

ME6404 THERMAL ENGINEERING

UNIT I

GAS POWER CYCLES

CONTENTS

TECHNICAL TERMS

- 1.1 The Otto Cycle
- 1.2 Diesel Cycle
- 1.3 Solved Problems
- 1.4 Dual Cycle
- 1.5 The Brayton Cycle
- 1.6 Actual PV diagram of four stroke engine
- 1.7 Solved Problems
- 1.8 Two Marks University Questions
- 1.9 University Essay Questions

TECHNICAL TERMS:**1. Gas Power Cycles:**

Working fluid remains in the gaseous state through the cycle. Sometimes useful to study an idealised cycle in which internal irreversibilities and complexities are removed. Such cycles are called: Air Standard Cycles

2. The mean effective pressure (MEP)

A fictitious pressure that, if it were applied to the piston during the power stroke, would produce the same amount of net work as that produced during the actual cycle.

3. Thermodynamics:

is the science of the relations between heat, work and the properties of system

4. Boundary:

System is a fixed and identifiable collection of matter enclosed by a real or imaginary surface which is impermeable to matter but which may change its shape or volume. The surface is called the boundary

5. Surroundings:

Everything outside the system which has a direct bearing on the system's behaviour.

6. Extensive Property:

Extensive properties are those whose value is the sum of the values for each subdivision of the system, eg mass, volume.

7. Intensive Property:

Properties are those which have a finite value as the size of the system approaches zero, eg pressure, temperature, etc.

8. Equilibrium:

A system is in thermodynamic equilibrium if no tendency towards spontaneous change

exists within the system. Energy transfers across the system disturb the equilibrium state of the system but may not shift the system significantly from its equilibrium state if carried out at low rates of change. I mentioned earlier that to define the properties of a

system, they have to be uniform throughout the system.

Therefore to define the state of system, the system must be in equilibrium.

(Inequilibrium of course implies non-uniformity of one or more properties).

9. **Process:**

A process is the description of what happens when a system changes its state by going through a succession of equilibrium states.

10. **Cyclic**

Process:

A cyclic process is one for which the initial and final states of the system are identical.

11. **Isentropic process:**

is one in which for purposes of engineering analysis and calculation, one may assume that the process takes place from initiation to completion without an increase or decrease in the entropy of the system, i.e., the entropy of the system remains constant.

12. **Isentropic flow:**

An **isentropic flow** is a flow that is both adiabatic and reversible. That is, no heat is added to the flow, and no energy transformations occur due to friction or dissipative effects. For an isentropic flow of a perfect gas, several relations can be derived to define the pressure, density and temperature along a streamline.

13. **Adiabatic heating**

Adiabatic heating occurs when the pressure of a gas is increased from work done on it by its surroundings, e.g. a piston. Diesel engines rely on adiabatic heating during their compression

14. Adiabatic cooling:

Adiabatic cooling occurs when the pressure of a substance is decreased as it does work on its surroundings. Adiabatic cooling occurs in the Earth's atmosphere with orographic lifting and lee waves. When the pressure applied on a parcel of air decreases, the air in the parcel is allowed to expand; as the volume increases, the temperature falls and internal energy decreases

UNIT-I

GAS POWER CYCLES

1.1 The Otto Cycle

The Otto cycle, which was first proposed by a Frenchman, Beau de Rochas in 1862, was first used on an engine built by a German, Nicholas A. Otto, in 1876. The cycle is also called a constant volume or explosion cycle. This is the equivalent air cycle for reciprocating piston engines using spark ignition. Figures 1 and 2 show the P-V and T-s diagrams respectively.

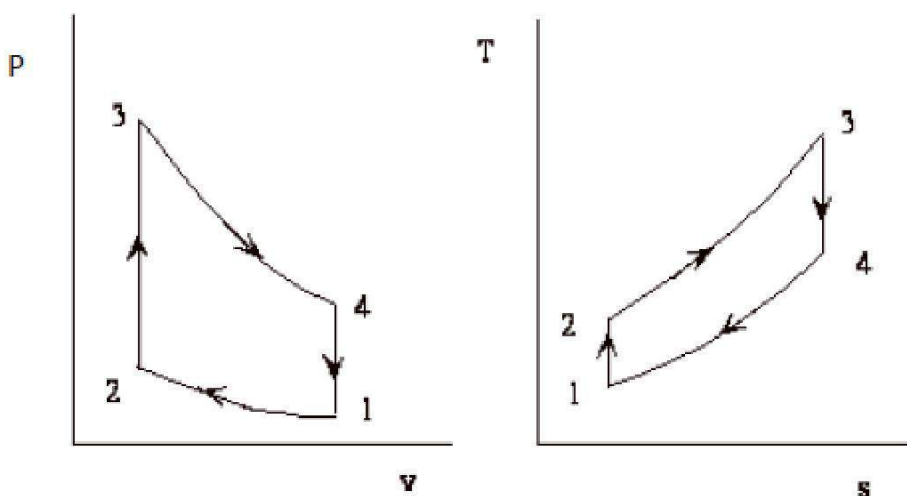


Fig 1.1 P-V Diagram of Otto Cycle. Fig 1.2 T-S Diagram of Otto Cycle.

At the start of the cycle, the cylinder contains a mass M of air at the pressure and volume indicated at point 1. The piston is at its lowest position. It moves upward and the gas is compressed isentropically to point 2. At this point, heat is added at constant volume which raises the pressure to point 3. The high pressure charge now expands isentropically, pushing the piston down on its expansion stroke to point 4 where the charge rejects heat at constant volume to the initial state, point 1.

The isothermal heat addition and rejection of the Carnot cycle are replaced by the constant volume processes which are, theoretically more plausible, although in practice, even these processes are not practicable.

The heat supplied, Q_s , per unit mass of charge, is given

by $cv(T_3 - T_2)$

The heat rejected, Q_r per unit mass of charge is given

by $cv(T_4 - T_1)$

and the thermal efficiency is given by

$$\eta = 1 - \frac{1}{r^{\gamma-1}}$$

$$\begin{aligned}\eta_{th} &= 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)} \\ &= 1 - \frac{T_1}{T_2} \left\{ \frac{\left(\frac{T_4}{T_1} - 1\right)}{\left(\frac{T_3}{T_2} - 1\right)} \right\}\end{aligned}$$

$$\text{Now } \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \frac{T_4}{T_3}$$

$$\text{And since } \frac{T_1}{T_2} = \frac{T_4}{T_3} \text{ we have } \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Hence, substituting in Eq. 3, we get, assuming that r is the compression ratio V_1/V_2

$$\begin{aligned}\eta_{th} &= 1 - \frac{T_1}{T_2} \\ &= 1 - \left(\frac{V_2}{V_1}\right)^{\gamma-1} \\ &= 1 - \frac{1}{r^{\gamma-1}}\end{aligned}$$

In a true thermodynamic cycle, the term expansion ratio and compression ratio are synonymous. However, in a real engine, these two ratios need not be equal because of the valve timing and therefore the term expansion ratio is preferred sometimes.

Equation 4 shows that the thermal efficiency of the theoretical Otto cycle increases with increase in compression ratio and specific heat ratio but is independent of the heat added (independent of load) and initial conditions of pressure, volume and temperature.

Mean effective pressure and air standard efficiency

It is seen that the air standard efficiency of the Otto cycle depends only on the compression ratio. However, the pressures and temperatures at the various points in the cycle and the net work done, all depend upon the initial pressure and temperature and the heat input from point 2 to point 3, besides the compression ratio.

A quantity of special interest in reciprocating engine analysis is the mean effective pressure. Mathematically, it is the net work done on the piston, W , divided by the piston displacement volume, $V_1 - V_2$. This quantity has the units of pressure. Physically, it is that constant pressure which, if exerted on the piston for the whole outward stroke, would yield work equal to the work of the cycle. It is given by

$$\text{Mean Effective Pressure (Pm)} = P_1 r^{\frac{1}{\gamma}} \frac{[(\gamma-1)(r^{\gamma-1}-1)]}{[(\gamma-1)(r-1)]}$$

$$mep = \frac{W}{V_1 - V_2}$$

$$= \frac{\eta Q_{2-3}}{V_1 - V_2}$$

where Q_{2-3} is the heat added from points 2 to 3. Work done per kg of air

$$W = \frac{P_3 V_3 - P_4 V_4}{\nu - 1} - \frac{P_2 V_2 - P_1 V_1}{\nu - 1} = mep V_s = P_m (V_1 - V_2)$$

$$mep = \frac{1}{(V_1 - V_2)} \left[\frac{P_3 V_3 - P_4 V_4}{\nu - 1} - \frac{P_2 V_2 - P_1 V_1}{\nu - 1} \right]$$

The pressure ratio P_3/P_2 is known as explosion ratio r_p

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2} \right)^\nu = r^\nu \Rightarrow P_2 = P_1 r^\nu,$$

$$P_3 = P_2 r_p = P_1 r^\nu r_p,$$

$$P_4 = P_3 \left(\frac{V_3}{V_4} \right)^\nu = P_1 r^\nu r_p \left(\frac{V_2}{V_1} \right)^\nu = P_1 r_p$$

$$\frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = r$$

$$\therefore V_s = V_c (r - 1)$$

Substituting the above values in Eq 5A

$$mep = P_1 \frac{r(r_p - 1)(r^{\gamma-1} - 1)}{(r - 1)(\gamma - 1)}$$

$$V_1 - V_2 = V_1 \left(1 - \frac{V_2}{V_1} \right)$$

$$= V_1 \left(1 - \frac{1}{r} \right)$$

Here r is the compression ratio,

V_1/V_2 From the equation of state:

$$V_1 = M \frac{R_0}{m} \frac{T_1}{p_1}$$

R_0 is the universal gas constant Substituting for V_1 and for $V_1 - V_2$,

$$mep = \eta \frac{Q_{2-3} \frac{p_1 m}{MR_0 T_1}}{1 - \frac{1}{r}}$$

The quantity Q_{2-3}/M is the heat added between points 2 and 3 per unit mass of air (M is the mass of air and m is the molecular weight of air); and is denoted by Q' , thus

$$mep = \eta \frac{Q' \frac{p_1 m}{R_0 T_1}}{1 - \frac{1}{r}}$$

We can non-dimensionalize the mep by dividing it by p_1 so that we can obtain the following equation

$$\frac{mep}{p_1} = \eta \left[\frac{1}{1 - \frac{1}{r}} \right] \left[\frac{Q' m}{R_0 T_1} \right]$$

Since $\frac{R_0}{m} = c_v (\gamma - 1)$, we can substitute it in Eq. 25 to get

$$\frac{mep}{p_1} = \eta \frac{Q'}{c_v T_1} \frac{1}{\left[1 - \frac{1}{r} \right] [\gamma - 1]}$$

The dimensionless quantity mep/p_1 is a function of the heat added, initial temperature, compression ratio and the properties of air, namely, c_v and γ . We see that the mean effective pressure is directly proportional to the heat added and inversely proportional to the initial (or ambient) temperature. We can substitute the value of η from Eq. 8 in Eq. 14 and obtain the value

of mep/p_1 for the Otto cycle in terms of the compression ratio and heat added. In terms of the pressure ratio, p_3/p_2 denoted by r_p we could obtain the value of mep/p_1 as follows:

$$\frac{mep}{p_1} = \frac{r(r_p - 1)(r^{\gamma-1} - 1)}{(r - 1)(\gamma - 1)}$$

We can obtain a value of r_p in terms of Q' as follows:

$$r_p = \frac{Q'}{c_v T_1 r^{\gamma-1}} + 1$$

Choice of Q'

We have said that,

$$Q' = \frac{Q_{2-3}}{M}$$

M is the mass of charge (air) per cycle, kg.

Now, in an actual engine

$$Q_{2-3} = M_f Q_c$$

$$= FM_a Q_c \text{ in kJ/cycle}$$

M_f is the mass of fuel supplied per cycle, kg

Q_c is the heating value of the fuel, KJ/kg

M_a is the mass of air taken in per cycle

F is the fuel air ratio = M_f/M_a

Substituting,

$$Q' = \frac{FM_a Q_c}{M}$$

$$\text{Now } \frac{M_a}{M} \approx \frac{V_1 - V_2}{V_1}$$

$$\text{And } \frac{V_1 - V_2}{V_1} = 1 - \frac{1}{r}$$

So, substituting for M_a/M

$$Q' = FQ_c \left(1 - \frac{1}{r}\right)$$

For isooctane, FQ_c at stoichiometric conditions is equal to 2975 KJ/kg, thus

$$Q' = 2975(r - 1)/r$$

At an ambient temperature, T_1 of 300K and C_v for air is assumed to be 0.718 KJ/kgK, we get a value of $Q''/cvT_1 = 13.8(r - 1)/r$.

Under fuel rich conditions, $\phi = 1.2$, $Q''/cvT_1 = 16.6(r - 1)/r$

Under fuel lean conditions, $\phi = 0.8$, $Q''/cvT_1 = 11.1(r - 1)/r$

1.2 Diesel Cycle

This cycle, proposed by a German engineer, Dr. Rudolph Diesel to describe the processes of his engine, is also called the constant pressure cycle. This is believed to be the equivalent air cycle for the reciprocating slow speed compression ignition engine. The P -V and T-s diagrams are shown in Figs 4 and 5 respectively.

