-5)

CONDENSER EFFICIENCY :

Condenser efficiency indicate the effectiveness of heat exchange between exhaust steam and cooling water. Due to presence of air, temperature in the condenser is less than saturation temperature (t_g) corresponding to absolute pressure in the condenser. Therefore, the outlet temperature of cooling water is always less than t_g .

Definition :

Condenser efficiency is defined as the ratio of the actual rise in the temperature ($t_2 - t_1$) of the cooling water to the maximum possible temperature rise ($t_g - t_1$)

$$\text{Condenser } \eta = \frac{t_2 - t_1}{t_g - t_1}$$

Where t_1 = inlet temperature of cooling water

t_2 = outlet temperature of cooling water

t_g = saturation temperature at abs. pressure

5.9 DALTON'S LAW :

In a mixture of gases, which do not interact chemically, each gas behaves as if the other were not present. Consider air and vapour contained in a condenser. Then the total pressure exerted on the walls of condenser is equal to the sum of the partial pressures of the air and steam at the common temperature.

Mathematically

$$p = p_s + p_a \quad \text{----- (i)}$$

Where p = Total pressure in the condenser

p_s = Partial pressure of steam

p_a = Partial pressure of air

Also, air and steam exerts its own partial pressure and occupies whole volume of the condenser in which they are contained.

(Chapter-5)

5.16 Heat Power Engineering-II

Mathematically

$$V = V_g = V_a$$

Mass of air present in a condenser may be obtained by using the characteristic gas equation

$$\text{i.e., } p_a V_a = m_a RT$$

$$\text{or } m_a = \frac{p_a V_a}{RT}$$

Where $p_a = p - p_g$ and $V_a = V = V_g$ at a given temperature

Capacity of air pump :

Air leakage into the condenser reduces the vacuum. It is necessary to remove air continuously by air pump. The capacity of air pump is expressed in terms of volume of air extracted per min (m^3/min).

$$\text{Then volume of air/min} = \frac{m_a RT}{p_a}$$

$$\text{and capacity of pump} = \frac{\text{Volume of air / min}}{\text{Volumetric efficiency of pump}}$$

5.10 MASS OF COOLING WATER :

The mass of water required can be estimated by equating the heat removed from exhaust steam for condensing it to heat absorbed by cooling water.

Let m_s = mass of steam condensed, kg / min.

h_{wet} = enthalpy of exhaust steam, kJ/kg

h_{fl} = enthalpy of condensate, kJ/kg

$$\text{Heat removed from steam /min} = m_s (h_{wet} - h_{fl})$$

$$= m_s [x.h_{fg} + (h_f - h_{fl})]$$

$$= m_s [x.h_{fg} + C (t_g - t_c)]$$

(Chapter-5)

Steam Condensers 5.17

Where C = Specific heat of water, = 4.189 kJ/kg K.

t_g = Saturation temperature

t_c = Condensate temperature

If $t_c = t_g$, Then heat removed /min = $m_s (x.h_{fg})$

Heat gained by cooling water/min = $m.C. (t_2 - t_1)$

where m = mass of cooling water, kg / min.

C = specific heat, kJ/kg K

$(t_2 - t_1)$ = rise in temperature of cooling water

$$\therefore m.C. (t_2 - t_1) = m_s [x.h_{fg} + C (t_g - t_c)]$$

The above formula holds good for surface as well as jet condensers. For jet condensers ($t_2 = t_c$).

Example 5.1 :

The vacuum in a condenser is 690 mm Hg. When the barometer reads 760 mm Hg. The mean condensate temperature is 30° C. Find the vacuum efficiency.

Solution :

Vacuum = 690 mm Hg

Barometer reading = 760 mm Hg

Condensate temperature, t_g = 30°C

p_s at 30°C = 0.04242 bar

= 0.04242 × 750

= 31.815 mm Hg

$$\text{Vacuum } \eta = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{690}{760 - 31.815}$$

$$= 0.9475$$

$$= 94.75\% \text{ Ans.}$$

(Chapter-5)

Example 5.2 :

In a condenser vacuum is 715 mm of Hg when the barometer reads 765 mm of Hg. The inlet temperature of cooling water is 15°C and outlet temperature of water is 25°C. Determine the condenser efficiency. [Oct./Nov. 2007]

Solution :

Vacuum = 715 mm Hg

Barometer reading = 765 mm Hg

∴ Abs. pressure = 765 - 715 = 50 mm Hg = 0.0667 bar

Saturation temperature at 0.0667 bar, t_g 38.84°C

Inlet temperature, t_1 = 15°C

Outlet temperature, t_2 = 25°C

$$\begin{aligned} \text{Condenser efficiency} &= \frac{t_2 - t_1}{t_g - t_1} \\ &= \frac{25 - 15}{38.84 - 15} = 0.52 \\ &= 52\% \text{ Ans.} \end{aligned}$$

Example 5.3 :

The absolute pressure in the surface condenser is 0.1134 bar. When the barometer indicates 757 mm Hg. The condensate temperature is 40°C. Determine

(a) Partial pressure of air in condenser

(b) Vacuum efficiency

(c) Mass of air present in the condenser per kg of steam.

Solution :

Absolute pressure, p = 0.1134 bar

Barometer reading = 757 mm Hg

= 1.0093 bar

∴ Vacuum = 1.0093 - 0.1134 = 0.8959 bar

= 671.925 mm Hg

p_g at 40°C = 0.07375 bar (from steam tables)

= 55.31 mm Hg

and v_g at 40°C = 19.546 m³/kg

From Dalton's law

Partial pressure of air

$$\begin{aligned} p_a &= p - p_g \\ &= 0.1134 - 0.07375 \\ &= 0.03965 \text{ bar Ans.} \end{aligned}$$

$$\begin{aligned} \text{Vacuum efficiency} &= \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{671.925}{757 - 55.31} = 0.9575 \\ &= 95.75\% \text{ Ans.} \end{aligned}$$

Mass of air/kg of steam

$$\begin{aligned} m_a &= \frac{p_a V_a}{RT} \quad (V_a = V_g = 19.546; T = 40 + 273 \\ &= 313^\circ\text{K}) \\ &= \frac{10^{-2} \times 0.03965 \times 19.546}{0.287 \times 313} \\ &= 0.8627 \text{ kg/kg of steam Ans.} \end{aligned}$$

Example 5.4 :

In a steam condenser, the mean condensate temperature is 35°C. The vacuum in the condenser is 675 mm Hg when the barometer reads 760 mm Hg. The water enters at 20°C and leaves at 27°C. Determine the condenser efficiency and vacuum efficiency.

Solution :

Vacuum = 675 mm Hg

Barometer reading = 760 mm Hg

Absolute pressure = 760 - 675

= 85 mm Hg, = 0.1133 bar

5.20 Heat Power Engineering-II
Saturation temperature corresponding to 0.1133 bar = 48.28°C

$$\begin{aligned}\text{Condenser efficiency} &= \frac{t_2 - t_1}{t_g - t_1} \\ &= \frac{27 - 20}{48.28 - 20} = 0.2482 \\ &= 24.82 \% \text{ Ans.}\end{aligned}$$

$$\text{Vacuum } \eta = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{\text{Actual vacuum}}{\text{Barometer reading} - p_g}$$

$$\begin{aligned}\text{Where } p_g &= \text{Partial pressure of steam at } 35^\circ\text{C (condensate temp)} \\ &= 0.05622 \text{ bar} = 42.165 \text{ mm Hg.}\end{aligned}$$

$$\begin{aligned}\therefore \text{Vacuum } \eta &= \frac{675}{760 - 42.165} = 0.94 \\ &= 94\% \text{ Ans.}\end{aligned}$$

Example 5.5 :

A surface condenser is designed to handle 1200 kg of steam per hour. The steam enters at 0.08 bar and 0.9 dry and the condensate leaves at the corresponding saturation temperature. The pressure is constant throughout condenser. Estimate the cooling water flow rate if the inlet and outlet temperatures of cooling water are 20°C and 30°C. Assume specific heat of water as 4.2 kJ/kg K.

Solution :

$$\text{Mass of steam} = 1200 \text{ kg/hr}$$

$$\text{At 0.08 bar, } h_{fg} = 2403.2 \text{ kJ/kg}$$

$$\text{Dryness fraction } x = 0.9$$

$$\begin{aligned}\text{Rise in cooling water temperature, } (t_2 - t_1) &= (30 - 20) \\ &= 10^\circ\text{C}\end{aligned}$$

$$\text{Heat absorbed by water} = \text{Heat removed from steam}$$

$$m.C. (t_2 - t_1) = m_s [x.h_{fg} + (t_g - t_c)]$$

(Chapter-5)

Steam Condensers 5.21

$$\begin{aligned}m.C. dt &= m_s x h_{fg} \quad (\because t_s = t_c) \\ m \times 4.2 \times 10 &= 1200 \times 0.9 \times 2403.2 \\ m &= \frac{1200 \times 0.9 \times 2403.2}{4.2 \times 10} \\ &= 61796.6 \text{ kg/hr}\end{aligned}$$

$$\text{Mass of cooling water} = 61796.6 \text{ kg/hr Ans.}$$

Example 5.6 :

Determine the mass of injection water per kg of steam condensed if the exhaust steam is at 0.15 bar abs and has a dryness fraction of 0.9. Temperature of injection water at inlet is 20°C. Allow 5°C of under cooling. Take specific heat of water as 4.2 kJ/kg K.

Solution :

$$\text{Dryness fraction, } x = 0.9$$

From steam tables at 0.15 bar,

$$t_g = 54^\circ\text{C}, \quad h_f = 226 \text{ kJ/kg}, \quad h_{fg} = 2373.2 \text{ kJ/kg}$$

Condensate temperature,

$$t_c = 54 - 5 = 49^\circ\text{C}$$

$$\text{For jet condenser, } t_2 = t_c = 49^\circ\text{C}$$

$$\therefore m.C. (t_c - t_1) = m_s [x.h_{fg} + C (t_g - t_c)]$$

$$\therefore \text{mass of cooling water,}$$

$$m = \frac{1[0.9 \times 2373.2 + 4.2 \times 5]}{4.2(49 - 20)}$$

$$= 17.7 \text{ kg/kg of steam Ans.}$$

Example 5.7 :

The following observations were recorded during a test on a surface condenser

$$\text{Mean condensate temperature} = 35^\circ\text{C}$$

5.22 Heat Power Engineering-II

Hot well temperature	= 30°C
Condenser vacuum	= 700 mm Hg
Barometer reading	= 765 mm Hg
Condensate collected	= 15.5 kg/min
Cooling water	= 36800 kg/hr
Rise of cooling water temperature	= 12.5°C

Calculate (a) Dryness fraction of steam as it enters the condenser

(b) Mass of air present /m³ volume of condenser

(c) Vacuum efficiency.