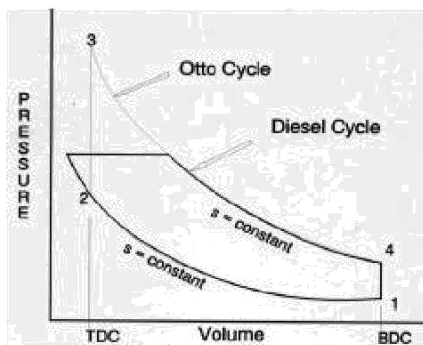


Fig 1.3 P-V Diagram of Diesel Cycle.

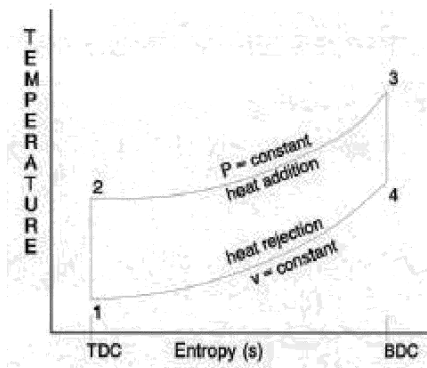


Fig 1.4 T-S Diagram of Diesel Cycle.

The cycle has processes which are the same as that of the Otto cycle except that the heat is added at constant pressure. The heat supplied, Q_s is given by $C_p(T_3 - T_2)$

Whereas the heat rejected, Q_r is given by $C_v(T_4 - T_1)$

And the thermal efficiency is given by

$$\eta_{th} = 1 - \frac{c_v(T_4 - T_1)}{c_p(T_3 - T_2)}$$

$$= 1 - \frac{1}{\gamma} \left[\frac{T_1 \left(\frac{T_4}{T_1} - 1 \right)}{T_2 \left(\frac{T_3}{T_2} - 1 \right)} \right]$$

From the T-s diagram, Fig. 5, the difference in enthalpy between points 2 and 3 is the same as that between 4 and 1, thus

$$\Delta s_{2-3} = \Delta s_{4-1}$$

$$\therefore c_v \ln\left(\frac{T_4}{T_1}\right) = c_p \ln\left(\frac{T_3}{T_2}\right)$$

$$\therefore \ln\left(\frac{T_4}{T_1}\right) = \gamma \ln\left(\frac{T_3}{T_2}\right)$$

$$\therefore \frac{T_4}{T_1} = \left(\frac{T_3}{T_2}\right)^\gamma \text{ and } \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \frac{1}{r^{\gamma-1}}$$

Substituting in eq.

$$\eta_{th} = 1 - \frac{1}{r^{\gamma-1}} \left[\frac{\left(\frac{T_3}{T_2}\right)^\gamma - 1}{\frac{T_3}{T_2} - 1} \right]$$

$$\text{Now } \frac{T_3}{T_2} = \frac{V_3}{V_2} = r_e = \text{cut-off ratio}$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \left[\frac{r_e^\gamma - 1}{\gamma(r_e - 1)} \right] \quad (26)$$

When Eq. 26 is compared with Eq. 8, it is seen that the expressions are similar except for the term in the parentheses for the Diesel cycle. It can be shown that this term is always greater than unity.

$$\text{Now } r_e = \frac{V_3}{V_2} = \frac{V_3}{V_4} \cdot \frac{V_4}{V_1} \cdot \frac{V_1}{V_2} = \frac{r}{r_c} \text{ where } r \text{ is the compression ratio and } r_c \text{ is the expansion ratio}$$

Thus, the thermal efficiency of the Diesel cycle can be written as

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \left[\left(\frac{r}{r_e} \right)^{\gamma} - 1 \right]$$

Let $r_e = r - \Delta$ since r is greater than r_e . Here, Δ is a small quantity. We therefore

$$\frac{r}{r_e} = \frac{r}{r - \Delta} = \frac{r}{r \left(1 - \frac{\Delta}{r} \right)} = \left(1 - \frac{\Delta}{r} \right)^{-1}$$

have

We can expand the last term binomially so that

$$\left(1 - \frac{\Delta}{r} \right)^{-1} = 1 + \frac{\Delta}{r} + \frac{\Delta^2}{r^2} + \frac{\Delta^3}{r^3} + \dots$$

$$\text{Also } \left(\frac{r}{r_e} \right)^{\gamma} = \frac{r^{\gamma}}{(r - \Delta)^{\gamma}} = \frac{r^{\gamma}}{r^{\gamma} \left(1 - \frac{\Delta}{r} \right)^{\gamma}} = \left(1 - \frac{\Delta}{r} \right)^{-\gamma}$$

$$\left(1 - \frac{\Delta}{r} \right)^{-\gamma} = 1 + \gamma \frac{\Delta}{r} + \frac{\gamma(\gamma+1)}{2!} \frac{\Delta^2}{r^2} + \frac{\gamma(\gamma+1)(\gamma+2)}{3!} \frac{\Delta^3}{r^3} + \dots$$

Substituting in Eq. 27, we get

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \left[\frac{\frac{\Delta}{r} + \frac{(\gamma+1)}{2!} \frac{\Delta^2}{r^2} + \frac{(\gamma+1)(\gamma+2)}{3!} \frac{\Delta^3}{r^3} + \dots}{\frac{\Delta}{r} + \frac{\Delta^2}{r^2} + \frac{\Delta^3}{r^3} + \dots} \right] \quad (28)$$

Since the coefficients of $\frac{\Delta}{r}, \frac{\Delta^2}{r^2}, \frac{\Delta^3}{r^3}, \dots$ etc are greater than unity, the quantity in the brackets in Eq. 28 will be greater than unity. Hence, for the Diesel cycle, we subtract times a quantity greater than unity from one, hence for the same r , the Otto cycle efficiency is greater than that for a Diesel cycle.

If $\frac{\Delta}{r}$ is small, the square, cube, etc of this quantity becomes progressively smaller, so the thermal efficiency of the Diesel cycle will tend towards that of the Otto cycle. From the foregoing we can see the importance of cutting off the fuel supply early in the forward stroke, a condition which, because of the short time available and the high pressures involved, introduces practical difficulties with high speed engines and necessitates very rigid fuel injection gear.

In practice, the diesel engine shows a better efficiency than the Otto cycle engine because the compression of air alone in the former allows a greater compression ratio to be employed. With a mixture of fuel and air, as in practical Otto cycle engines, the maximum temperature developed by compression must not exceed the self ignition temperature of the mixture; hence a definite limit is imposed on the maximum value of the compression ratio.

Thus Otto cycle engines have compression ratios in the range of 7 to 12 while diesel cycle engines have compression ratios in the range of 16 to 22.

$$mep = \frac{1}{V_1} \left[P_2(V_3 - V_2) + \frac{P_3V_3 - P_4V_4}{\gamma - 1} - \frac{P_2V_2 - P_1V_1}{\gamma - 1} \right] \quad (29)$$

The pressure ratio P_3/P_2 is known as explosion ratio r_p

$$\begin{aligned} \frac{P_2}{P_1} &= \left(\frac{V_1}{V_2} \right)^\gamma = r^\gamma \Rightarrow P_2 = P_1 r^\gamma, \\ P_3 &= P_2 = P_1 r^\gamma \\ P_4 &= P_3 \left(\frac{V_3}{V_4} \right)^\gamma = P_1 r^\gamma \left(\frac{V_2}{V_1} \right)^\gamma = P_1 r^\gamma \\ V_4 &= V_1, V_2 = V_c, \\ \frac{V_1}{V_2} &= \frac{V_c + V_s}{V_c} = r \\ \therefore V_s &= V_c(r - 1) \end{aligned}$$

Substituting the above values in Eq 29 to get Eq (29A) In terms of the cut-off ratio, we can obtain another expression for mep/p_1 as follows

$$mep = P_1 \frac{\gamma r'^{\gamma} (r_c - 1) - r(r_c^{\gamma} - 1)}{(r - 1)(\gamma - 1)} \quad (29A)$$

We can obtain a value of r_c for a Diesel cycle in terms of Q' as follows:

$$r_c = \frac{Q'}{c_p T_1 r'^{\gamma-1}} + 1 \quad (30)$$

We can substitute the value of η from Eq. 38 in Eq. 26, reproduced below and obtain the value of mep/p_1 for the Diesel cycle.

$$\frac{mep}{P_1} = \eta \frac{Q'}{c_p T_1} \frac{1}{\left[1 - \frac{1}{r}\right](\gamma - 1)}$$

For the Diesel cycle, the expression for mep/p_3 is as follows:

$$\frac{mep}{p_3} = \frac{mep}{p_1} \left(\frac{1}{r'^{\gamma}} \right) \quad (31)$$

Modern high speed diesel engines do not follow the Diesel cycle. The process of heat addition is partly at constant volume and partly at constant pressure. This brings us to the dual cycle.

1.3. Solved Problems

1. In an Otto cycle air at 1bar and 290K is compressed isentropic ally until the pressure is 15bar. The heat is added at constant volume until the pressure rises to 40bar. Calculate the air standard efficiency and mean effective pressure for the cycle. Take $C_v = 0.717$ KJ/Kg K and $R_{univ} = 8.314$ KJ/Kg K.

GIVEN DATA:

$$\text{Pressure (P1)} = 1\text{bar} = 100\text{KN/m}^2$$

$$\text{Temperature(T1)} = 290\text{K}$$

$$\text{Pressure (P2)} = 15\text{bar} = 1500\text{KN/m}^2$$

$$\text{Pressure (P3)} = 40\text{bar} = 4000\text{KN/m}^2$$

$$C_v = 0.717 \text{ KJ/KgK}$$

$$R_{\text{univ}} = 8.314 \text{ KJ/Kg K}$$

TO FIND:

- i) Air Standard Efficiency (η_{otto})
- ii) Mean Effective Pressure (P_m)

SOLUTION:

Here it is given $R_{\text{univ}} = 8.314 \text{ KJ/Kg K}$

We know that ,

$$\gamma = C_p / C_v \quad (\text{Here } C_p \text{ is unknown})$$

$$R_{\text{univ}} = M R$$

Since For air (O_2) molecular weight (M) = 28.97

$$8.314 = 28.97 R$$

$$\therefore R = 0.2869$$

(Since gas constant $R = C_p - C_v$)

$$0.2869 = C_p - 0.717$$

$$\therefore C_p = 1.0039 \text{ KJ/Kg K}$$

$$\gamma = \frac{C_p}{C_v} = \frac{1.0039}{0.717} = 1.4$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}}$$

$$\eta = 1 - \frac{1}{r^{1.4-1}}$$

Here 'r' is unknown.

We know that,

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

$$= \left(\frac{1500}{100}\right)^{\frac{1}{1.4}}$$

$$\therefore r = 6.919$$

$$\eta_{\text{otto}} \\ \therefore \eta_{\text{otto}} = 53.87\%$$

Mean Effective Pressure (Pm) =

$$= 1 - \frac{1}{6.919^{0.4}}$$

Pm =

$$P_1 r^{\frac{[(\gamma-1)(r^{\gamma-1}-1)]}{[(\gamma-1)(r-1)]}}$$

$$P_m = 569.92 \text{ KN/m}^2$$

$$\frac{(100)(6.919)[(2.67-1)(6.919^{0.4}-1)]}{[(1.4-1)(6.919-1)]}$$

Problem 2

Estimate the loss in air standard efficiency for the diesel engine for the compression ratio 14 and the cutoff changes from 6% to 13% of the stroke.

Given Data

Case (i)	Case (ii)
Compression ratio (r) = 14	compression ratio (r) = 14
$\rho = 6\% V_s$	$\rho = 13\% V_s$

To Find

Loss in air standard efficiency.

Solution

$$r = \frac{V_1}{V_2} = \frac{V_c + V_s}{V_c}$$

Compression ratio (r) =

$$14 = 1 + \frac{V_s}{V_c}$$

$$\frac{V_c}{V_s} = 13$$

Case (i):

Cutoff ratio (ρ) = V_3/V_2

$$\frac{V_3}{V_2} = \frac{V_c + 6\%V_s}{V_c}$$

$$= 1 + \frac{6\%V_s}{V_c}$$

$$\rho = \frac{V_3}{V_2} = 1 + (0.06)(13)$$

$$\rho = 1.78$$

We know that,

$$\begin{aligned}\eta_{\text{diesel}} &= 1 - \frac{1}{\gamma \times r^{V-1}} \left[\frac{\rho^V - 1}{\rho - 1} \right] \\ &= 1 - \left(\frac{1}{(1.4)(14)^{1.4-1}} \right) \frac{[1.78^{1.4} - 1]}{[1.78 - 1]} \\ &= 1 - (0.2485)(1.5919) \\ &= 0.6043 \times 100\%\end{aligned}$$

$$\eta_{\text{diesel}} = 60.43\%$$

case (ii):

$$\text{cutoff ratio } (\rho) = \frac{V_3}{V_2} = \frac{V_c + 13\%V_s}{V_c}$$

$$= 1 + (0.13)(13)$$

$$\rho = 2.69$$

$$\begin{aligned}\eta_{\text{diesel}} &= 1 - \frac{1}{\gamma \times r^{V-1}} \left[\frac{\rho^V - 1}{\rho - 1} \right] \\ &= 1 - \left(\frac{1}{(1.4)(14)^{1.4-1}} \right) \frac{[2.69^{1.4} - 1]}{[2.69 - 1]} \\ &= 1 - (0.24855)(1.7729)\end{aligned}$$

$$= 0.5593 \times 100\%$$

$$= 55.93\%$$

$$\text{Loss in air standard efficiency} = (\eta_{\text{diesel CASE(i)}}) - (\eta_{\text{diesel CASE(i)}})$$

$$= 0.6043 - 0.5593$$

$$= 0.0449$$

$$= 4.49\%$$

Problem3

The compression ratio of an air standard dual cycle is 12 and the maximum pressure on the cycle is limited to 70bar. The pressure and temperature of the cycle at the beginning of compression process are 1bar and 300K. Calculate the thermal efficiency and Mean Effective Pressure. Assume cylinder bore = 250mm, Stroke length = 300mm, $C_p = 1.005 \text{ KJ/Kg K}$,

$$C_v = 0.718 \text{ KJ/Kg K}.$$

Given data:

$$\text{Assume } Q_{s1} = Q_{s2}$$

$$\text{Compression ratio } (r) = 12$$

$$\text{Maximum pressure } (P_3) = (P_4) = 7000 \text{ KN/m}^2$$

$$\text{Temperature } (T_1) = 300\text{K}$$

Diameter (d) = 0.25m

Stroke length (l) = 0.3m

To find:

- (i) Dual cycle efficiency (η_{dual})
- (ii) Mean Effective Pressure (P_m)

Solution:

By Process 1-2:

$$\frac{T_2}{T_1} = \left[\frac{V_2}{V_1} \right]^{\gamma-1}$$

$$= [r]^{\gamma-1}$$

$$T_2 = 300[12]^{1.4-1}$$

$$T_2 = 810.58\text{K}$$

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

$$P_2 = [12]^{1.4} \times 100$$

$$P_2 = 3242.3\text{KN/m}^2$$

By process 2-3:

$$\frac{P_2}{T_2} = \frac{P_3}{T_3}$$

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

$$T_3 = \left[\frac{7000}{3242.3} \right] 810.58$$

$$T_3 = 1750\text{K}$$

Assuming $Q_{s1} = Q_{s2}$

$$mC_v[T_3 - T_2] = mC_p[T_4 - T_3]$$

$$0.718 [1750 - 810.58] = 1.005 [T_4 -$$

$$1750] \quad T_4 = 2421.15\text{K}$$

By process 4-5:

$$\begin{aligned} \frac{T_4}{T_5} &= \left[\frac{V_5}{V_4} \right]^{\gamma-1} \\ &= \left[\frac{r}{\rho} \right]^{1.4-1} \end{aligned}$$

We know that, $\rho = \frac{V_4}{V_3} = \frac{T_4}{T_3} = \frac{2421.15}{1750} = 1.38$

$$\frac{T_4}{T_5} = \left[\frac{12}{1.38} \right]^{0.4}$$

$$T_5 = \frac{2421.15}{\left(\frac{12}{1.38} \right)^{0.4}}$$

$$T_5 = 1019.3\text{K}$$

Heat supplied $Q_s = 2 \times m C_v \times [T_3 - T_2]$

$$= 2 \times 1 \times 0.718 \times [1750 - 810.58]$$

$$Q_s = 1349 \text{ KJ/Kg}$$

Heat rejected $Q_r = m C_v [T_5 - T_1]$

$$Q_r = 516.45 \text{ KJ/Kg}$$

$$\eta_{\text{dual}} = \frac{Q_s - Q_r}{Q_s} = \frac{832.55}{1349} \times 100$$

$$\eta_{\text{dual}} = 61.72\%$$

Stroke volume $(V_s) = \frac{\pi}{4} \times d^2 \times l$

$$= \frac{\pi}{4} \times 0.25^2 \times 0.3$$

$$V_s = 0.0147 \text{ m}^3$$

Mean effective pressure

$$(p_m) = W/V_s$$

$$= 832.58/0.0147$$

$$p_m = 56535 \text{ KN/m}^2$$

1.4 Dual Cycle

P-V Diagram of Dual Cycle.

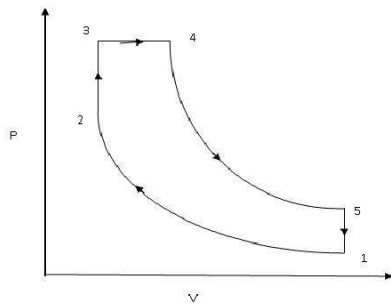


Fig 1.5 P-V Diagram

Process 1-2: Reversible adiabatic compression. Process 2-3: Constant volume heat addition. Process 3-4: Constant pressure heat addition. Process 4-5: Reversible adiabatic expansion.

Process 5-1: Constant volume heat reject

T-S Diagram of Carnot Cycle.

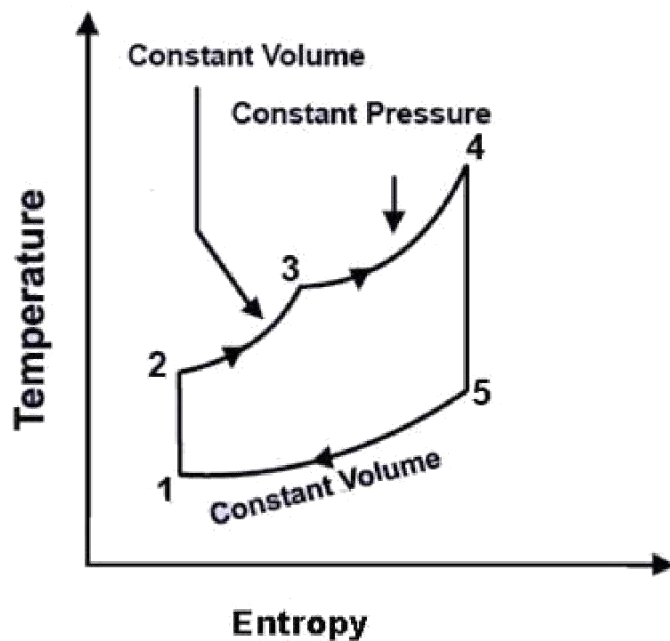


Fig 1.6 T-S Diagram

The cycle is the equivalent air cycle for reciprocating high speed compression ignition engines. The P-V and T-s diagrams are shown in Figs.6 and 7. In the cycle, compression and expansion processes are isentropic; heat addition is partly at constant volume and partly at constant pressure while heat rejection is at constant volume as in the case of the Otto and Diesel cycles.

The heat supplied, Q_s per unit mass of charge is given by $c_v(T_3 - T_2) + c_p(T_3'' - T_2)$ (32)

whereas the heat rejected, Q_r per unit mass of charge is given by $c_v(T_4 - T_1)$

and the thermal efficiency is given by

$$\eta_{th} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2) + c_p(T_3 - T_2)} \quad (33A)$$

$$= 1 - \left[\frac{T_1 \left(\frac{T_4}{T_1} - 1 \right)}{T_2 \left(\frac{T_3}{T_2} - 1 \right) + \gamma T_3 \left(\frac{T_3}{T_2} - 1 \right)} \right] \quad (33B)$$

$$= 1 - \frac{\frac{T_4}{T_1} - 1}{\frac{T_2}{T_1} \left(\frac{T_3}{T_2} - 1 \right) + \frac{\gamma T_3}{T_2} \left(\frac{T_3}{T_2} - 1 \right)} \quad (33C)$$

From thermodynamics

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = r_p \quad (34)$$

the explosion or pressure ratio and

$$\frac{T_{3'}}{T_3} = \frac{V_{3'}}{V_3} = r_c \quad (35)$$

the cut-off ratio.

$$\text{Now, } \frac{T_4}{T_1} = \frac{p_4}{p_1} = \frac{p_4}{p_{3'}} \cdot \frac{p_{3'}}{p_3} \cdot \frac{p_3}{p_2} \cdot \frac{p_2}{p_1}$$

$$\text{Also } \frac{p_4}{p_{3'}} = \left(\frac{V_{3'}}{V_4} \right)^\gamma = \left(\frac{V_{3'}}{V_3} \cdot \frac{V_3}{V_4} \right)^\gamma = \left(r_c \frac{1}{r} \right)^\gamma$$

$$\text{And } \frac{p_2}{p_1} = r^\gamma$$

$$\text{Thus } \frac{T_4}{T_1} = r_p r_c^\gamma$$

$$\text{Also } \frac{T_2}{T_1} = \left(\frac{V_1}{V_2} \right)^\gamma = r^{\gamma-1}$$

Therefore, the thermal efficiency of the dual cycle is

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \left[\frac{r_p r_c^\gamma - 1}{(r_p - 1) + \gamma r_p (r_c - 1)} \right] \quad (36)$$

We can substitute the value of η from Eq. 36 in Eq. 14 and obtain the value of mep/p_1 for the dual cycle.

In terms of the cut-off ratio and pressure ratio, we can obtain another expression for mep/p_1 as follows:

$$\frac{mep}{p_1} = \frac{\gamma r_p r' (r_c - 1) + r' (r_p - 1) - r (r_p r_c - 1)}{(r - 1)(\gamma - 1)} \quad (37)$$

For the dual cycle, the expression for mep/p_3 is as follows:

$$\frac{mep}{p_3} = \frac{mep}{p_1} \left(\frac{p_1}{p_3} \right) \quad (38)$$

Since the dual cycle is also called the limited pressure cycle, the peak pressure, p_3 , is usually specified. Since the initial pressure, p_1 , is known, the ratio p_3/p_1 is known. We can correlate r_p with this ratio as follows:

$$r_p = \frac{p_3}{p_1} \left(\frac{1}{r'^{\gamma}} \right) \quad (39)$$

We can obtain an expression for r_c in terms of Q'' and r_p and other known quantities as follows:

$$r_c = \frac{1}{\gamma} \left(\left[\left\{ \frac{Q'}{c_v T_1 r'^{\gamma-1}} \right\} \frac{1}{r_p} \right] + (\gamma - 1) \right) \quad (40)$$

We can also obtain an expression for r_p in terms of Q'' and r_c and other known quantities as follows:

$$r_p = \frac{\left[\frac{Q'}{c_v T_1 r'^{\gamma-1}} + 1 \right]}{1 + \gamma r_c - \gamma} \quad (41)$$

1.5 The Brayton Cycle

The Brayton cycle is also referred to as the Joule cycle or the gas turbine air cycle because all modern gas turbines work on this cycle. However, if the Brayton cycle is to be used for reciprocating piston engines, it requires two cylinders, one for compression and the other for expansion. Heat addition may be carried out separately in a heat exchanger or within the expander itself.

The pressure-volume and the corresponding temperature-entropy diagrams are shown in Figs 10 and 11 respectively.

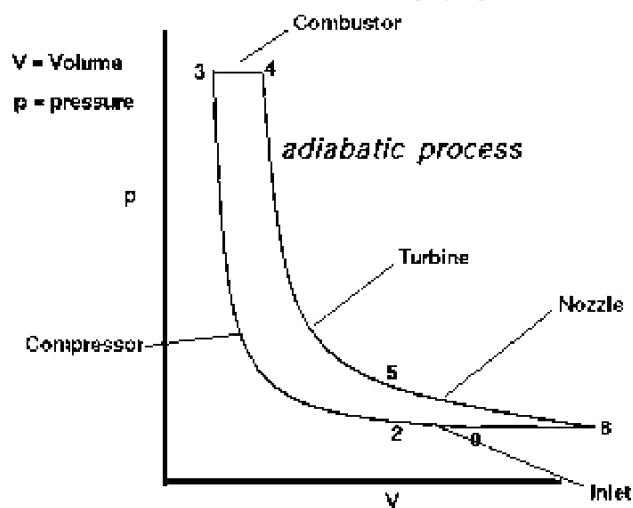


Fig 1.7 PV diagram Brayton Cycle

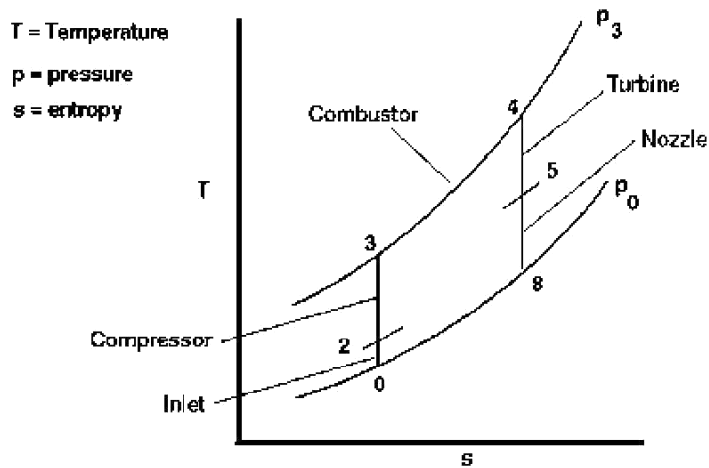


Fig 1.8 TS Diagram

The cycle consists of an isentropic compression process, a constant pressure heat addition process, an isentropic expansion process and a constant pressure heat rejection process. Expansion is carried out till the pressure drops to the initial (atmospheric) value.