Solution :

Absolute pressure,	$p = (765 - 700)$
	= 65 mm Hg
	= 0.0866 bar
Partial pressure of steam at 35°C,	$p_s = 0.0562$ bar
	= 42.15 mm Hg
From Dalton's law	$p_s = p - p_a$
	= 0.086 - 0.0562
	= 0.0304 bar

Let x be the dryness fraction of steam, equating the heat removed from steam to the heat gained by cooling water,

$$m_s [x h_g + (h_f - h_n)] = m.C. (t_2 - t_1)$$

Assume $C = 4.18$ kJ/kg K, and from steam tables

at 0.0562 bar i.e., 35°C, $h_g = 2418.8$ kJ/kg

$h_f = 146.6$ kJ/kg

and h_n at 30°C = 125.7 kJ/kg

$$15.5 [2418.8x + (146.6 - 125.7)] = \frac{36800}{60} \times 4.18 \times 12.5$$

$$2418.8x = 2067.53 - 20.9$$

$$\text{or } x = \frac{2046.63}{2418.8}$$

$$= 0.846 \text{ Ans.}$$

(Chapter-5)

5.23 Steam Condensers

Dryness fraction can also be obtained by using the equation

$$m_s [x h_g + C (t_g - t_c)] = m.C. (t_2 - t_1)$$

$$15.5 [2418.8x + 4.18 (35 - 30)] = \frac{36800}{60} \times 4.18 \times 12.5$$

$$\Rightarrow x = 0.846$$

Mass of air present/m³ of volume of condenser

$$m_a = \frac{P_a V_a}{RT}$$

$$= \frac{10^3 \times 0.0304 \times 1}{0.288 \times (273 + 35)}$$

$$= 0.0343 \text{ kg/m}^3 \text{ of condensate Ans.}$$

$$\text{Vacuum efficiency} = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}}$$

$$= \left[\frac{700}{765 - 42.15} \right] \times 100$$

$$= 96.8\% \text{ Ans.}$$

Example 5.8 :

The vacuum at the air extraction pipe in a condenser is 710 mm Hg with barometer reading 760 mm Hg and the temperature is 36°C. The leakage of air amounts to 4 kg per 10000 kg of steam. Determine the capacity of air pump required for condenser. Take volumetric efficiency of pump as 85%. Assume R for air as 0.287 kJ/kg K.

Also find the mass of steam associated with this air.

Solution :

$$\text{Absolute pressure, } p = 760 - 710 = 50 \text{ mm Hg}$$

$$= 0.0667 \text{ bar}$$

$$\text{Partial pressure of steam at 36°C, } p_s = 0.0594 \text{ bar}$$

From Dalton's law,

$$\text{Partial pressure of air, } p_a = 0.0667 - 0.0594$$

$$= 0.0073 \text{ bar}$$

(Chapter-5)

5.24 Heat Power Engineering-II

$$\text{Mass of air/kg of steam, } m_a = \frac{4}{10000} = 0.0004 \text{ kg}$$

$$\text{Volume of air, } V_a = \frac{m_a RT}{p_a}; \text{ Where } T = 6 + 273$$

$$= 309 \text{ K}$$

$$= \frac{0.0004 \times 0.287 \times 309}{10^2 \times 0.0073}$$

$$= 0.0486 \text{ m}^3/\text{kg of steam}$$

$$\therefore \text{Capacity of air pump} = \frac{\text{Volume of air}}{\text{Volumetric efficiency}}$$

$$= \frac{0.0486}{0.85}$$

$$= 0.0573 \text{ m}^3/\text{kg of steam Ans.}$$

From Dalton's law

$$V_f = V_a = 0.0486 \text{ m}^3$$

$$\text{Specific volume of steam at } 0.0594 \text{ bar} = 23.967 \text{ m}^3$$

$$\therefore \text{Mass of steam associated with air} = \frac{0.0486}{23.967}$$

$$= 0.00203 \text{ kg Ans.}$$

Example 5.9 :

A surface condenser deals with 12000 kg of steam per hour. Air leakage into the condenser is found to be 4 kg/hr. The vacuum and temperature at the air pump suction are 700 mm of Hg and 36°C respectively. The barometric pressure is 760 mm Hg. Compute the volumetric capacity in m³/min of wet air pump.

Solution :

$$\text{Absolute pressure, } p = 760 - 700$$

$$= 60 \text{ mm Hg}$$

$$= 0.08 \text{ bar}$$

(Chapter 5)

5.25

Partial pressure of steam at 36°C $p_s = 0.0594 \text{ bar}$

From Dalton's law,

$$\text{Partial pressure of air, } p_a = 0.08 - 0.0594$$

$$= 0.0206 \text{ bar}$$

$$\text{Mass of air leakage, } m_a = 4 \text{ kg/hr}$$

$$\text{Temperature of air, } T = 36 + 273$$

$$= 309 \text{ K}$$

$$\text{Volume of air, } V_a = \frac{m_a RT}{p_a}$$

$$\text{Assume } R = 0.287 \text{ kJ/kg K}$$

$$V_a = \frac{4 \times 0.287 \times 309}{10^2 \times 0.0206}$$

$$= 172.2 \text{ m}^3/\text{hr}$$

$$\text{Volume of condensate} = \frac{\text{mass}}{\text{density}}$$

$$= \frac{12000}{1000}$$

$$= 12 \text{ m}^3/\text{hr}$$

$$\text{Volumetric capacity of wet pump} = \text{vol of air} + \text{vol. of condensate}$$

$$= 172.2 + 12$$

$$= 184.2 \text{ m}^3/\text{hr Ans.}$$

Example 5.10 :

Following observations were made during a trial on a steam condenser

$$\text{Barometric pressure} = 760 \text{ mm of Hg}$$

$$\text{Vacuum reading} = 700 \text{ mm of Hg}$$

$$\text{Rate of cooling water} = 1000 \text{ kg/minute}$$

$$\text{Inlet temp. of cooling water} = 20^\circ\text{C}$$

$$\text{Outlet temp. of cooling water} = 30^\circ\text{C}$$

$$\text{Quantity of steam condensed} = 20 \text{ kg/min}$$

$$\text{Hotwell temperature} = 32^\circ\text{C}$$

$$\text{Mean temp. of the condensate} = 35^\circ\text{C}$$

(Chapter 5)

5.26 Heat Power Engineering-II

Determine :

- (i) Vacuum efficiency
- (ii) Condenser efficiency
- (iii) Dryness fraction of exhaust steam
- (iv) Sub-cooling of condensate
- (v) Amount of air leakage per kg of steam condensed

Assume, R for air = 0.287 kJ/kg K , and C for water = 4.18 kJ/kg K

[Oct./Nov. 2009]

Solution :

$$\begin{aligned} \text{Absolute pressure, } p &= 760 - 700 \\ &= 60 \text{ mm Hg} = 0.08 \text{ bar} \end{aligned}$$

$$\begin{aligned} \text{Partial pressure of steam at } 35^\circ\text{C, } p_s &= 0.0562 \text{ bar} \\ &= 42.15 \text{ mm Hg} \end{aligned}$$

$$\begin{aligned} \therefore \text{Partial pressure of air } p_a &= 0.08 - 0.0562 \\ &= 0.0238 \text{ bar} \end{aligned}$$

(i) **Vacuum efficiency :**

$$\begin{aligned} \text{Vacuum } \eta &= \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{700}{760 - 42.15} = 0.975 \\ &= \mathbf{97.5 \% \text{ Ans.}} \end{aligned}$$

(ii) **Condenser efficiency :**

$$\text{Condenser } \eta = \frac{t_g - t_1}{t_g - t_1}$$

$$\begin{aligned} \text{Where } t_g &= \text{saturation temperature at } 0.08 \text{ bar (abs. pressure)} \\ &= 41.54^\circ\text{C (from steam tables)} \end{aligned}$$

$$\begin{aligned} \therefore \text{Condenser } \eta &= \frac{30 - 20}{41.54 - 20} = 0.464 \\ &= \mathbf{46.4\% \text{ Ans.}} \end{aligned}$$

(iii) **Dryness fraction of exhaust steam :**

The dryness fraction can be calculated by equating the heat lost by steam to heat gained by cooling water.

(Chapter-5)

5.27 Steam Condensers

$$\text{Heat lost by steam} = \text{Heat gained by cooling water}$$

$$m_s [x h_g + (h_f - h_n)] = m_c (t_2 - t_1)$$

$$\text{At } 0.08 \text{ bar, } L = 2403.2 \text{ kJ/kg, } h_f = 173.9 \text{ kJ/kg,}$$

$$\text{and } h_n \text{ at } 32^\circ\text{C} = 134 \text{ kJ/kg}$$

$$20 [2403.2 x + (173.9 - 134)] = 1000 \times 4.18 (30 - 20)$$

$$2403.2 x = (2090 - 39.9)$$

$$\text{or } x = \mathbf{0.853 \text{ Ans.}}$$

(iv) **Sub-cooling (under-cooling) of condensate**

$$= \text{condensate temperature} - \text{hotwell temp.}$$

$$= 35 - 32 = \mathbf{3^\circ\text{C Ans.}}$$

(v) **Amount of air leakage/kg of steam condensed :**

$$\text{Specific volume of steam at } 35^\circ\text{C, } V_g = 25.245 \text{ m}^3/\text{kg}$$

$$\text{From Dalton's law, } V_g = V_a = 25.245 \text{ m}^3$$

Mass of air leakage/kg of steam condensed

$$m_a = \frac{p_a V_a}{RT}$$

$$= \frac{10^2 \times 0.0238 \times 25.245}{0.287 \times (273 + 35)}$$

$$= \mathbf{1.47 \text{ kg/kg of steam Ans.}}$$

5.11 **SOURCES OF AIR IN CONDENSERS :**

The main sources of air in the condensers are given below

1. Air leaks through various joints into the condenser where the pressure is lower than atmospheric pressure.
2. Air dissolved in feed water is retained in steam and therefore it is carried into the condenser through exhaust steam.
3. In case of Jet condensers, cooling water is also a source of dissolved air.

(Chapter-5)

5.12 EFFECTS OF AIR LEAKAGE INTO THE CONDENSERS :

The performance of condensers is effected by the amount of air present in the condenser. The presence of air in the condenser has the following effects.

- It increases the back pressure thereby reduces the work output of the prime mover and hence the efficiency of the cycle.
- It lowers the partial pressure of steam and at lower pressure, it has more latent heat. Thus more cooling water is required.
- Heat transfer is reduced due to air film over the surface. Therefore surface area of tubes to be increased. So a larger condenser is required.