Heat supplied in the cycle, Q_s , is given by $C_p(T_3 - T_2)$

Heat rejected in the cycle, Q_s , is given by $C_p(T_4 - T_1)$

Hence the thermal efficiency of the cycle is given by

$$\eta_{th} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$= 1 - \frac{T_1}{T_2} \left[\frac{\left(\frac{T_4}{T_1} - 1 \right)}{\left(\frac{T_3}{T_2} - 1 \right)} \right] \quad (42)$$

$$\text{Now } \frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4}$$

$$\text{And since } \frac{T_2}{T_1} = \frac{T_3}{T_4} \text{ we have } \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Hence, substituting in Eq. 62, we get, assuming that r_p is the pressure ratio p_2/p_1

$$\eta_{th} = 1 - \frac{T_1}{T_2}$$

$$= 1 - \frac{1}{\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}}}$$

$$= 1 - \frac{1}{r_p^{\frac{\gamma-1}{\gamma}}} \quad (43)$$

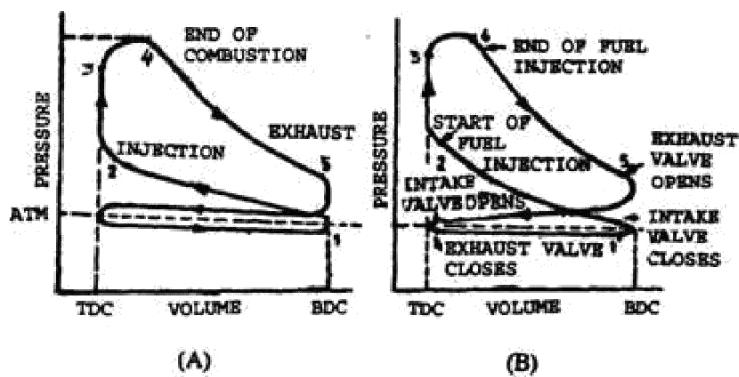
This is numerically equal to the efficiency of the Otto cycle if we put

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1} \right)^{\gamma-1} = \left(\frac{1}{r} \right)^{\gamma-1}$$

$$\text{so that } \eta_{th} = 1 - \frac{1}{r^{\gamma-1}} \quad (43A)$$

where r is the volumetric compression ratio.

1.6 Actual PV diagram of four stroke engine



1.9 Actual PV diagram of four stroke engine

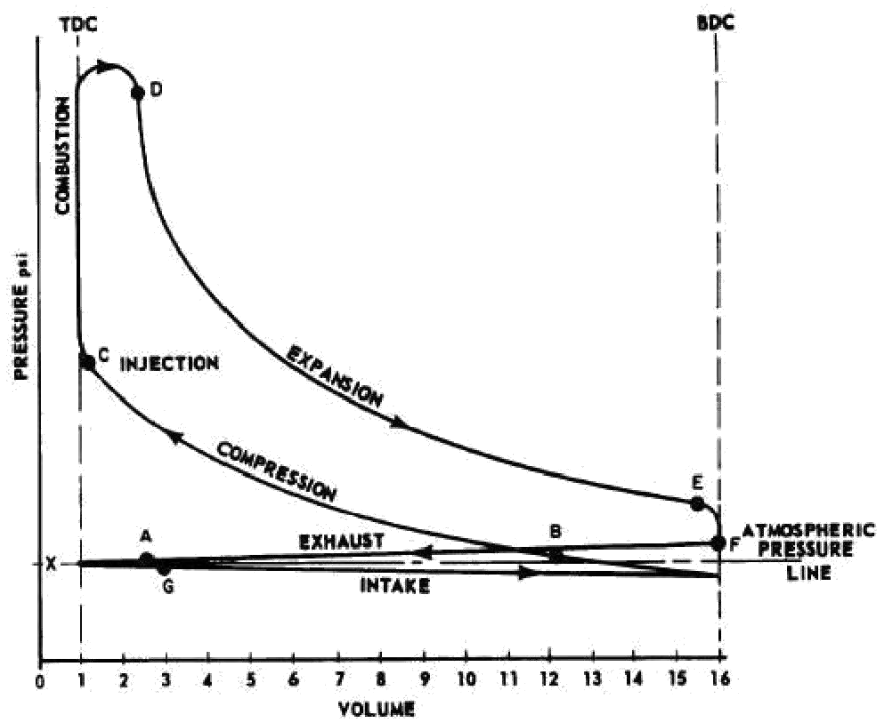


Fig 1.10 Theoretical PV diagram for four stroke engine

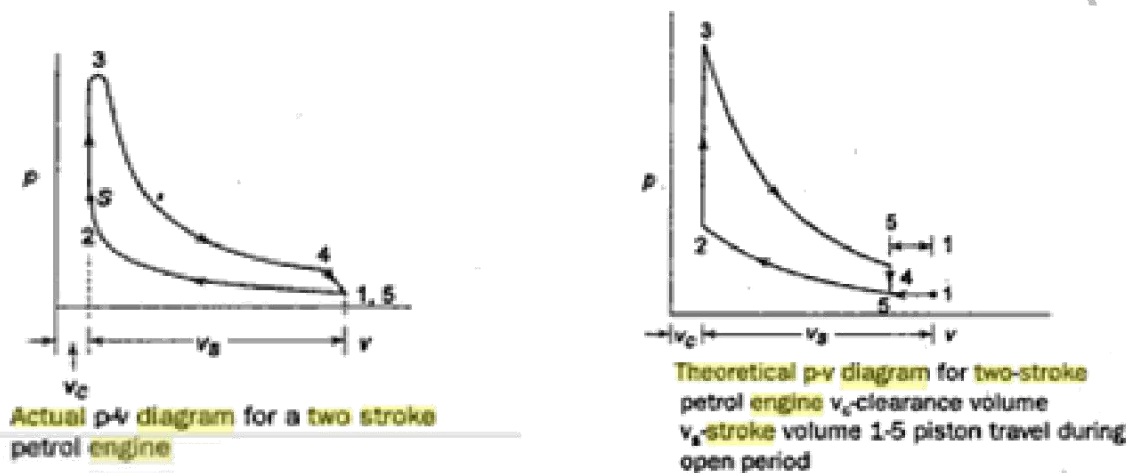


Fig 1.11 Theoretical and Actual PV diagram of two strokes Petrol Engine:

1.7 Solved Problems

4. The compression ratio of an air standard dual cycle is 12 and the maximum pressure on the cycle is limited to 70bar. The pressure and temperature of the cycle at the beginning of compression process are 1bar and 300K. Calculate the thermal efficiency and Mean Effective Pressure. Assume cylinder bore = 250mm, Stroke length = 300mm, $C_p=1.005\text{KJ/Kg K}$, $C_v=0.718\text{KJ/Kg K}$.

Given data:

Assume $Q_{s1} = Q_{s2}$

Compression ratio (r) = 12

Maximum pressure (P_3) = (P_4) = 7000 kN/m²

Temperature (T_1) = 300K

Diameter (d) = 0.25m

Stroke length (l) = 0.3m

To find:

Dual cycle efficiency (η_{dual})

Mean Effective Pressure (P_m)

Solution:

By Process 1-2:

$$\frac{T_2}{T_1} = \left[\frac{V_2}{V_1} \right]^{\gamma-1}$$

$$= [r]^{\gamma-1}$$

$$T_2 = 300[12]^{1.4-1}$$

$$T_2 = 810.58\text{K}$$

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

$$P_2 = [12]^{1.4} \times 100$$

$$P_2 = 3242.3 \text{ kN/m}^2$$

By process 2-3:

$$\frac{P_2}{T_2} = \frac{P_3}{T_3}$$

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

$$T_3 = \left[\frac{7000}{3242.3} \right]^{810.58}$$

$$T_3 = 1750 \text{ K}$$

Assuming $Q_{s1} = Q_{s2}$

$$m C_v [T_3 - T_2] = m C_p [T_4 - T_3] \quad 0.718$$

$$[1750 - 810.58] = 1.005 [T_4 - 1750]$$

$$T_4 = 2421.15 \text{ K}$$

By process 4-5:

$$\frac{T_4}{T_5} = \left[\frac{V_5}{V_4} \right]^{\gamma-1}$$

$$= \left[\frac{r}{\rho} \right]^{1.4-1}$$

We know that, $\rho = \frac{V_4}{V_3} = \frac{T_4}{T_3} = \frac{2421.15}{1750} = 1.38$

$$\frac{T_4}{T_5} = \left[\frac{12}{1.38} \right]^{0.4}$$

$$T_5 = \frac{2421.15}{\left(\frac{12}{1.38} \right)^{0.4}}$$

$$T_5 = 1019.3\text{K}$$

Heat supplied

$$Q_s = 2 \times m C_v \times [T_3 - T_2]$$

$$= 2 \times 1 \times 0.718 \times [1750 - 810.58]$$

$$Q_s = 1349\text{KJ/Kg}$$

Heat rejected

$$Q_r = m C_v [T_5 - T_1]$$

$$Q_r = 516.45 \text{ KJ/Kg}$$

$$\eta_{\text{dual}} = \frac{Q_s - Q_r}{Q_s} = \frac{832.55}{1349} \times 100$$

$$\eta_{\text{dual}} = 61.72\%$$

Stroke volume

$$(V_s) = \frac{\pi}{4} \times d^2 \times l$$

$$= \frac{\pi}{4} \times 0.25^2 \times 0.3$$

$$V_s = 0.0147\text{m}^3$$

$$\text{Mean effective pressure (} p_m) = W/V_s$$

$$= 832.58/0.0147$$

$$P_m = 56535 \text{ KN/m}^2$$

5. A diesel engine operating an air standard diesel cycle has 20cm bore and 30cm stroke. the clearance volume is 420cm^3 . if the fuel is injected at 5% of the stroke, find the air standard efficiency.

Given Data:-

$$\text{Bore diameter (d)} = 20\text{cm} = 0.2\text{m}$$

$$\text{Stroke, (l)} = 30\text{cm} = 0.3\text{m}$$

$$\text{Clearance volume, (} v_2) = 420\text{cm}^3 = 420/100^3 = 4.2 \times 10^{-4} \text{ m}^3$$

To Find:-

Air standard efficiency, (η_{diesel})

Solution:-

$$\text{Compression ratio, } r = v_1/v_2$$

$$= (v_c + v_s)/v_c$$

We know that,

$$\text{Stroke volume, } v_s = \text{area} \times \text{length}$$

$$= \left(\frac{\pi}{4}\right) d^2 \times l$$

$$= \left(\frac{\pi}{4}\right) (0.2^2) \times 0.3$$

$$V_s = 9.4 \times 10^{-3} \text{ m}^3$$

Therefore,

$$\text{Compression ratio, } (r) = \frac{4.2 \times 10^{-4} + 9.42 \times 10^{-3}}{4.2 \times 10^{-4}}$$

$$r = 23.42$$

$$\text{Cut off ratio, } \rho = v_3 / v_2$$

$$= (v_c + 5\% v_s) / v_c$$

$$= 1 + (5\% v_s) / v_c$$

$$= 1 + \frac{(0.05 \times 9.42 \times 10^{-3})}{4.2 \times 10^{-4}}$$

$$\rho = 2.12$$

We know the equation,

$$\eta_{diesel} = 1 - \left(\frac{1}{r^{(\gamma-1)}} \right) \times \left(\frac{\rho^\gamma - 1}{\rho - 1} \right)$$

$$= 1 - \frac{1}{1.4 \times 23.42^{1.4-1}} \left(\frac{2.12^{1.4} - 1}{2.12 - 1} \right)$$

$$= 1 - (0.20229)(1.6636)$$

$$= 0.6634 \times 100$$

$$\eta_{diesel} = 66.34\%$$

TWO MARK UNIVERSITY QUESTIONS:

1. What is a thermodynamic cycle?
2. What is meant by air standard cycle?
3. Name the various "gas power cycles".
4. What are the assumptions made for air standard cycle analysis
5. Mention the various processes of the Otto cycle.
6. Mention the various processes of diesel cycle.
7. Mention the various processes of dual cycle.
9. Define air standard cycle efficiency.
10. Define mean effective pressure as applied to gas power cycles. How it is related to indicate power of an I.C engine?
11. Define the following terms. (i) Compression ratio (ii) Cut off ratio, (iii) .Expansion ratio

UNIVERSITY ESSAY QUESTIONS:

1. Derive an expression for the air standard efficiency of Otto cycle in terms of volume ratio. (16)
2. Derive an expression for the air standard efficiency of Diesel cycle. (16)
3. Derive an expression for the air standard efficiency of Dual cycle. (16)
4. Explain the working of 4 stroke cycle Diesel engine. Draw the theoretical and actual PV diagram.
5. Derive the expression for air standard efficiency of Brayton cycle in terms of pressure ratio.
6. A Dual combustion air standard cycle has a compression ratio of 10. The constant pressure part of combustion takes place at 40 bar. The highest and the lowest temperature of the cycle are 1725°C and 270°C respectively. The pressure at the beginning of compression is 1 bar. Calculate (i) the pressure and temperature at key points of the cycle. (ii) The heat supplied at constant volume, (iii) the heat supplied at constant pressure. (iv) The heat rejected. (v) The work output. (vi) The efficiency and (vii) mep. (16)
7. An Engine-working on Otto cycle has a volume of 0.45 m³, pressure 1 bar and temperature 300°C at the beginning of compression stroke. At the end of compression stroke, the pressure is 11 bar and 210 KJ of heat is added at constant volume. Determine (i) Pressure, temperature and volumes at salient points in the cycle. (ii) Efficiency.