5.13 AIR PUMP :

The function of air pump is to maintain a desired vacuum in the condenser by removing air. Air pump which removes both air and condensate is called *wet air pump*, and which is only used for removing the air is called *dry air pump*. There are several types of air pumps.

5.13.1 Edwards's Air Pump :

It is a reciprocating piston type wet air pump. The working principle of Edward's air pump is illustrated in Fig. 5.11. When the piston is at the top, the condensate and air from the condenser is collected at conical bottom of the barrel. As the piston moves down, a vacuum is created in the barrel above the piston and delivery valve is closed. When the ports are uncovered, the piston forces the air and condensate through ports into the barrel above the piston. During the upward motion of the piston the air and condensate mixture is compressed to a pressure greater than atmospheric pressure causing the delivery valve to open. Then the air and condensate are delivered to the hot well. During the downward motion of the piston again vacuum is created in the barrel, delivery valve closed and the air and condensate mixture collected at the bottom is pushed through the ports. Then the cycle will be repeated.

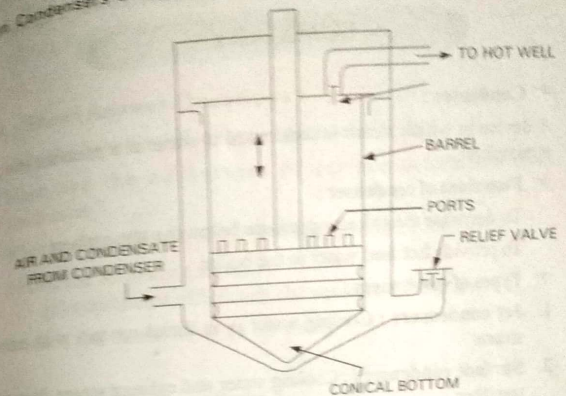


Fig. 5.11

The piston is provided with conical head which facilitate forcing the air and condensate into the barrel above the piston. When the pressure at the bottom of the pump exceeds that of the atmosphere, water sealed relief valve opens automatically discharging some of air and condensate to the atmosphere.

5.13.2 Air Ejector :

The principle of air ejector is illustrated in Fig. 5.12. Steam enters a convergent-divergent nozzle and leaves at A at low pressure but with high velocity. The pipe C is connected to the air exit of the condenser and due to low pressure at A, the air from the condenser is drawn up the pipe and mixed with steam at A. The high velocity air-steam mixture is passed through the diffuser B where the velocity gradually decreases with increase in pressure above atmosphere, and finally exhaust into atmosphere.

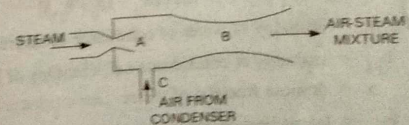


Fig. 5.12 Air Ejector

SUMMARY

☞ Condenser :

A device in which steam is condensed to water at a pressure less than atmosphere.

☞ Functions of condenser :

- To decrease the exhaust pressure below the atmosphere
- To provide hot feed water to the boiler

☞ Types of condensers :

1. **Jet condensers** : Cooling water is in direct contact with exhaust steam.
2. **Surface condensers** : Cooling water and exhaust steam do not mix together

$$\text{Vacuum efficiency} = \frac{\text{actual vacuum}}{\text{ideal vacuum}} = \frac{p_{\text{bar}} - p}{p_{\text{bar}} - p_s}$$

Where p = absolute pressure in the condensate temperature

p_s = partial pressure of steam at condensate temperature

p_{bar} = atmospheric pressure, recorded by barometer.

$$\text{☞ Condenser efficiency} = \frac{t_2 - t_1}{t_g - t_1}$$

where t_1 = inlet temperature of water

t_2 = outlet temperature of water

t_g = saturation temperature at abs. pressure (condenser pressure).

☞ **Cooling water** : Cooling water required may be estimated by using the relation,

Heat gained by water = heat lost by exhaust steam

$$m.C. (t_2 - t_1) = ms [xh_{fg} + (h_f - h_{f1})]$$

Where h_f = enthalpy of water of condenser pressure

h_{f1} = enthalpy of condensate i.e., enthalpy of hot well

x = dryness fraction of steam

h_{fg} = latent heat of exhaust steam

$(t_2 - t_1)$ = rise in temperature of cooling water

UNIT -5

RANKINE CYCLE:

A Rankine vapour cycle is based on a modified carnot cycle to overcome its limitations. It consists of four steady flow processes as in figure

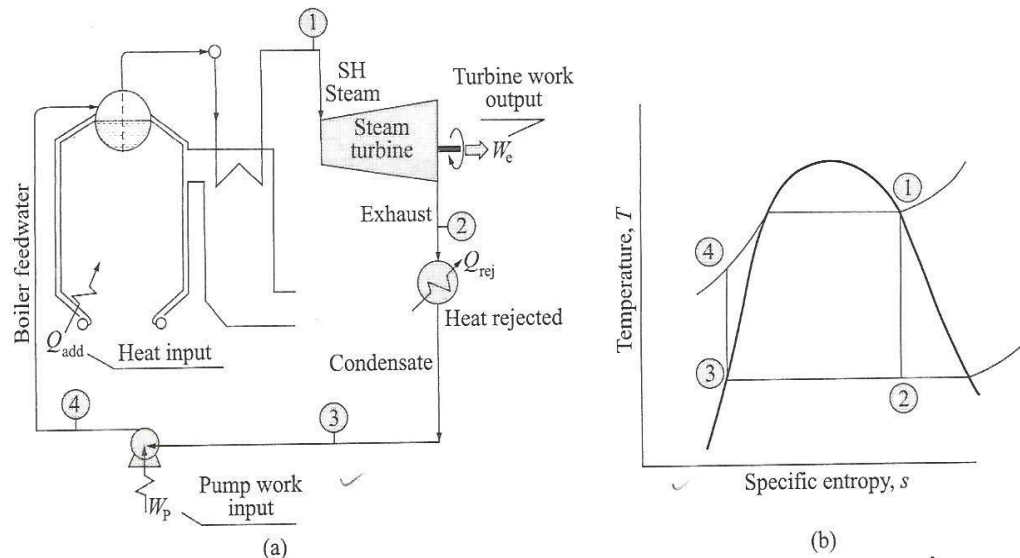


Fig.1 Schematic layout of Rankine cycle

Process 4-1:

Heat is added in the boiler to the BFW, which is a constant pressure process generating a dry, saturated steam at saturation temp. corresponding to the boiler operating pressure,

$$Q_{add} = m(h_1 - h_4)$$

Process 1-2:

The steam is reversibly and adiabatically expanded in the turbine.

$$\text{So, turbine work input} = W_e = m(h_1 - h_2)$$

Process 2-3:

Constant pressure heat rejection in the condenser. The cooling water extracts the latent heat of condensation from the exit steam exhausted to the condenser. The condensation is complete. The entire vapour is converted into a condensate.

$$\text{Therefore heat rejection} = m(h_2 - h_3)$$

Process 3-4:

Pump work. This work must be apportioned from the turbine output.

$$W_p = m(h_4 - h_3)$$

Net work output, $W_{net} = W_e - W_p$

Therefore, efficiency,

$$\begin{aligned}\eta &= \frac{W_{net}}{Q_{add}} \\ &= \frac{W_e - W_p}{Q_{add}} \\ &= \frac{m(h_1 - h_2) - m(h_4 - h_3)}{m(h_1 - h_4)}\end{aligned}$$

RANKINE CYCLE WITH SUPERHEATED STEAM:

If the heating of the working fluid (BFW) is continued beyond the dry saturation point, i.e well into superheat regime before feeding it to the turbine, i.e; state instead of state 1 , the amount of heat added increase bringing about an incipient increase in the work output

$$(\Delta. W_{output} = \text{area } 1 - 1' - 2' - 2)$$

So, superheating begets a higher cycle efficiency.

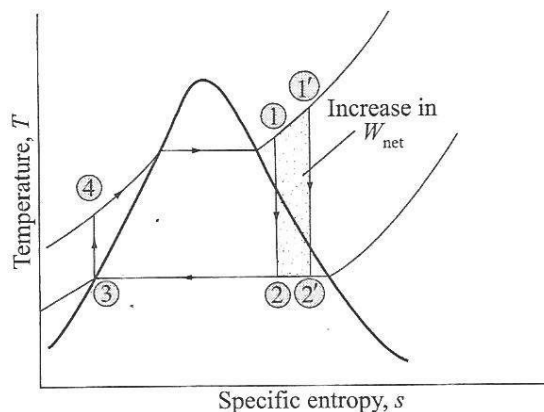


Fig.2 Rankine cycle for Super heated Steam

CONCEPT OF MEAN TEMPERATURE OF HEAT ADDITION:

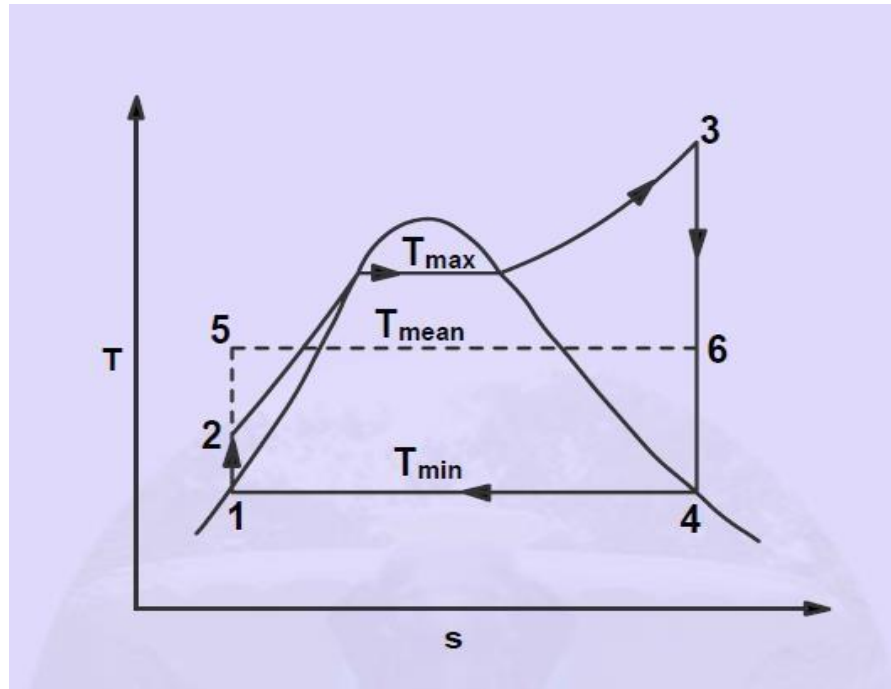


Fig.4 Mean Temperature of Heat Addition

Note:

If T_m is the mean temperature of heat addition as shown in the above figure, so that the area under curve 2-3 is equal to area under curve 5-6, then heat added.

$$Q_1 = H.A = (h_3 - h_2) = T_m(s_3 - s_2)$$

$$T_m = \frac{(h_3 - h_2)}{(s_3 - s_2)}$$

$$\text{If } Q_2 = \text{heat rejected} = (h_4 - h_1) = T_{\min}(s_4 - s_1) = T_{\min}(s_3 - s_2)$$

$$\eta_{\text{rankine}} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_{\min}(s_3 - s_2)}{T_m(s_3 - s_2)}$$

$$\eta_{\text{rankine}} = 1 - \frac{T_{\min}}{T_m}$$

METHODS TO IMPROVE CYCLE PERFORMANCE:

There are three ways to increase the efficiency of the simple ideal Rankine cycle.

1. Decreasing the condenser pressure:

The effect of lowering the condenser pressure on the Rankine cycle efficiency is illustrated on a T-s diagram on the left. Steam exits as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure in the condenser. So lower the pressure in the condenser, lower the temperature of the steam, which is the heat rejection temperature. The blue area is the net work increases due to the decreasing of the condenser pressure.

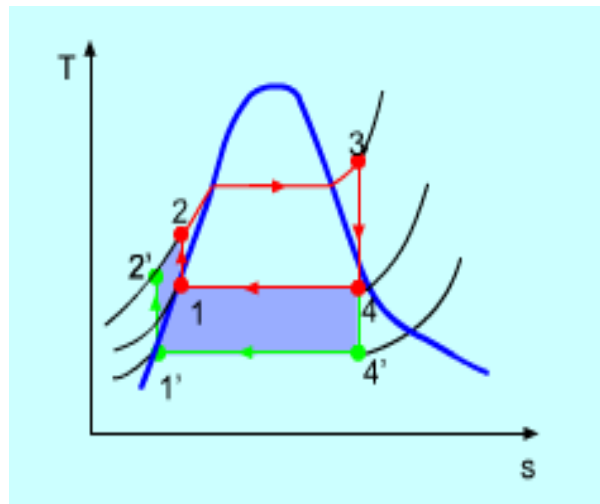


Fig.5 The Effect of Lowing the Condenser Pressure

2. Superheating the steam to a high temperature:

The effect of superheating the steam to a high temperature on the Rankine cycle efficiency is illustrated on a T-s diagram on the left. By superheating the stream to a high temperature (from state 3 to state 3'), the average steam temperature during heat addition can be increased. The blue area is the network increased due to superheating the steam to a high temperature.

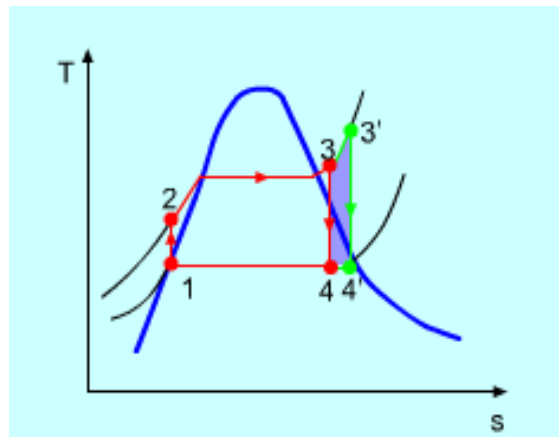


Fig.6 The Effect of Superheating the Steam to a Higher Temperature

3. Increasing the boiler pressure:

The effect of increasing the boiler pressure on the Rankine cycle efficiency is illustrated on a T-s diagram on the left. If the operating pressure of the boiler is increased, (process 2-3 to process 2'-3'), then the boiling temperature of the steam raises automatically. For a fixed inlet turbine temperature, the blue area is the network increased and the gray area is the network decreased. Also, the moisture content of the steam

increases from state 4 to state 4', which is an undesirable side effect. This side effect can be corrected by reheating the steam, and results in the reheat Rankine cycle.

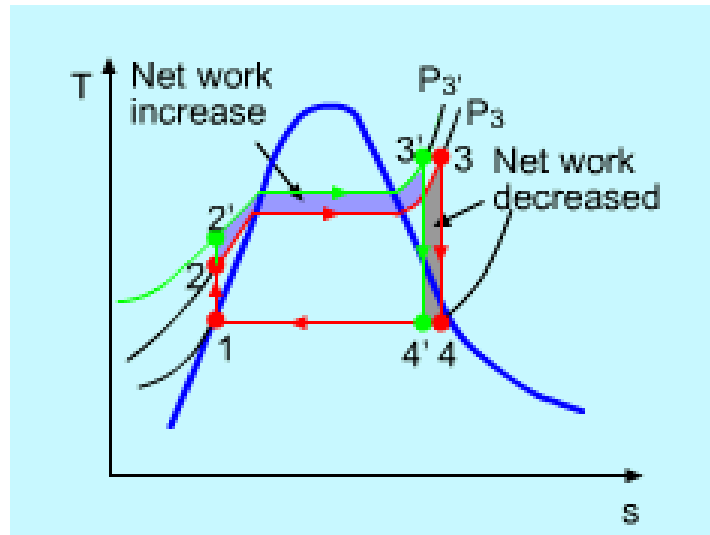


Fig.7 The Effect of Increase the Boiler Pressure

REHEAT CYCLE:

- The efficiency of the Rankine cycle can increase by expanding the steam in the turbine in two stages, and reheat it in between.
- In other words, modify the simple ideal Rankine cycle with reheat process.
- Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants.

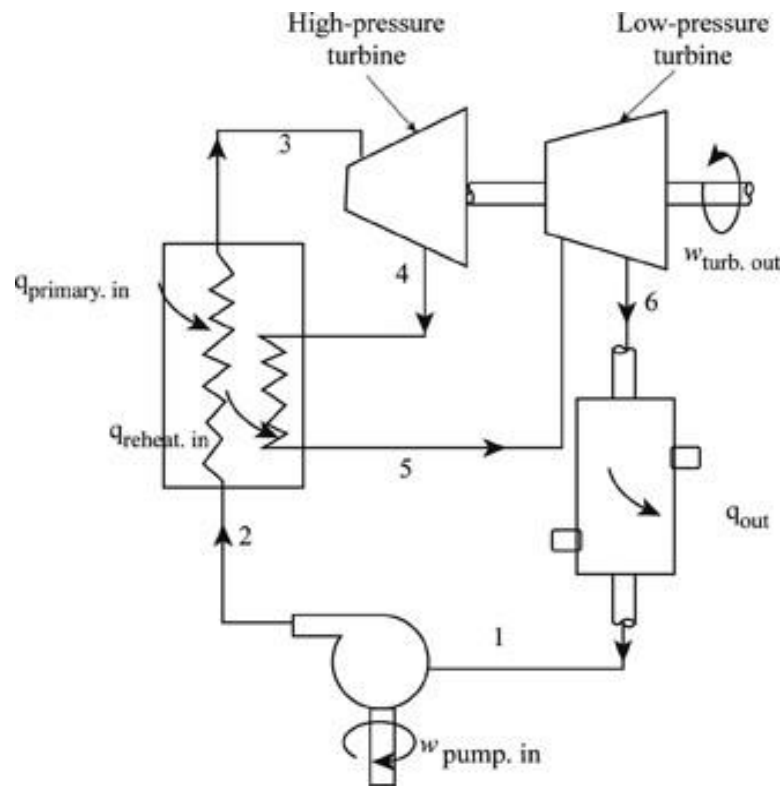


Fig.8 Schematic of Reheat cycle

The processes of reheat rankine cycle are explained below using the enthalpy-entropy diagram.

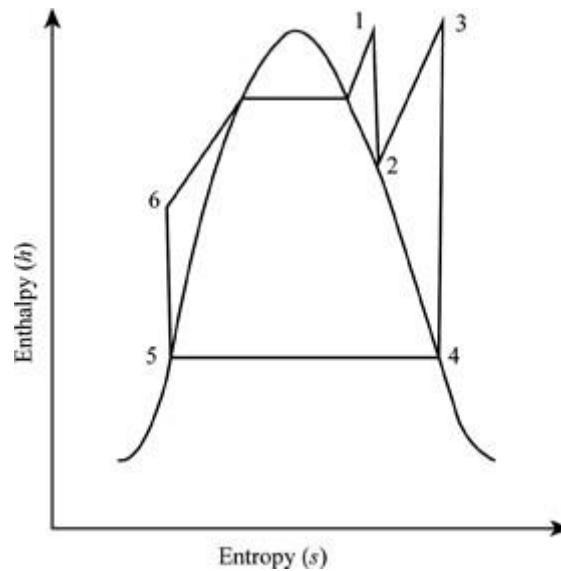


Fig.9 h-s Diagram of Reheat cycle

Process 1-2

This process is known as high pressure turbine.

Obtain the work-done by turbine using enthalpies of inlet and exit.

$$W_T = h_1 - h_2$$

Where, h_1 is the enthalpy for inlet state 1 and h_2 is the enthalpy for outlet state 2.

To find the values of h_1 and h_2 , use the given values of pressure and temperature at point 1 to calculate first entropy at state 1 (S_1) and enthalpy (h_1) from the superheated steam table.

Next to find the variable is dryness fraction to calculate the value of entropy at exit state 2 (S_2) with the given formula.

$$\begin{aligned} s_1 &= s_2 \\ &= s_{f_2} + x_2 s_{fg_2} \end{aligned}$$

Write the formula for enthalpy at exit state 2.

$$h_2 = h_{f_2} + x_2 h_{fg_2}$$

Process 2-3

This process is known as boiler.

In this process calculate the values of h_3 and S_3 .

Process 3-4

This process is known as low pressure turbine.

In this process calculate the values of h_4 and S_4 .

Where, S_4 is the entropy at state 4.

Process 4-5

This process is known as condenser.

In this process calculate the heat expelled from condenser.

$$Q_R = h_4 - h_5$$

Process 5-6

This process is known as pump.

In this process work done is calculated by the pump.

$$W_p = h_6 - h_5$$

Where, h_6 is the enthalpy at state 6.

Process 6-1

This process is known as boiler.

In this process obtain the heat supplied.

$$Q_s = h - h_6$$

$$= (h - h_5 - W_p) J$$

Where, h_5 is the enthalpy at state 5.

RANKINE CYCLE WITH REGENERATION:

Internal heat is exchanged between the expanding fluid (steam) in the turbine and compressed fluid (water) before heat addition. The aim is to increase the cycle efficiency. Heat exchanged in turbine is not practical, hence feed water heating. The compressed water out of the pump is heated in a number of steps rather than continuously, by steam bled from the turbine at selected stage. These heat exchangers are called feed water heaters (F.W.H).