8. Explain the working of 4-stroke cycle Diesel engine. Draw the theoretical and actual valve-timing diagram for the engine. Explain the reasons for the difference.
9. Air enters the compressor of a gas turbine at 100 KPa and 25 °C. For a pressure ratio of 5 and a maximum temperature of 850°C. Determine the thermal efficiency using the Brayton cycle. (16)
10. The following data is referred for an air standard diesel cycle compression ratio = 15 heat added = 200 KJ/Kg- minimum temperature in the cycle = 25°C Suction pressure = 1 bar Calculate
1. Pressure and temperature at the Salient point. 2. Thermal efficiency 3. Mean effective pressure, 4. Power output of the cycle, if flow rate of air is 2 Kg/s (16)

Sample Problems

1. A Dual combustion air standard cycle has a compression ratio of 10. The constant pressure part of combustion takes place at 40 bar. The highest and the lowest temperature of the cycle are 1727° C and 27° C respectively. The pressure at the beginning of compression is 1 bar. Calculate-
(i) The pressure and temperature at key points of the cycle. (ii) The heat supplied at constant volume, (iii) The heat supplied at constant pressure (iv) The heat rejected (v) The Work output, (vi) The efficiency and (vii) Mean effective pressure.
2. An Engine working on Otto cycle has a volume of 0.45 m³, pressure 1 bar and Temperature 300°C, at the beginning of compression stroke. At the end of Compression stroke, the pressure is 11 bar and 210 KJ of heat is added at constant Volume.
Determine i. Pressure, temperature and volumes at salient points in the cycle. ii. Efficiency.

ME1251 THERMAL ENGINEERING

UNIT II

INTERNAL COMBUSTION ENGINES

CONTENTS

TECHNICAL TERMS

- 2.1 Classification of IC engine
- 2.2 Components of I.C engine 1.Cylinder block
- 2.3 Theoretical valve timing diagram of four stroke engine
- 2.4 Comparison of two stroke and four stroke engines
- 2.6 Simple Carburetor
- 2.7 Diesel Pump and Injector system
- 2.8 Diesel knocking and detonation
- 2.9 Ignition System
- 2.10 Comparison between Battery and Magneto Ignition System
- 2.11 Lubrication System
- 2.12 Cooling System
 - 2.12.1 Air Cooled System
 - 2.12.2 Water Cooling System
- 2.13 Emission Formation in C.I. Engine
- 2.14 Principle C.I. Engine Exhaust Constituents
- 2.15 Sample problems
- 2.16 Solved Problems
- 2.17 Two Marks University Questions
- 2.18 University Essay Questions

TECHNICAL TERMS

1. **IC Engines:** Air and fuel mixture flows through inlet valve and exhaust leaves through exhaust valve Converts reciprocating motion to rotary motion using piston and crank shaft
2. **TDC:** Top Dead Center: Position of the piston where it forms the smallest volume
3. **BDC:** Bottom Dead Center: Position of the piston where it forms the largest volume
4. **Stroke:** Stroke means Distance between TDC and BDC
5. **Bore:** Bore Diameter of the piston (internal diameter of the cylinder)
6. **Clearance volume:** The clearance volume means minimum volume formed is called the clearance
7. **Compression ratio:** The compression ratio means ratio of total cylinder volume to clearance volume.
8. **MEP:** Mean effective pressure: A const. theoretical pressure that if acts on piston produces work same as that during an actual cycle $W_{net} = MEP \times \text{Piston area} \times \text{Stroke}$
9. **Common layouts of engines are:**

Reciprocating:

- Two-stroke engine
- Four-stroke engine (Otto cycle)
- Six-stroke engine
- Diesel engine
- Atkinson cycle
- Miller cycle

10. Two-Stroke Engine:

Engines based on the two-stroke cycle use two strokes (one up, one down) for every power stroke. Since there are no dedicated intake or exhaust strokes

11. **Cylinder:** A cylindrical vessel in which a piston makes an up and down motion.

- 12. Piston:** A cylindrical component making an up and down movement in the cylinder
- 13. Combustion chamber:** A portion above the cylinder in which the combustion of the fuel-air mixture takes place
- 14. Intake and exhaust ports:** Ports that carry fresh fuel-air mixture into the combustion chamber and products of combustion away
- 15. Crankshaft:** A shaft that converts reciprocating motion of the piston into rotary motion
- 16. Connecting rod:** A rod that connects the piston to the crankshaft
- 17. Spark plug:** An ignition-source in the cylinder head that initiates the combustion process
- 18. Four stroke engine:** Engines based on the four-stroke ("Otto cycle") have one power stroke for every four strokes (up-down-up-down) and employ sparkplug ignition. Combustion occurs rapidly, and during combustion the volume varies little ("constant volume") They are used in cars, larger boats, some motorcycles, and many light aircraft. They are generally quieter, more efficient, and larger than their two-stroke counterparts.
- 19. AFR: Air-fuel ratio** is the mass ratio of air to fuel present in an internal combustion engine. If exactly enough air is provided to completely burn all of the fuel, the ratio is known as the stoichiometric mixture, often abbreviated to **stoich**. AFR is an important measure for anti-pollution and performance-tuning reasons

UNIT-II

INTERNAL COMBUSTION ENGINES

2.1 Classification of IC engine:

Normally IC engines are classified into I.C.I engines and

2.S.I engines

Some of the important classifications are given below,

1. Number of strokes -two stroke and four stroke
2. Working Cycles -Otto, Diesel, Dual cycle
3. Cylinder arrangement -In-line, V-type, Opposed, Radial
4. Valve Arrangement -T-head, F-head, L-head, I-head
5. Fuel Used -Petrol, Diesel, Gas
6. Combustion chamber design -Open, divided
7. Cooling System -Water and air cooling
8. According to the number of cylinders -Single and Multi
9. According to the speed -Slow, medium, and high speed engines
10. According to the application -Stationary, Automotive, Marine, Locomotive, Aircraft etc.,

2.2 Components of I.C engine 1.Cylinder block:

The cylinder block is the main body of the engine, the structure that supports all the other components of the engine. In the case of the single cylinder engine the cylinder block houses the cylinder, while in the case of multi-cylinder engine the number of cylinders are cast together to form the cylinder block. The cylinder head is mounted at the top of the cylinder block.

When the vehicle runs, large amounts of heat are generated within the cylinder block. To remove this heat the cylinder block and the cylinder head are cooled by water flowing through the water jackets within larger engines such as those found in cars and trucks. For smaller vehicles like motorcycles, fins are provided on the cylinder block and on the cylinder head to cool them. The bottom portion of the cylinder block is called a crankcase. Within the crankcase is where lubricating oil, which is used for lubricating various moving parts of the engine, is stored.

Cylinder:

As the name suggests it is a cylindrical shaped vessel fitted in the cylinder block. This cylinder can be removed from the cylinder block and machined whenever required to. It is also called a liner or sleeve. Inside the cylinder the piston moves up and down, which is called the reciprocating motion of the piston. Burning of fuel occurs at the top of the cylinder, due to which the reciprocating motion of the piston is produced. The surface of the cylinder is finished to a high finish, so that there is minimal friction between the piston and the cylinder.

Piston:

The piston is the round cylindrical component that performs a reciprocating motion inside the cylinder. While the cylinder itself is the female part, the piston is the male part. The piston fits perfectly inside the cylinder. Piston rings are fitted over the piston. The gap between the piston and the cylinder is filled by the piston rings and lubricating oil. The piston is usually made up of aluminum

Piston rings:

The piston rings are thin rings fitted in the slots made along the surface of the piston. It provides a tight seal between the piston and the cylinder walls that prevents

leaking of the combustion gases from one side to the other. This ensures that that motion of the piston produces as close as to the power generated from inside the cylinder.

Combustion chamber:

It is in the combustion chamber where the actual burning of fuel occurs. It is the uppermost portion of the cylinder enclosed by the cylinder head and the piston. When the fuel is burnt, much thermal energy is produced which generates excessively high pressures causing the reciprocating motion of the piston.

Inlet manifold:

Through the inlet manifold the air or air-fuel mixture is drawn into the cylinder.

Exhaust manifold:

All the exhaust gases generated inside the cylinder after burning of fuel are discharged through the exhaust manifold into the atmosphere.

Inlet and exhaust valves:

The inlet and the exhaust valves are placed at the top of the cylinder in the cylinder head. The inlet valve allows the intake of the fuel during suction stroke of the piston and to close thereafter. During the exhaust stroke of the piston the exhaust valves open allowing the exhaust gases to release to the atmosphere. Both these valves allow the flow of fuel and gases in single direction only.

Spark plug:

The spark plug is a device that produces a small spark that causes the instant burning of the pressurized fuel.

Connecting rod:

It is the connecting link between the piston and the crankshaft that performs the rotary motion. There are two ends of the connecting rod called the small end and big end.

The small end of the connecting rod is connected to the piston by gudgeon pin, while the big end is connected to crankshaft by crank pin

Crankshaft:

The crankshaft performs the rotary motion. It is connected to the axle of the wheels which move as the crankshaft rotates. The reciprocating motion of the piston is converted into the rotary motion of the crankshaft with the help of connecting rod. The

crankshaft is located in the crankcase and it rotates in the bushings.

Camshaft:

It takes driving force from crankshaft through gear train or chain and operates the inlet valve as well as exhaust valve with the help of cam followers, push rod and rocker arms.

2.3 Theoretical valve timing diagram of four stroke engine:

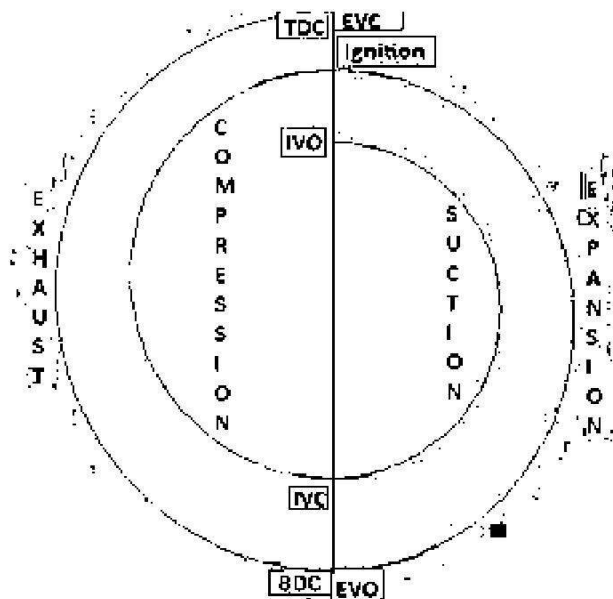
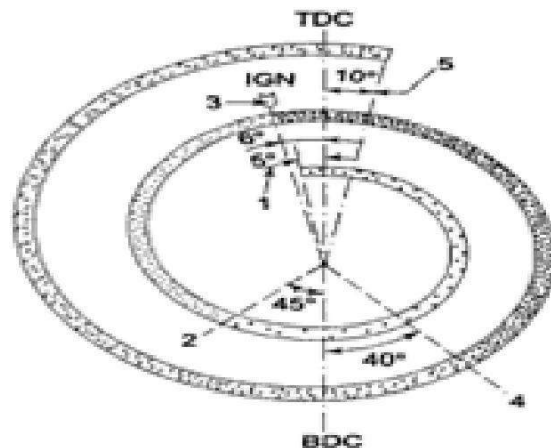


Fig 2.1 Valve Timing Diagram-4stroke engine/otto cycle (Theoretical)

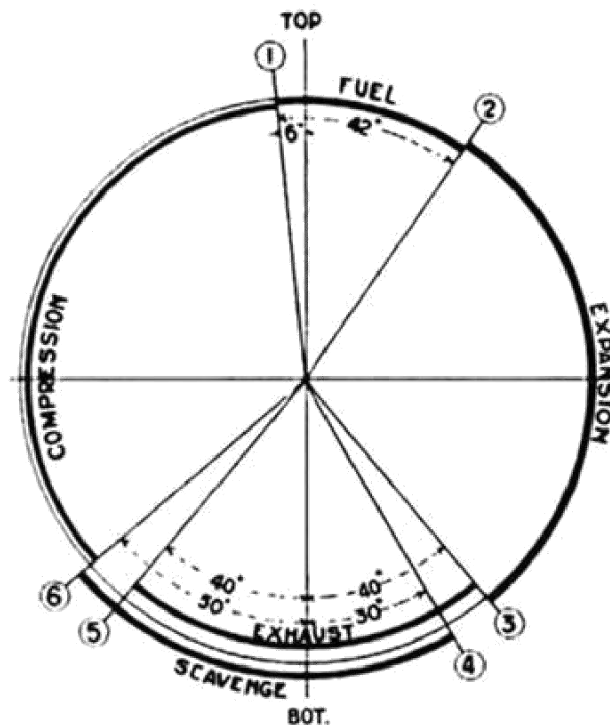
2.3.1 Actual valve timing diagram of four stroke engine:



Valve timing 1. Inlet valve open at 5° before TDC; 2. Inlet valve closed at 45° after BDC; 3. Ignition at 6° before TDC; 4. Exhaust valve open at 40° before BDC; 5. Exhaust valve closed at 10° after TDC

Fig 2.2 Actual valve timing diagram of four stroke engine

2.3.2 Theoretical port timing diagram of two stroke engine:



1- IPO 2- IPC 3- EPO 4- TPO 5- TPC 6-EPC

Fig 2.3 Theoretical port timing diagram of two stroke engine

2.4 Comparison of two stroke and four stroke engines:

Table 2.1 Comparison of two stroke and four stroke engines

SI No.	Four stroke Cycle engine	Two Stroke Cycle Engine
1	For every two revolutions of the crankshaft, there is one power stroke i.e., after every four piston strokes.	For every one revolution of the crankshaft, there is one power stroke i.e., after every two piston strokes.
2	For some power, more space is required.	For the same power less space is required.
3	Valves are required - inlet and exhaust valves.	Ports are made in the cylinder walls - inlet, exhaust and transfer port.
4	As the valves move frequently, lubrication is essential.	Arrangement of ports, reduce wear and tear and lubrication is not very essential.
5	Heavier flywheel is required because the turning moment (torque) of the crankshaft is not uniform i.e. one working stroke in every two revolution.	Lighter flywheel is required because the turning moment of the crankshaft is much more uniform i.e. one working stroke for every revolution.
6	These engines are water cooled, making it complicated in design and difficulty to maintain	These engines are generally air cooled, simple in design and easy to maintain.
7	The fuel-air change (mixture) is completely utilized thus efficiency is higher	As inlet and outlet port open simultaneous, some times fresh charge escapes with the exhaust gases are not always completely removed. This causes lower efficiency