TYPES OF F.W.H:

- 1) Open or direct contact F.W.H.
- 2) Closed with drains cascaded backward.
- 3) Closed with drain pumps forward.

A careful examination of the T-s diagram of the Rankine cycle redrawn in Fig below reveals that heat is transferred to the working fluid during process 2-2' at a relatively low temperature. This lowers the average heat addition temperature 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. One such possibility is to transfer heat to the feedwater from the expanding steam in a counterflow 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.

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 (FWH).

Regeneration not only improves cycle efficiency, but also provides a convenient means of deaerating the feedwater (removing the air that leaks in at the condenser) to prevent corrosion in the boiler. It also helps control the large volume flow rate of the steam at the final stages of the turbine (due to the

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).



1. The heating process in the boiler reduced.
2. The thermal stresses set up in the boiler are minimized. This is due to the fact that temperature ranges in the boiler are reduced.
3. The thermal efficiency is improved because the average temperature of heat addition to the cycle is increased.
4. A small size condenser is required.

1. The plant becomes more complicated.
2. Because of addition of heaters greater maintenance is required.
3. The heaters are costly and the gain in thermal efficiency is not much in comparison to the heavier costs.

11

An open (or direct-contact) feed water heater is basically a mixing chamber, where the steam extracted from the turbine mixes with the feed water exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure. The schematic of a steam power plant with one open feed water heater (also called single-stage regenerative cycle) and the T-s diagram of the cycle are shown in Fig. 11. In an ideal regenerative Rankine cycle, steam enters the turbine at the boiler pressure (state 5) and expands isentropically to an intermediate pressure (state 6). Some steam is extracted at this state and routed to the feed water heater, while the remaining steam continues to expand isentropically to the condenser pressure (state 7). This steam leaves the condenser as a saturated liquid at the condenser pressure (state 1). The condensed water, which is also called the feed water, then enters an isentropic pump, where it is compressed to the feed water heater pressure (state 2) and is routed to the feed water heater, where it mixes with the steam extracted from the turbine.

The fraction of the steam extracted is such that the mixture leaves the heater as a saturated liquid at the heater pressure (state 3). A second pump raises the pressure of the water to the boiler pressure (state 4). The cycle is completed by heating the water in the boiler to the turbine inlet state (state 5).

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 rates are different in different components.

If the mass flow rate through the boiler is m , for example, it is $(1 - y)m$ through the condenser. This aspect of the regenerative Rankine cycle should be considered in the analysis of the cycle as well as in the interpretation of the areas on the T-s diagram. In light of Fig. 11, the heat and work interactions of a regenerative Rankine cycle with one feed water heater can be expressed per unit mass of steam flowing through the boiler as follows:

$$\begin{aligned}
 q_{\text{in}} &= h_5 - h_4 \\
 q_{\text{out}} &= (1 - y)(h_7 - h_1) \\
 w_{\text{turb,out}} &= (h_5 - h_6) + (1 - y)(h_6 - h_7) \\
 w_{\text{pump,in}} &= (1 - y)w_{\text{pump I,in}} + w_{\text{pump II,in}}
 \end{aligned}$$

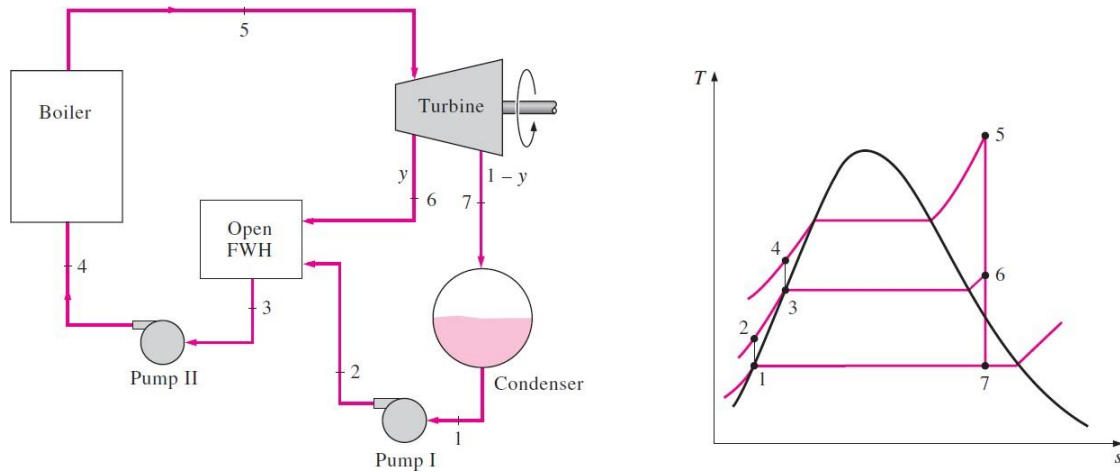


Fig.11 The ideal regenerative Rankine cycle with an open feedwater heater.

Where,

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

$$w_{\text{pump I, in}} = v_1 (P_2 - P_1)$$

$$w_{\text{pump II, in}} = v_3 (P_4 - P_3)$$

The thermal efficiency of the Rankine cycle increases as a result of regeneration. This is because regeneration raises the average temperature at which heat is transferred to the steam in the boiler by raising the temperature of the water before it enters the boiler. The cycle efficiency increases further as the number of feed water heaters is increased. Many large plants in operation today use as many as eight feed water heaters. The optimum number of feed water heaters is determined from economic considerations. The use of an additional feed water heater cannot be justified unless it saves more from the fuel costs than its own cost.

BOILERS:

- ❖ A boiler (or steam generator) is a closed vessel in which water, under pressure, is converted into steam. The heat is transferred to the boiler by all three modes of heat transfer i.e. conduction, convection and radiation.
- ❖ Major types of boilers are: (i) fire tube boiler and (ii) water tube boiler
Based on Pressures of operation: (i) High Pressure boiler and (ii) Low Pressure boiler
- ❖ Generally water tube boilers are used for electric power stations.

Differences Between Low Pressure boiler, High Pressure boiler:

S.NO	HIGH PRESSURE BOILER	LOW PRESSURE BOILER
1.	In this type of boilers, steam pressure is above 15 psi.	In this type of boilers, steam pressure never exceeds 15 psi.
2.	Hot water pressure always exceeds 260 psi	Water pressure never exceeds 260 psi
3.	Temperature will always be greater than 250oC250oC	Temperature will never be greater than 250oC250oC.
4	For the safety due to high working pressures, these boilers are required to be monitored all the time.	Since these boilers work on low working pressure, they don't need continuous monitoring.
5	Applications involve production of steam in industries and in power plant	These are used for water heating and steam from it used to heat rooms.
6	For similar size, the output is relatively high.	For similar size, the output is relatively less.
7	High maintenance is required	Low maintenance required
8	Operating costs are relatively higher	Operating costs are relatively low.

BOILER MOUNTINGS AND ACCESSORIES:

Boiler Mounting:

Boiler Mountings are the components generally mounted on the surface of the boiler to have safety during operation. These are the essential parts of the boiler, without which the boiler operation is not possible.

The following are the important mountings of the boiler:

1. Water level indicator
2. Safety valve
3. Pressure gauge
4. Steam stop valve
5. Feed check valve
6. Main hole

1. Water level indicator: this device indicates the exact level of water in the boiler tube.
2. Safety valve: it is a mechanical device used to safeguard the boiler, in case the pressure inside the boiler rises above its normal working atmosphere.
3. Pressure gauge: the pressure gauge commonly used is the bourdon pressure gauge mounted on the front top of the boiler shell.
4. Steam stop valve: the function of the steam stop valve is to stop or open the steam supply from the boiler to the point of application.
5. Feed check valve: a valve placed at the boiler end to regulate the flow of water.
6. Main hole: it is the opening provided of cleaning or inspection.

Boiler Accessories:

These are those devices which are installed with a boiler and its neighbouring area to increase the efficiency of the boiler. These are not the essential part of the boiler and thus without installing these devices, the boiler operation can be accomplished though at a lower efficiency. The following are the important accessories of the boiler are:

1. Feed water pump
2. Injector
3. Pressure reducing valve
4. Economiser
5. Air pre heater

6. Super heater
 7. Steam drier or separator
 8. Steam trap
1. Feed water pump: the feed water pump is used to feed water to the boiler.
 2. Injector: function of the injector is the same as that of feed pump i.e.; to deliver feed water to boiler under pressure.
 3. Pressure reducing valve: the function of the pressure reducing valve is to maintain constant pressure on its delivery side of the valve irrespective of fluctuating demand of steam from the boiler.
 4. Economiser: a considerable quantity of heat of the plant is carried away by flue gases, so to utilise these waste heat. An economiser is placed in the path of the flue gases to preheat the feed water.
 5. Air pre – heater: the function of the pre- heater is to heat the air before it enters the combustion chamber.
 6. Super heater: super pre-heater is used for stationary boiler. The function of the super heater is to convert dry saturated steam into super-heated at the desired temperature.
 7. Steam drier: the function of the steam drier or separator is to separate water particles from steam before it is supplied to the point of application.

DRAUGHT:

- The combustion in the boiler requires supply of sufficient quality of air and removal of exhaust gases
- The Circulation of air is caused by difference of pressure is known as draught. Thus draught is the differential in pressure between the two points.

A draught tube may be

1. Natural Draught
2. Mechanical Draught

Natural Draught:

- A natural Draught is provided by the chimney or stack.
- Natural draught has its limitation. Modern plants has high rate of heat transfer and Draught losses are very high. In view of this Natural draught is used only for small boilers.

Mechanical Draught:

Modern large size plants use very large size of boilers of capacity above 1000,000 kg per hour. Such boiler needs tremendous volume of air (around 200000 m³) per minute. A Chimney provides this. Therefore mechanical draught is used.

- In a mechanical draught the system the movement air is due to the action of fan. A Mechanical Draught consists of forced Draught or induced draught or both.
- In forced draught system the fan is installed near the boiler .the fan force the air through the furnace, economizer, air preheater and chimney. The pressure of air, throughout the system, is above atmospheric and air is forced to flow through the system.
- In an induced draught system the, the fan is installed near the base of the chimney .The burnt gases are sucked out from the boiler, thus reducing the pressure inside the boiler. To less than atmosphere. This induces fresh air to enter the furnace.
- A mechanical Draught need additional capital investment and maintenance. But it required for proper operation of modern power plant. In super thermal power plant, each boiler may use two forced fans and two induced fan.