Sl. No.	Petrol Engine	Diesel Engine
1	The exhaust is less noisy.	The exhaust is noisy due to short time available for exhaust.
2	Intake (Petrol) and air is admitted into the cylinder during suction stroke.	Air alone is admitted into the cylinder during suction stroke.
3	Fuel Ignition: - By spark plug - Spark Ignition (SI) engine.	By the compressed hot air Compression Ignition (CI) Engine.
4	Cycle of operation: - Otto cycle (constant volume cycle)	Diesel Cycle.
5	Compression Ratio Low (7 to 8).	High (16 to 17).
6	Fuel admission Through carburetor.	Through fuel injector.
7	Engine speed: - high speed; can run up to 5000 rpm since petrol ' engine is lighter.	Low speed; about up to 3500 rpm
8	Weight:- Because of low compression ratio, the engine cylinder undergoes less pressure. It weights about 0.5 to 3 kg per KW (kilo watt) of power produced.	Higher compression ratio in diesel engines result in higher pressure, therefore diesel engines are sturdier and heavier. Diesel engine weights about 2 to 10 kg per KW of power produced
9	Lubricating Property: - Petrol does not have lubricating properties	Diesel has lubricating properties.
10	Engine starting in cold condition is easy	Greater cranking effort is required to overcome the higher compression ratio, due to the cold air in the combustion chamber.
11	Fire hazard: - Petrol is highly volatile and there is a greater risk of fire.	Diesel 'is less volatile and has a reduced risk of fire.
12	Engine cost: - Less costly since the fuel systems	More costly since the fuel injection systems are more expensive.

UNIT-II

2.6 Simple Carburetor:

The function of a carburetor is to vaporize the petrol (gasoline) by means of engine suction and to supply the required air and fuel (petrol) mixture to the engine cylinder. During the suction stroke, air flows from atmosphere into the cylinder. As the air passes through the venturi, velocity of air increases and its pressure falls below the atmosphere. The pressure at the nozzle tip is also below the atmospheric pressure. The pressure on the fuel surface of the fuel tank is atmospheric. Due to which a pressure difference is created, which causes the flow of fuel through the fuel jet into the air stream. As the fuel and air pass ahead of the venturi, the fuel gets vaporized and required uniform mixture is supplied to the engine. The quantity of fuel supplied to the engine depends upon the opening of throttle valve which is governed by the governor.

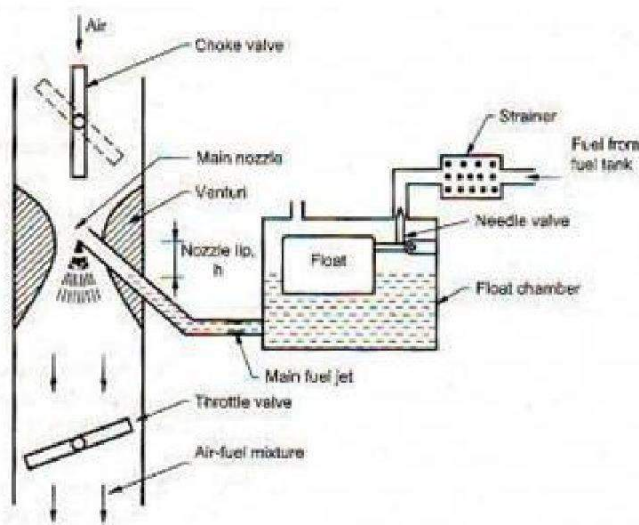


Fig 2.4 Simple Carburetor

The main parts of a simple carburetor are:

Float chamber: The level of fuel in the float chamber is maintained slightly below the tip of the nozzle. If the level of petrol is above then the petrol will run from the nozzle and drip from the carburetor. If the petrol level is kept low than the tip of the nozzle then part of pressure head is lost in lifting the petrol up to the tip of nozzle. Generally it is kept at 5mm from the level of petrol in the float chamber. The level of the fuel is kept constant with the help of float and needle

<http://www.francisxavier.ac.in>

valve. The needle valve closes the inlet supply from main tank if the level rises above the required level. If the level of fuel decreases then the needle valve opens the supply. Generally the fuel level is kept 5mm below the nozzle tip.

Venturi: When the mixture passes through the narrowest section its velocity increases and pressure falls below the atmospheric. As it passes through the divergent section, pressure increases again.

Throttle valve: It controls the quantity of air and fuel mixture supplied to the engine through intake manifold and also the head under which the fuel flows.

Choke: It provides an extra rich mixture during to the engine starting and in cold weather to warm up the engine. The choke valve is nearly closed during cold starting and warming. It creates a high vacuum near the fuel jet which causes flow of more fuel from the jet.

2.7 Diesel Pump and Injector system:

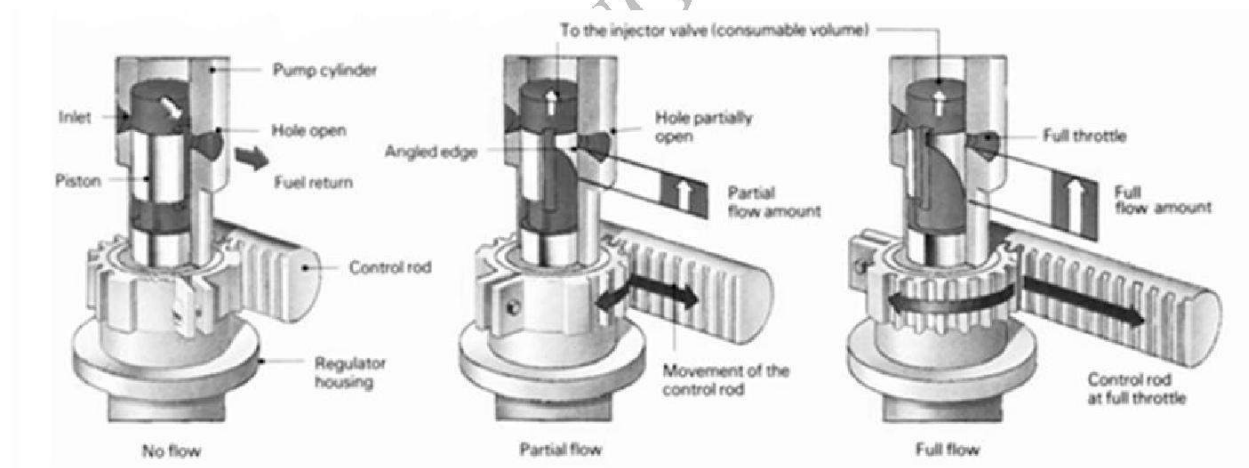


Fig. 2.5 Diesel Pump and Injector system:

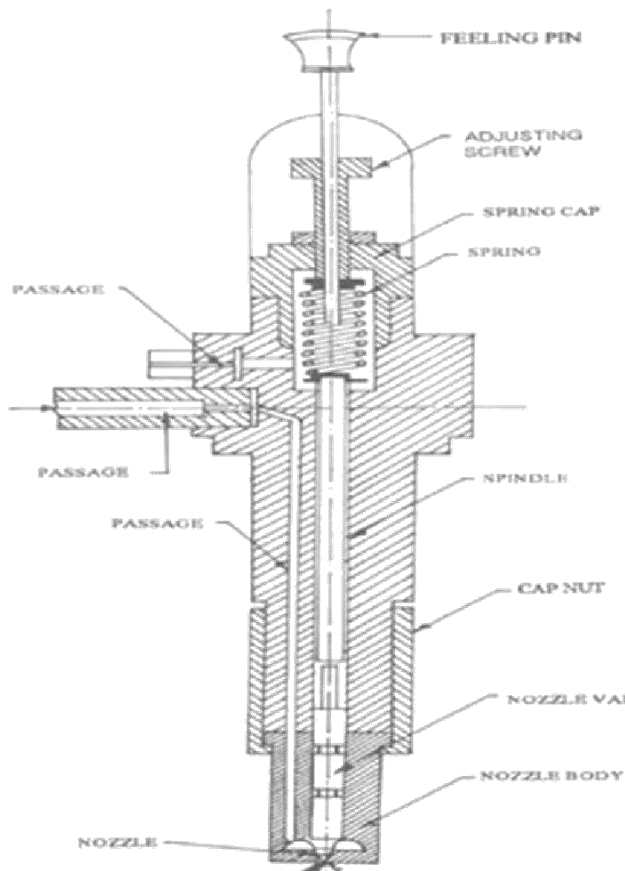


Fig. 2.6 Fuel injector

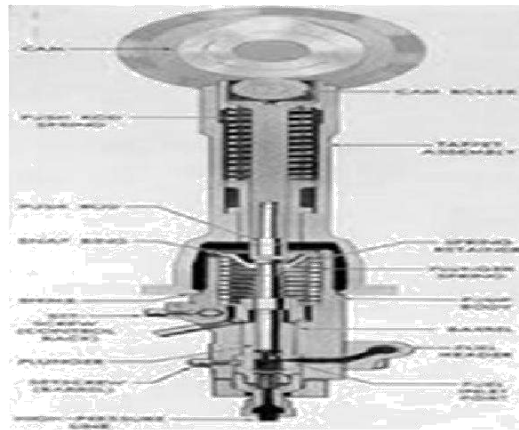


Fig 2.7 fuel injector

2.8 Diesel knocking and detonation:

We already know that if the delay period is long, a large amount of fuel will be injected and accumulated in the chamber. The auto ignition of this large amount of fuel may cause high rate of pressure rise and high maximum pressure which may cause knocking in diesel engines. A long delay period not only increases the amount of fuel injected by the moment of ignition, but also improve the homogeneity of the fuel air mixture and its chemical preparedness for explosion type self ignition similar to detonation in SI engines. It is very instructive to compare the phenomenon of detonation in SI engines with that of knocking in CI engines. There is no doubt that these two phenomena are fundamentally similar. Both are processes of auto ignition subject to the ignition time lag characteristic of the fuel air mixture. However, differences in the knocking phenomena of the SI engine and the CI engine should also be carefully noted:

1. In the SI engine, the detonation occurs near the end of combustion whereas in the CI engine detonation occurs near the beginning of combustion as shown in fig. 6.10.
2. The detonation in the SI engine is of a homogeneous charge causing very high rate of pressure rise and very high maximum pressure. In the CI engine the fuel and air are in perfectly mixed and hence the rate of pressure rise is normally lower than that in the detonating part of the charge in the SI engine.
3. Since in the CI engine the fuel is injected into the cylinder only at the end of the compression stroke there is no question of pre ignition or pre mature ignition as in the SI engine.
4. In the SI engine it is relatively easy to distinguish between knocking and non- knocking operation as the

<http://www.francisxavier.ac.in>

human ear easily find the distinction. However, in the case of the CI engine the normal ignition is itself by auto ignition and hence no CI engines have a sufficiently high rate of pressure rise per degree crank angle to cause audible noise. When such noise becomes excessive or there is excessive vibration in engine structure, in the opinion of the observer, the engine is sending to knock. It is clear that personal judgment is involved here. Thus in the CI engine there is no definite distinction between normal and knocking combustion. The maximum rate of pressure rise in the CI engine may reach as high as 10bar per crank degree angle.

It is most important to note that factors that tend to reduce detonation in the SI engine increase knocking in CI engine and vice versa because of the following reason. The detonation of knocking in the SI engine is due to simultaneous auto ignition of the last part of the charge. To eliminate detonation in the SI engine we want to prevent all together the auto ignition of the last part of the charge and therefore desire a long delay period and high self ignition temperature of the fuel. To eliminate knocking the CI engine we want to achieve auto ignitions early as possible therefore desire a short delay period and low self ignition temperature of the fuel. Table 6.2 gives the factors which reduce knocking in the SI and CI engines

Table 2.2: Factors tending to reduce knocking in SI and CI engine

Sr. No.	Factors	SI Engine	CI Engine
1	Self ignition temperature of fuel	High	Low
2	Time lag or delay period for fuel	Long	Short
3	Compression ratio	Low	High
4	Inlet temperature	Low	High
5	Inlet pressure	Low	High
6	Combustion chamber wall temperature	Low	High
7	Speed	High	Low
8	Cylinder size	Small	Large

It is also clear from the table and discussion that a good CI engine fuel is a bad SI engine fuel and a good SI engine is bad CI engine fuel. In other words diesel oil has low self ignition temperature and short time lag where as petrol have high self ignition temperature and a long ignition lag. In terms of fuel rating diesel oil has high cetane number (40 – 60) and low octane number (about 30) and petrol has high octane number (80 – 90) and low cetane number (18).

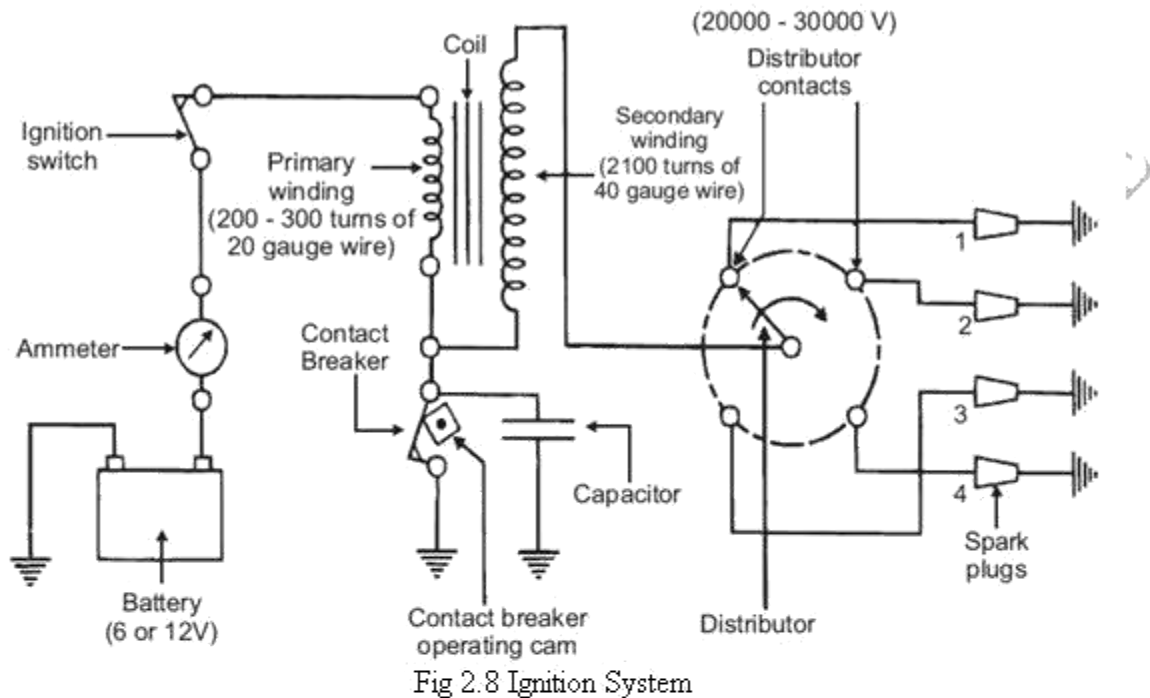
2.9 Ignition System:

Basically Convectional Ignition systems are of 2 types : (a) Battery or Coil Ignition System, and (b) Magneto Ignition System. Both these conventional, ignition systems work on mutual electromagnetic induction principle. Battery ignition system was generally used in 4-wheelers, but now-a-days it is more commonly used in 2-wheelers also (i.e. Button start, 2-wheelers like Pulsar, Kinetic Honda; Honda-Activa, Scooty, Fiero, etc.). In this case 6 V or 12 V batteries will supply necessary current in the primary winding. Magneto ignition system is mainly used in 2-wheelers, kick start engines. (Example, Bajaj Scooters, Boxer, Victor, Splendor, Passion, etc.). In this case magneto will produce and supply current to the primary winding. So in magneto ignition system magneto replaces the battery. **Battery or Coil Ignition System** Figure shows line diagram of battery ignition system for a 4-cylinder petrol engine. It mainly consists of a 6 or 12 volt battery, ammeter, ignition switch, auto-transformer (step up transformer), contact breaker, capacitor, distributor rotor, distributor contact points, spark plugs, etc. Note that the Figure 4.1 shows the ignition system for 4-cylinder petrol engine, here there are 4-spark plugs and contact breaker cam has 4-corners. (If it is for 6-cylinder engine it will have 6-spark plugs and contact breaker cam will be a hexagon).

The ignition system is divided into 2-circuits:

i. Primary Circuit :

- a. It consists of 6 or 12 V battery, ammeter, ignition switch, primary winding it has 200-300 turns of 20 SWG (Sharps Wire Gauge) gauge wire, contact breaker, capacitor.



(ii) Secondary Circuit:

It consists of secondary winding. Secondary **Ignition Systems** winding consists of about 21000 turns of 40 (S WG) gauge wire. Bottom end of which is connected to bottom end of primary and top end of secondary winding is connected to centre of distributor rotor. Distributor rotors rotate and make contacts with contact points and are connected to spark plugs which are fitted in cylinder heads (engine earth). (iii) **Working** : When the ignition switch is closed and engine is cranked, as soon as the contact breaker closes, a low voltage current will flow through the primary winding. It is also to be noted that the contact breaker cam opens and closes the circuit 4-times (for 4 cylinders) in one revolution. When the contact breaker opens the contact, the magnetic field begins to collapse. Because of this collapsing magnetic field, current will be induced in the secondary winding. And because of more turns (@ 21000 turns) of secondary, voltage goes up to 28000-30000 volts. This high voltage current is brought to centre of the distributor rotor. Distributor rotor rotates and supplies this high voltage current to proper spark plug depending upon the engine firing order. When the high voltage current jumps the

spark plug gap, it produces the spark and the charge is ignited-combustion starts-products of combustion expand and produce power. **Magneto Ignition System** In this case magneto will produce and supply the required current to the primary winding. In this case as shown, we can have rotating magneto with fixed coil or rotating coil with fixed magneto for producing and supplying current to primary, remaining arrangement is same as that of a battery ignition system.

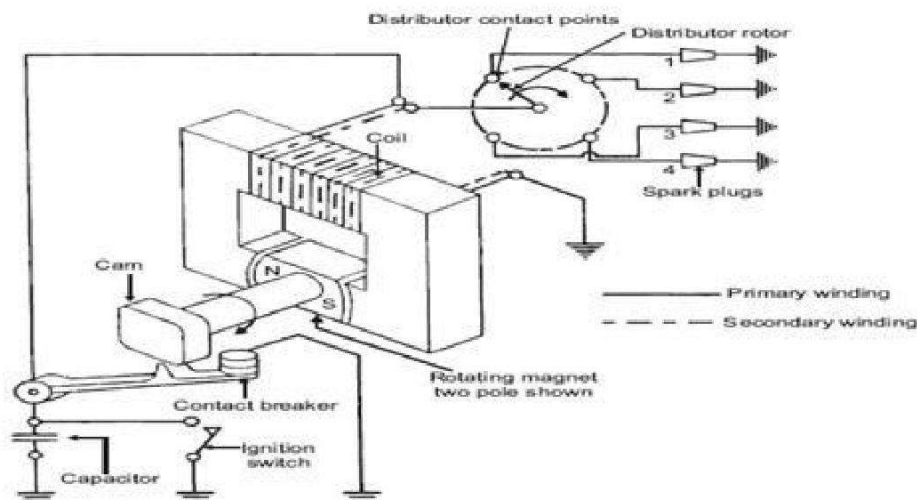


Fig 2.9 Ignition System Secondary Circuit

2.10 Comparison between Battery and Magneto Ignition System:

Table 2.3 Comparison between Battery and Magneto Ignition System

Battery Ignition	Magneto Ignition
Battery is a must.	No battery needed.
Battery supplies current in primary circuit.	Magneto produces the required current for primary circuit.
A good spark is available at low speed also.	During starting the quality of spark is poor due to slow speed.
Occupies more space.	Very much compact.
Recharging is a must in case battery gets discharged.	No such arrangement required.
Mostly employed in car and bus for which it is required to crank the engine.	Used on motorcycles, scooters, etc.
Battery maintenance is required.	No battery maintenance problems.

2.11 Lubrication System:

2.11.1 Splash:

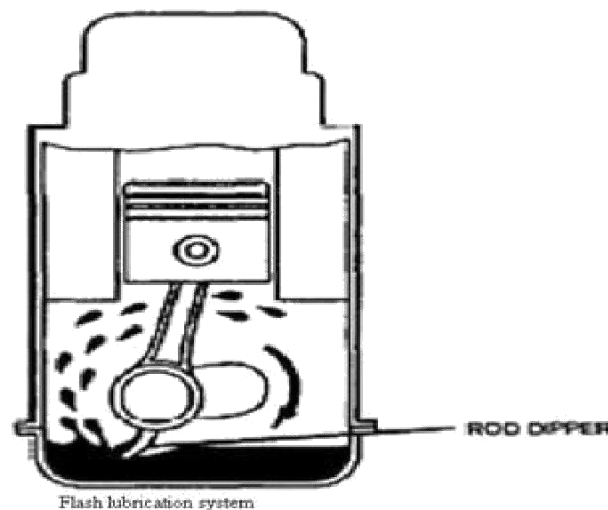


Fig 2.10 Splash

The splash system is no longer used in automotive engines. It is widely used in small four-cycle engines for lawn mowers, outboard marine operation, and so on. In the splash lubricating system, oil is splashed up from the oil pan or oil trays in the lower part of the crankcase. The oil is thrown upward as droplets or fine mist and provides adequate lubrication to

<http://www.francixavier.ac.in>

valve mechanisms, piston pins, cylinder walls, and piston rings. In the engine, dippers on the connecting-rod bearing caps enter the oil pan with each crankshaft revolution to produce the oil splash. A passage is drilled in each connecting rod from the dipper to the bearing to ensure lubrication. This system is too uncertain for automotive applications. One reason is that the level of oil in the crankcase will vary greatly the amount of lubrication received by the engine. A high level results in excess lubrication and oil consumption and a slightly low level results in inadequate lubrication and failure of the engine.

2.11.2 Combination Splash and Force Feed:

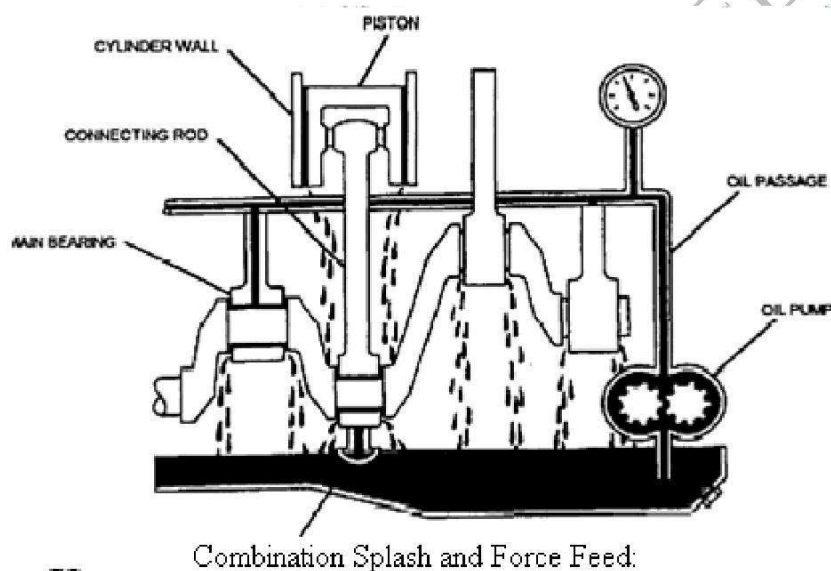


Fig 2.11 Combination Splash and Force Feed

In a combination splash and force feed, oil is delivered to some parts by means of splashing and other parts through oil passages under pressure from the oil pump. The oil from the pump enters the oil galleries. From the oil galleries, it flows to the main bearings and camshaft bearings. The main bearings have oil-feed holes or grooves that feed oil into drilled

<http://www.francisxavier.ac.in>

passages in the crankshaft. The oil flows through these passages to the connecting rod bearings. From there, on some engines, it flows through holes drilled in the connecting rods to the piston-pin bearings. Cylinder walls are lubricated by splashing oil thrown off from the connecting-rod bearings. Some engines use small troughs under each connecting rod that are kept full by small nozzles which deliver oil under pressure from the oil pump. These oil nozzles deliver an increasingly heavy stream as speed increases. At very high speeds these oil streams are powerful enough to strike the dippers directly. This causes a much heavier splash so that adequate lubrication of the pistons and the connecting-rod bearings is provided at higher speeds. If a combination system is used on an overhead valve engine, the upper valve train is lubricated by pressure from the pump.

2.11.3 Force Feed :

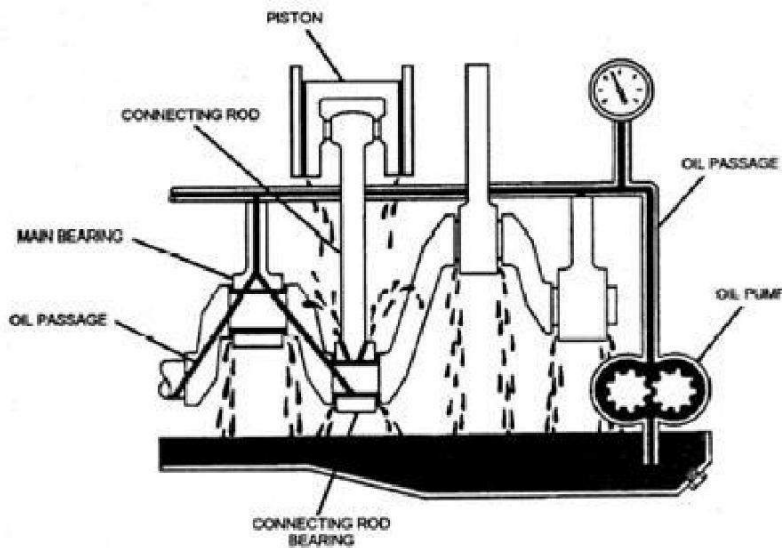


Fig 2.12 Force Feed

A somewhat more complete pressurization of lubrication is achieved in the force-feed lubrication system. Oil is forced by the oil pump from the crankcase to the main bearings and the camshaft bearings. Unlike the combination system the connecting-rod bearings are also fed oil under pressure from the pump.