STEAM NOZZLES AND TURBINES

Flow of steam through nozzles :

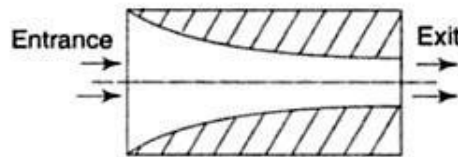
- The flow of steam through nozzles may be regarded as adiabatic expansion.
- The steam has a very high velocity at the end of the expansion, and the enthalpy decreases as expansion takes place.
- Friction exists between the steam and the sides of the nozzle; heat is produced as the result of the resistance to the flow.
- The phenomenon of super saturation occurs in the flow of steam through nozzles. This is due to the time lag in the condensation of the steam during the expansion.

Types of Nozzles:

1. Convergent Nozzle
2. Divergent Nozzle
3. Convergent-Divergent Nozzle

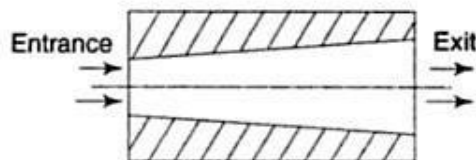
Convergent Nozzle

A typical convergent nozzle is shown in the Fig.16.1. In a convergent nozzle, the cross sectional area decreases continuously from its entrance to exit. It is used in a case where the back pressure is equal to or greater than the critical pressure ratio (which will be defined later).



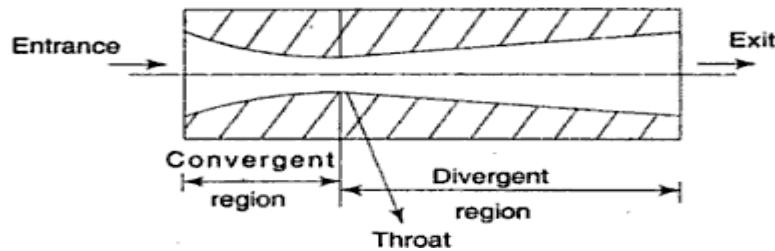
Divergent nozzle

The cross sectional area of divergent nozzle increases continuously from its entrance to exit. It is used in a case where the back pressure is less than the critical pressure ratio.



Convergent-Divergent Nozzle

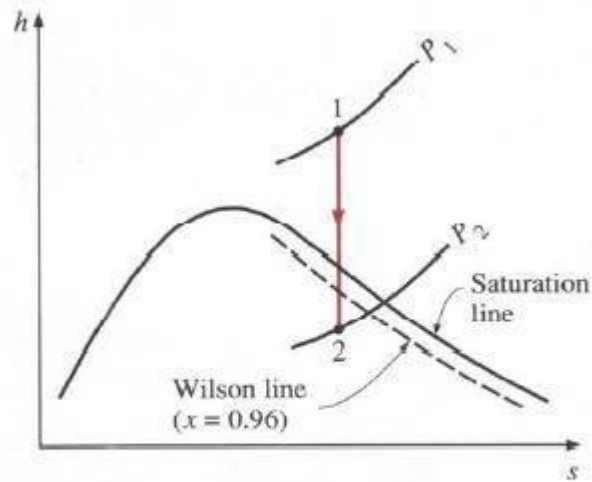
In this case, the cross sectional area first decreases from its entrance to throat and then increases from throat to exit. This case is also used in the case where the back pressure is less than the critical pressure. Also, in present day application, it is widely used in many **types** of **steam** turbines.



Supersaturated flow or Metastable flow in Nozzles :

As steam expands in the nozzle, its pressure and temperature drop, and it is expected that the steam start condensing when it strikes the saturation line. But this is not always the case. Owing to the high velocities, the residence time of the steam in the nozzle is small, and there may not sufficient time for the necessary heat transfer and the formation of liquid droplets. Consequently, the condensation of steam is delayed for a little while. This phenomenon is known as supersaturation, and the steam that exists in the wet region without containing any liquid is known as supersaturated steam.

The locus of points where condensation will take place regardless of the initial temperature and pressure at the nozzle entrance is called the Wilson line. The Wilson line lies between 4 and 5 percent moisture curves in the saturation region on the h-s diagram for steam, and is often approximated by the 4 percent moisture line. The supersaturation phenomenon is shown on the h-s chart below:



The h - s diagram for the isentropic expansion of steam in a nozzle.

Effects of Supersaturation:

The following are the effects of supersaturation in a **nozzle**.

- The temperature at which the supersaturation occurs will be less than the saturation temperature corresponding to that pressure. Therefore, the density of supersaturated steam will be more than that of equilibrium condition which gives the increase in the mass of steam discharged.
- Supersaturation increases the specific volume and entropy of the steam.
- Supersaturation reduces the heat drop. Thus the exit velocity of steam is reduced.
- Supersaturation increases the dryness fraction of the steam.

Critical Pressure Ratio:

The critical pressure ratio is the pressure ratio which will accelerate the flow to a velocity equal to the local velocity of sound in the fluid.

Critical flow nozzles are also called **sonic chokes**. By establishing a shock wave the sonic choke establish a fixed flow rate unaffected by the differential pressure, any fluctuations or changes in downstream pressure. A sonic choke may provide a simple way to regulate a gas flow.

The ratio between the critical pressure and the initial pressure for a nozzle can expressed as $p_c /$

$$p_1 = \left(\frac{2}{n+1} \right)^{n/(n-1)}$$

where

p_c = critical pressure (Pa)

p_1 = inlet pressure (Pa)

n = index of isentropic expansion or compression – or polytropic constant

For a perfect gas undergoing an adiabatic process the index – n – is the ratio of specific heats – k = c_p / c_v . There is no unique value for – n . Values for some common gases are

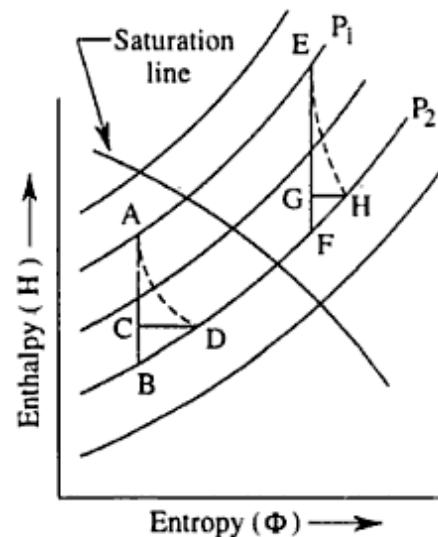
- Steam where most of the process occurs in the wet region : $n = 1.135$
- Steam superheated : $n = 1.30$
- Air : $n = 1.4$
- Methane : $n = 1.31$
- Helium : $n = 1.667$

Effect of Friction on Nozzles:

- 1) Entropy is increased.
- 2) Available energy is decreased.
- 3) Velocity of flow at throat is decreased.
- 4) Volume of flowing steam is decreased.
- 5) Throat area necessary to discharge a given mass of steam is increased.

Most of the **friction** occurs in the diverging part of a convergent-divergent **nozzle** as the length of the converging part is very small. The **effect of friction** is to reduce the available enthalpy drop by about 10 to 15 per cent. The velocity of steam will be then $V_2 = 44.72\sqrt{K(H_1 - H_2)}$ where K is the coefficient which allows for **friction** loss. It is also known as **nozzle efficiency** (η_n)

$$\therefore V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n}$$



Velocity of steam at nozzle exit:

$$V_2^2 = 2000(H_1 - H_2) + V_1^2 \quad \therefore \quad V_2 = \sqrt{2000(H_1 - H_2) + V_1^2}$$

As the velocity of steam entering the **nozzle** is very small, V_1 can be neglected.

$$\therefore \quad V_2 = \sqrt{2000(H_1 - H_2)} = 44.72\sqrt{(H_1 - H_2)} \text{ m/s}$$

· If frictional losses are taken into account then

$$V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n} \text{ m/s}$$

Mass of steam discharged through a nozzle:

$$m = A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1} \right)^{\frac{n+1}{n}} \right]}$$

Condition for maximum discharge through nozzle:

The nozzle is always designed for maximum discharge

$$\frac{m}{A} = \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1} \right)^{\frac{n+1}{n}} \right]}$$

The mass flow per unit area will be maximum at the throat because the throat area is minimum.

It is seen from the above equation that the discharge through a **nozzle** is a function of $\frac{P_2}{P_1}$ only, as the expansion index is fixed according to the steam supplied to the **nozzle**.

Therefore, $\frac{m}{A}$ is maximum when

$$\left[\left(\frac{P_2}{P_1} \right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1} \right)^{\frac{n+1}{n}} \right] \text{ is minimum}$$

Values for maximum discharge:

$$m = A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1} \right)^{\frac{n+1}{n}} \right]}$$

we know $\frac{P_2}{P_1} = \left(\frac{2}{n+1} \right)^{\frac{n}{n-1}}$

Putting the value of $\frac{P_2}{P_1}$ in the above equation

$$m_{\max} = A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \left[\left(\frac{2}{n+1} \right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}} \right]}$$

$$m_{\max} = A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}} \left[\left(\frac{2}{n+1} \right)^{\frac{2}{n-1} - \frac{n+1}{n-1}} - 1 \right]}$$

$$= A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}} \left[\left(\frac{2}{n+1} \right)^{\frac{1-n}{n-1}} - 1 \right]}$$

$$= A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}} \left[\left(\frac{2}{n+1} \right)^{-1} - 1 \right]}$$

$$= A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}} \left[\frac{n+1}{2} - 1 \right]}$$

$$= A \sqrt{2000 \frac{n}{n-1} \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}} \left(\frac{n-1}{2} \right)}$$

$$m_{\max} = A \sqrt{1000n \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}}}$$

Where P_1 is the initial pressure of the steam in kpa and v_1 is the specific volume of the steam in m^3/kg at the initial pressure.

Sample Problems on Steam Nozzle:

1. Steam at pressure of 1.5 Mpa and temperature of 260°C expands isentropically in a steam nozzle to a pressure 500 kPa with an actual enthalpy drop of 200 KJ/kg. If the nozzle outlet area is approximately 4 cm² and mass flow rate is 10 kg/s, calculate the number of nozzles required and adjust the outlet dimensions to suit this number.

2. Dry saturated steam at 10 bar pressure enters a convergent nozzle which is having 10mm throat diameter and 12 mm divergent portion length. Determine the diameter at the nozzle exit and cone angle of the divergent portion so that the steam may leave at 1 bar pressure. Assume the effects of Friction are negligible.

3. Steam at pressure of 22 bar and temperature 300°C is supplied to a group of five nozzles at the rate of 4.5 kg/s. The exit pressure of steam is 3 bar. Determine the following:

- i. The dimensions of the nozzles of rectangular cross section with an aspect ratio of 3:1. Neglect the friction effect and assume the expansion is metastable.
- ii. Degree of Under cooling and Supersaturation.
- iii. The loss in available heat drop due to irreversibility
- iv. Change in entropy

06/04/18

1) A nozzle expands steam from 14 bar, 300°C to 6 bar. If the flow rate is 1 kg/sec, find the throat and area. What should be the coefficient of velocity if exit velocity is 550 m/sec

$$P_1 = 6 \text{ bar}$$

$$\text{We have } \frac{P_2}{P_1} = \left(\frac{2}{\gamma+1} \right)^{\gamma/\gamma-1}$$

As the steam is superheated,

$$\gamma = 1.3$$

$$P_2 = 7.63 \text{ bar}$$

From mollier chart

$$3041.6$$

$$h_1 = 3041.6 \text{ kJ/kg}$$

$$h_2 = 2885 \text{ kJ/kg}$$

$$t_2 = 215^\circ\text{C}$$

$$C_2 = 44.72 \sqrt{h_1 - h_2} = 559.625 \text{ m/s}$$

$$\eta = \frac{A_2 C_2}{V_2}$$

$$A_2 = \frac{0.2855}{559.6} \times 10^4 = 5.108 \text{ cm}^2$$

From mollier chart,

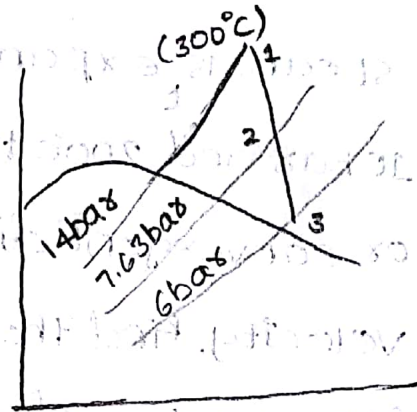
$$h_3 = 2844 \text{ kJ/kg}$$

$$t_3 = 195^\circ\text{C}$$

From steam tables,

$$v_2 = 0.2855 \text{ m}^3/\text{kg}$$

$$v_3 = 0.352 \text{ m}^3/\text{kg}$$



$$C_3 = 44.72 \sqrt{h_1 - h_3}$$

$$= 628.64 \text{ mm/s}$$

$$\therefore \sigma_3 = \frac{A_3 C_3}{V_3} \Rightarrow A_3 = 5.599 \text{ cm}^2$$

2) steam is expanded in a set of nozzles from 10 bar and 200°C to 5 bar. Is the nozzle convergent or convergent divergent. Neglecting the initial velocity, find the min. area of the nozzles to flow 3 kg/sec of steam under the given Assume expansion of steam is isentropic. ~~coefficient of discharge is 0.95~~ calculate the actual throat area.

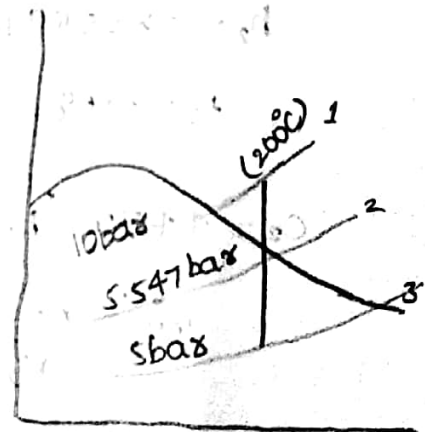
$$P_1 = 10 \text{ bar},$$

We have, $\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{n/n-1}$

As the steam is superheated,

$$\sigma = 1.3$$

$$P_2 = 5.457 \text{ bar}$$



Since the exit p_r is $p_{\text{bar}} < p_{\text{critical}}$,
the nozzle is convergent divergent nozzle

From moillier chart,

$$h_1 = 2829 \text{ kJ/kg}$$

$$h_2 = 2710 \text{ kJ/kg}$$

$$x_2 = 0.98$$

From steam tables

$$\text{for } P_2 = 5.547$$

$$v_{g2} = 0.344 \text{ m}^3/\text{kg}$$

$$v_2 = 0.3371 \text{ m}^3/\text{kg}$$

$$C_2 = 44.72 \sqrt{h_1 - h_2} = 487.85 \text{ m/s}$$

$$\dot{m} = \frac{A_2 C_2}{v_2}$$

$$A_2 = 20.729 \text{ cm}^2$$

3) An adiabatic steam nozzle is to be designed for a discharge of 10 kg/s of steam from 10 bar and 400°C to a watt pressure of 1 bar. The nozzle efficiency is 0.92 and the frictional loss is assumed to take place in the diverging portion of the nozzle only. Assume a critical pressure ratio of 0.54572 (P_2/P_1). Determine the throat and exit areas

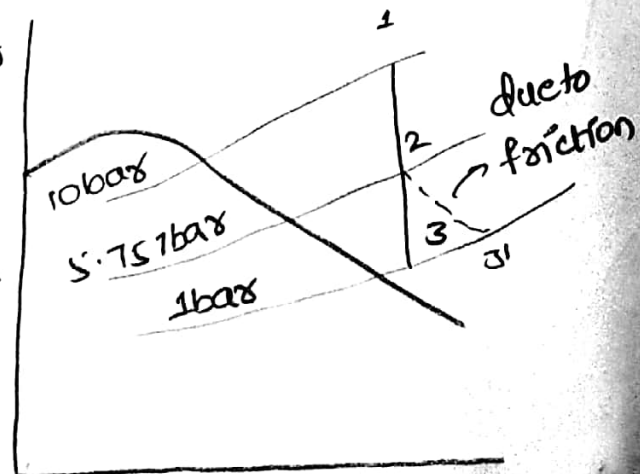
$$\frac{P_2}{P_1} = 0.5457, P_2 = 5.457 \text{ bar}$$

$$h_1 = 3270$$

$$h_2 = 3080$$

$$h_3 = 2710$$

From
moillier
diagram



From steam tables,

$$v_2 = 0.5$$

$$C_2 = 44.72 \sqrt{h_1 - h_2} = 598.3 \text{ m/s}$$

$$C_3' = 44.72 \sqrt{\eta_n(h - h_3)} = 1000.4 \text{ m/s}$$

$$\eta_n = \frac{A_2 V_2}{C_2}$$

$$= 83.56 \text{ m}^3/\text{kg}$$

$$\eta_n = \frac{h_1 - h_3'}{h_1 - h_3} \Rightarrow h_3 = 2763.52$$

4) Calculate the throat and exit diameters of a cd nozzle which will discharge 800 kg steam per hr from a pr. of 8 bar superheated to a chamber of 1.05 bar. Friction loss in the diverging part of the nozzle may be taken as 0.15 of the total enthalpy drop.

$$h_1 =$$

$$\frac{P_2}{P_1} = 0.5457$$

$$P_2 = 0.5457 \times 8 = 4.38 \text{ bar}$$

From mollier chart,

$$h_1 = 2889 \text{ kJ/kg}$$

$$h_2 = 2763 \text{ kJ/kg}$$

$$v_2 = 0.44 \text{ m}^3/\text{kg}$$

$$C_2 = 44.72 \sqrt{h_1 - h_2} = 501.99 \text{ m/s}$$

$$0.85 = \frac{h_1 - h_{3'}}{h_1 - h_3}$$

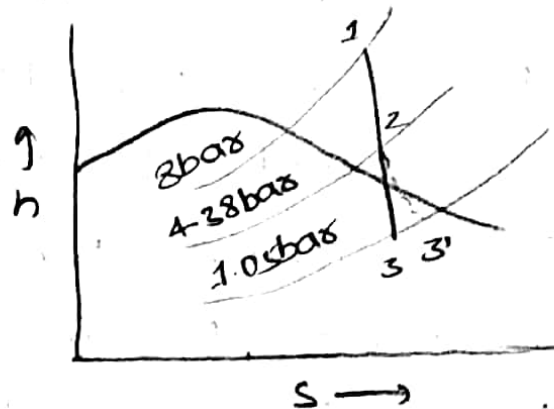
$$h_{3'} = 2622.1 \text{ kJ/kg}$$

$$C_{3'} = 44.72 \sqrt{h_1 - h_{3'}} = 730.6 \text{ m/s}$$

From mollier chart $x_{3'} = 0.965$

From steam tables,

$$v_{g3'} = 1.185 \text{ m}^3/\text{kg}$$



$$\dot{m} = \frac{800}{3600} = 0.2228 \text{ kg/s}$$

$$A_2 = \frac{\dot{m} v_2}{C_2}$$

$$\Rightarrow D_2 = 1.5943 \text{ cm}$$

$$v_3' = x_3 v_{g3}' = 1.14 \text{ m}^3/\text{kg}$$

$$m_3' = \frac{A_3 C_3'}{v_3}$$

$$\Rightarrow D_3 = 2.13 \text{ cm}$$

5) Nozzles of a DeLaval steam turbine are supplied with dry saturated steam at a pressure of 9 bar. The pressure at outlet is 1 bar. The turbine has 2 nozzles with a throat diameter as 2.5 mm. Assuming nozzle efficiency to be 90%, and that of the turbine motor is 35%. Find the mass of steam used per hr. and the power developed.

$$\eta = 1.135$$

$$P_2 = 5.22 \text{ bar}$$

From Mollier chart

$$h_1 = 2770 \text{ kJ/kg}$$

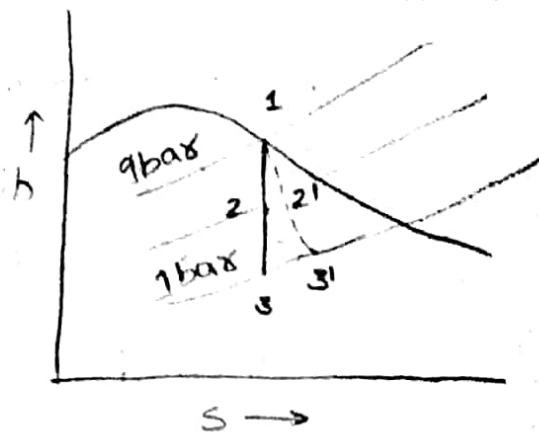
$$h_2 = 2670 \text{ kJ/kg}$$

$$h_3 = 2400 \text{ kJ/kg}$$

$$\eta_2' = 0.96, \eta_3' = 0.88$$

$$\frac{h_1 - h_2'}{h_1 - h_2} = \eta_n \Rightarrow h_2' = 2680$$

$$\frac{h_1 - h_3'}{h_1 - h_3} = \eta_n \Rightarrow h_3' = 2437 \text{ kJ/kg}$$



From steam tables,

$$v_{g2} = 0.361$$

$$v_2' = x_2 v_{g2} =$$

$$C_2' = 44.72 \sqrt{h_1 - h_2'} = 424.2 \text{ m/s}$$

$$A_2 = (4.9 \times 10^{-6}) \times 2 = 9.82 \times 10^{-6}$$

$$\dot{m} = 0.012 \text{ kg/s}$$

$$P = (h_1 - h_3') \dot{m} = 4 \text{ kW}$$

Steam boiler can be defined as a closed vessel in which steam is produced from water by combustion of fuel.

Requirements of Good boiler:

- (i) max. quantity of steam at req. pr., temp and quality
- (ii) Light in weight and compact
- (iii) safe in working with min joints
- (iv) Initial, installation and maintenance cost should be low
- (v) Capable of quick start and should meet fluctuations in demand
- (vi) All parts should be accessible for inspection
- (vii) min. refractory material should be used
- (viii) Heating surface should be free from contaminants.

Classification of boilers:

(i) Depending on boiler axis orientation

a) Horizontal b) Vertical c) Inclined

(ii) Depending on tube

a) ~~fire-tube~~ fire-tube boiler ex: Cochran boiler, Lancashire, locomotive

b) water-tube boiler ex: Babcock and Wilcox, Stirling

a) In the fire-tube boilers, hot gases are inside the tube and water surrounds the tubes

b) In the water-tube boilers, water is inside the tubes and the hot gases surround them

(iii) Depending on furnace location:

a) Externally fixed boilers

b) Internally fixed boilers

(iv) Depending on circulation of water:

a) Natural circulation type of boilers

b) Forced circulation type of boilers

(v) Depending on operating pressure

a) High pr. boilers b) Low pr. boilers

ex: Babcock and Wilcox,
Stirling

ex: Cornish, Cochran,
Lancashire, Locomotive

(vi) Depending on mobility:

a) stationary b) Mobile

(vii) Depending on no. of tubes

a) single tube b) multi tube

<u>Particulars</u>	<u>Fire tube boilers</u>	<u>Water tube boilers</u>
1) Position of water and water hot gases	hot water gases inside the tubes and water outside the tubes	Water inside the tubes and hot gases outside the tubes
2) Mode of firing	Internally fired	Generally externally fired.
3) Operating pressures	Operating pr. limited to 20 bar	Can work upto a high pr. of 100 bar
4) Rate of steam production	Lower	Higher
5) Suitability	Not suitable for large power plants	Suitable for large power plants
6) Risk of bursting	Involves lesser risk on explosion due to lower pr.'s	Involves more risk on bursting due to higher pressures
7) Failure of feed water	Less harmful	More harmful
8) Chances of explosion	Less	More
9) Treatment of water	Not so necessary	More necessary

10) Maintenance

comparatively less

comparatively more

11) Accessibility of various parts

Various parts are not so easily accessible for cleaning, inspection

Easily accessible

12) Efficiency

Low

More

Fire tube boilers.

Cochran

~~Cochran~~ boiler

- Vertical multi-tubular

- Internally fired

- Stationary

Furnace (Hemispherical)

↳ strongest

↳ maximum absorption of radiant heat

Adv,

can use any kind of fuel

ex: wood, paddy husk,

oil fuel, coal

Steam production rates

20 kg/hr to 3000 kg/hr

Steam pressures: 20 bar

$\eta_{\text{Cochran}} = 70-75\%$ (max)

Water tube boilers:

- Horizontal

Babcock and Wilcox boilers:

- Natural circulation boiler.

Steam capacity: 40,000 t/h (max)

Steam pressure: 40 bar to 125 bar
 $\approx 100 \text{ bar}$

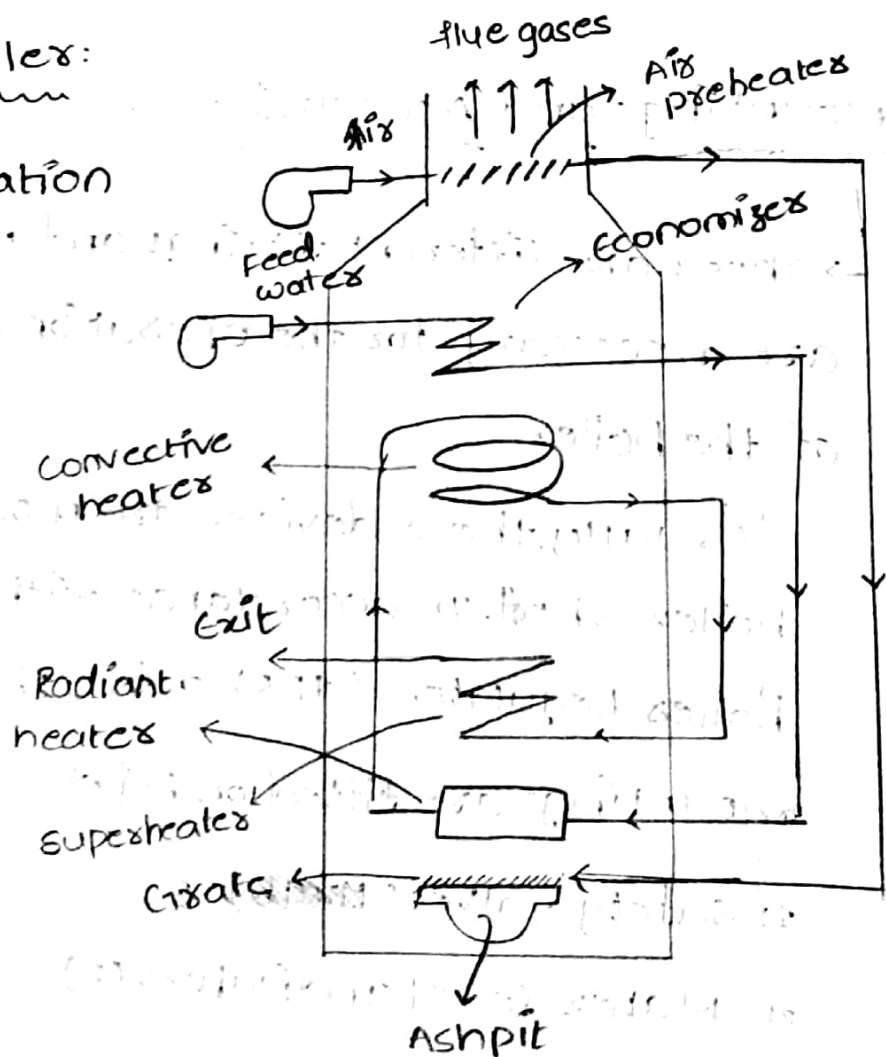
$\theta \approx 15^\circ$

11/04/2018

High pressure boilers:

Benson boiler:

- Forced circulation
- Maintenance is very high



capacity: 150 Tonnes/hr

pressure = 200 bar

temperature = 650°C

In the Benson boiler, there is no drum due to which the weight of the boiler reduces (almost 20%) and becomes compact and portable. Also, within 10 mins Benson's boiler goes to peak value

The only problem with the Benson boiler is the maintenance of its parts.

Mountings and Accessories:

↳ These are different fittings and devices which are necessary for the operation and safety of the boiler.

Usually, these devices are mounted on the boiler shell. In accordance with the Indian Boilers Regulation (IBR) Idea, the following mountings are fitted on boiler

- 1) safety valves (~~more~~): 2
- 2) Water level indicators (2)
- 3) Pressure gauge
- 4) steam stop valve

5) Feed check valve

6) Blow off cock

7) Manhole and mudhole

Water level indicator:

The function of water level indicator is to indicate the level of water.

Pressure gauge:

It is to measure the pressure inside the vessel.

Safety valves:

Function of safety valve is to release the excess steam when the steam inside the boiler exceeds the rated pressure.

As per IBR, every boiler must be fitted with at least 2 safety valves.

Various types of safety valves

(i) Dead-weight safety valve

(ii) Lever safety valve

(iii) Spring loaded safety valve

(iv) High steam and low water safety valve

Fusible plug:

The function of the fusible plug is to protect the boiler against damage due to overheating for low water level.

Blow off cock:

The function of blow off cock is to discharge the portion of water when the boiler is in operation to blow out mud, scale, sediments periodically.

Also, it can empty the boiler when necessary for cleaning, inspection and repair.

Feed check valve:

Function of it is to control the supply of water to the boiler and to prevent the escaping of water from boiler.

Boiler Accessories:

These are the auxiliary plates -

required for steam boilers for their proper operation and for increase in efficiency

Some boiler accessories are:

- (i) Air preheaters
- (ii) Economizers

(iii) super heater

(iv) steam separator

(v) steam trap etc.

Economizer: An economizer is a device in which the waste heat of the flue gases is utilized for heating the feed water.

Air-preheater: The function of this is to increase the temp of air before it enters the furnace.

Generally it is placed after the economizer.

Super-heater: The function of this is to increase the temp of the steam above its saturation point. efficiency increases due to superheating.

There are 2 types of super heaters:

Convective super heater, Radiant super heater

Steam trap: The function of this is to drain away automatically the condensed steam from the steam pipes, from the steam jackets and steam separators.

Steam separator: The function of this is to remove the entrained water particles from the steam.

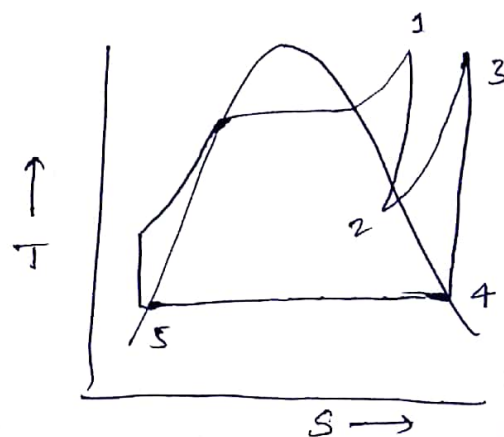
Vacuum efficiency:

It is defined as the ratio of actual vacuum to the max. obtainable vacuum.

Condensor efficiency:

It is defined as the ratio of the difference b/w outlet and inlet temperature of cooling water to the difference b/w the temp corresponding to the vacuum in the condenser and the inlet temperature of cooling water.

Reheat factor: It is defined as the ratio of cumulative isentropic heat drop to overall isentropic heat drop



$$\text{Reheat factor} = \frac{(h_1 - h_2) + (h_3 - h_4)}{h_1 - h_4}$$

Catalytic converter

Two way
catalytic converter

(reduces the concⁿ of
HC and CO)

Three way
catalytic converter

(reduces the concⁿ
of HC, CO, NO_x)

Non-exhaust emissions

Blow by losses:

PCV ← positive crankcase ventilation