Oil passages are drilled in the crankshaft to lead oil to the connecting-rod bearings. The passages

<http://www.francisxavier.ac.in>

deliver oil from the main bearing journals to the rod bearing journals. In some engines, these

opening are holes that line up once for every crankshaft revolution. In other engines, there are annular grooves in the main bearings through which oil can feed constantly into the hole in the crankshaft. The pressurized oil that lubricates the connecting-rod bearings goes on to lubricate the pistons and walls by squirting out through strategically drilled holes. This lubrication system is

used in virtually all engines that are equipped with semi floating piston pins.

2.11.4 Full Force Feed:

In a full force-feed lubrication system, the main bearings, rod bearings, camshaft bearings, and the complete valve mechanism are lubricated by oil under pressure. In addition, the full force-feed lubrication system provides lubrication under pressure to the pistons and the piston pins. This is accomplished by holes drilled the length of the connecting rod, creating an oil passage from the connecting rod bearing to the piston pin bearing. This passage not only feeds the piston pin bearings but also provides lubrication for the pistons and cylinder walls. This system is used in virtually all engines that are equipped with full-floating piston pins.

2.12 Cooling System:

2.12.1 Air Cooled System:

Air cooled system is generally used in small engines say up to 15-20 Kw and in aero plane engines. In this system fins or extended surfaces are provided on the cylinder walls, cylinder head, etc. Heat generated due to combustion in the engine cylinder will be conducted to the fins and when the air flows over the fins, heat will be dissipated to air. The amount of heat dissipated to air depends upon : (a) Amount of air flowing through the fins. (b) Fin surface area. I Thermal conductivity of metal used for fins.

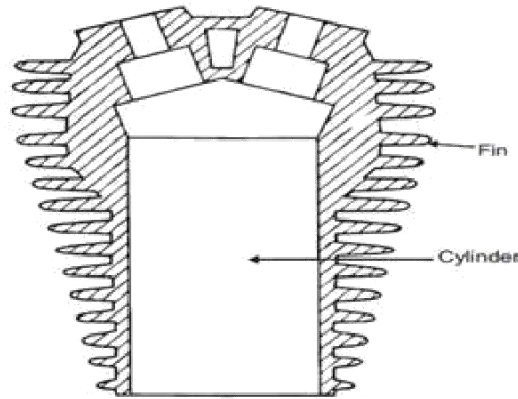


Fig 2.13 Air Cooled System

Advantages of Air Cooled System Following are the advantages of air cooled system: (a) Radiator/pump is absent hence the system is light. (b) In case of water cooling system there are leakages, but in this case there are no leakages. (c) Coolant and antifreeze solutions are not required. (d) This system can be used in cold climates, where if water is used it may freeze.

Disadvantages of Air Cooled System (a) Comparatively it is less efficient. (b) It is used only in aero planes and motorcycle engines where the engines are exposed to air directly.

2.12.2 Water Cooling System:

In this method, cooling water jackets are provided around the cylinder, cylinder head, valve seats etc. The water when circulated through the jackets, it absorbs heat of combustion. This hot water will then be cooling in the radiator partially by a fan and partially by the flow developed by the forward motion of the vehicle. The cooled water is again recirculated through the water jackets.

Thermo Siphon System: In this system the circulation of water is due to difference in temperature (i.e. difference in densities) of water. So in this system pump is not required but water is circulated because of density difference only.

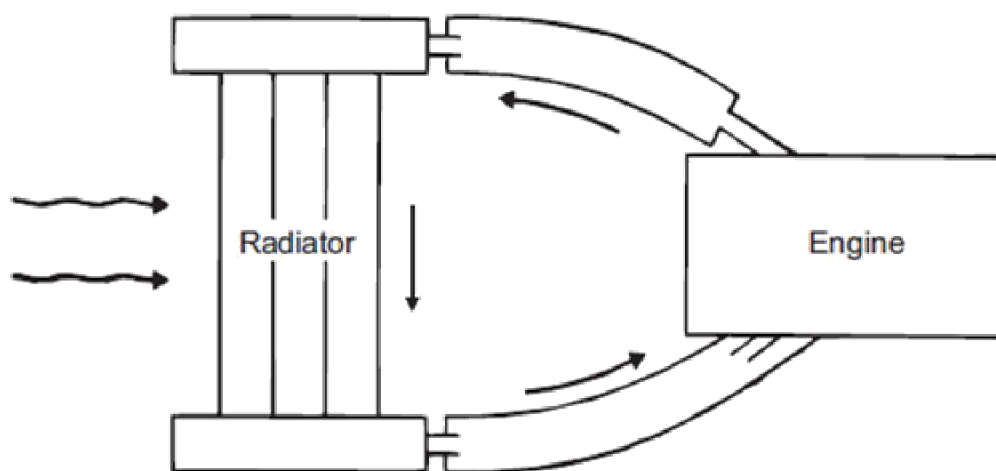


Fig 2.14 Thermo Siphon System

Pump Circulation System: In this system circulation of water is obtained by a pump. This pump is driven by means of engine output shaft through V-belts.

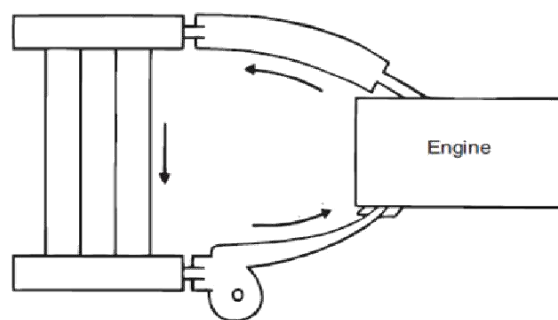


Fig 2.15 Pump Circulation System

Performance Calculation: Engine performance is an indication of the degree of success of the engine performs its assigned task, i.e. the conversion of the chemical energy contained in the fuel into the useful mechanical work. The performance of an engine is evaluated on the basis of the following : (a) Specific Fuel Consumption. (b) Brake Mean Effective Pressure. I Specific Power Output. (d) Specific Weight. (e) Exhaust Smoke and Other Emissions. The particular application of the engine decides the relative importance of these performance parameters. For Example : For an aircraft engine specific weight is more important whereas for an industrial engine specific fuel consumption is more important. For the evaluation of an engine performance few more parameters are chosen and the effect of various operating conditions, design concepts and modifications on these parameters are studied. The basic performance parameters are the

<http://www.francisxavier.ac.in>

following : (a) Power and Mechanical Efficiency. (b) Mean Effective Pressure and Torque. (c) Specific Output. (d) Volumetric Efficiency. (e) Fuel-air Ratio. (f) Specific Fuel Consumption. (g) Thermal Efficiency and Heat Balance. (h) Exhaust Smoke and Other Emissions. (i) Specific Weight. **Power and Mechanical Efficiency** The main purpose of running an engine is to obtain mechanical power. • Power is defined as the rate of doing work and is equal to the product of force and linear velocity or the product of torque and angular velocity. • Thus, the measurement of power involves the measurement of force (or torque) as well as speed. The force or torque is measured with the help of a dynamometer and the speed by a tachometer. The power developed by an engine and measured at the output shaft is called the brake power (bp) and is given by $bp = \frac{2\pi T n}{60}$ where, T is torque in N-m and N is the rotational speed in revolutions per minute. The total power developed by combustion of fuel in the combustion chamber is, however, more than the bp and is called indicated power (ip). Of the power developed by the engine, i.e. ip, some power is consumed in overcoming the friction between moving parts, some in the process of inducting the air and removing the products of combustion from the engine combustion chamber.

Indicated Power: It is the power developed in the cylinder and thus, forms the basis of evaluation of combustion efficiency or the heat release in the cylinder. Where, $I.P = \frac{P_m L A N k}{60}$ P_m = Mean effective pressure, N/m², L = Length of the stroke, m, A = Area of the piston, m², N = Rotational speed of the engine, rpm (It is N/2 for four stroke engine), and k = Number of cylinders. Thus, we see that for a given engine the power output can be measured in terms of mean effective pressure. The difference between the ip and bp is the indication of the power lost in the mechanical components of the engine (due to friction) and forms the basis of mechanical efficiency; which is defined as follows : Mechanical efficiency = $\frac{bp}{ip}$ The difference between ip and bp is called friction power (fp). $F_p = ip - bp$ Mechanical efficiency = $\frac{bp}{bp + fp}$

Mean Effective Pressure and Torque: Mean effective pressure is defined as a hypothetical/average pressure which is assumed to be acting on the piston throughout the power stroke. Therefore, $P_m = \frac{60 I.P}{L A N k}$ where, P_m = Mean effective pressure, N/m², I_p = Indicated power, Watt, L = Length of the stroke, m, A = Area of the piston, m², N = Rotational speed of the engine, rpm (It is N/2 for four stroke engine), and k = Number of cylinders. If the

mean effective pressure is based on b_p it is called the brake mean effective pressure (P_m), and if based on i_{hp} it is called indicated mean effective pressure (i_{mep}). Similarly, the friction mean effective pressure (f_{mep}) can be defined as, $f_{mep} = i_{mep} - b_{mep}$

The torque is related to mean effective pressure by the relation $B.P = 2\pi \text{Int}/60$ $I.P = P_m L A N k / 60$
 $2\pi \text{Int}/60 = [b_{mep} \cdot A \cdot L \cdot (Nk/60)]$ or, $T = (b_{mep} \cdot A \cdot L \cdot k) / 2\pi$

Thus, the torque and the mean effective pressure are related by the engine size. A large engine produces more torque for the same mean effective pressure. For this reason, torque is not the measure of the ability of an engine to utilize its displacement for producing power from fuel. It is the mean effective pressure which gives an indication of engine displacement utilization for this conversion. Higher the mean effective pressure, higher will be the power developed by the engine for a given displacement. Again we see that the power of an engine is dependent on its size and speed. Therefore, it is not possible to compare engines on the basis of either power or torque. Mean effective pressure is the true indication of the relative performance of different engines.

Specific Output: Specific output of an engine is defined as the brake power (output) per unit of piston displacement and is given by, $\text{Specific output} = B.P / A.L$ Constant = $b_{mep} \times \text{rpm}$ • The specific output consists of two elements – the b_{mep} (force) available to work and the speed with which it is working. • Therefore, for the same piston displacement and b_{mep} an engine operating at higher speed will give more output. • It is clear that the output of an engine can be increased by increasing either speed or b_{mep} . Increasing speed involves increase in the mechanical stress of various engine parts whereas increasing b_{mep} requires better heat release and more load on engine cylinder.

Volumetric Efficiency: Volumetric efficiency of an engine is an indication of the measure of the degree to which the engine fills its swept volume. It is defined as the ratio of the mass of air inducted into the engine cylinder during the suction stroke to the mass of the air corresponding to the swept volume of the engine at atmospheric pressure and temperature. Alternatively, it can be defined as the ratio of the actual volume inhaled during suction stroke measured at intake conditions to the swept volume of the piston. Volumetric efficiency, $\eta_v = \text{Mass of charge}$

actually sucked in Mass of charge corresponding to the cylinder intake The amount of air taken inside the cylinder is dependent on the volumetric efficiency of an engine and hence puts a limit on the amount of fuel which can be efficiently burned and the power output. For supercharged engine the volumetric efficiency has no meaning as it comes out to be more than unity.

Fuel-Air Ratio (F/A): Fuel-air ratio (F/A) is the ratio of the mass of fuel to the mass of air in the fuel-air mixture. Air-fuel ratio (A/F) is reciprocal of fuel-air ratio. Fuel-air ratio of the mixture affects the combustion phenomenon in that it determines the flame propagation velocity, the heat release in the combustion chamber, the maximum temperature and the completeness of combustion. Relative fuel-air ratio is defined as the ratio of the actual fuel-air ratio to that of the stoichiometric fuel-air ratio required to burn the fuel supplied. Stoichiometric fuel-air ratio is the ratio of fuel to air is one in which case fuel is completely burned due to minimum quantity of air supplied. Relative fuel-air ratio,
$$= (\text{Actual Fuel- Air ratio}) / (\text{Stoichiometric fuel-Air ratio})$$

Brake Specific Fuel Consumption: Specific fuel consumption is defined as the amount of fuel consumed for each unit of brake power developed per hour. It is a clear indication of the efficiency with which the engine develops power from fuel. B.S.F.C= Relative fuel-air ratio,
$$= (\text{Actual Fuel- Air ratio}) / (\text{Stoichiometric fuel-Air ratio})$$
 This parameter is widely used to compare the performance of different engines.

Thermal Efficiency and Heat Balance: Thermal efficiency of an engine is defined as the ratio of the output to that of the chemical energy input in the form of fuel supply. It may be based on brake or indicated output. It is the true indication of the efficiency with which the chemical energy of fuel (input) is converted into mechanical work. Thermal efficiency also accounts for combustion efficiency, i.e., for the fact that whole of the chemical energy of the fuel is not converted into heat energy during combustion. Brake thermal efficiency = $B.P / m_f \cdot C_v$ where, C_v = Calorific value of fuel, Kj/kg, and m_f = Mass of fuel supplied, kg/sec. • The energy input to the engine goes out in various forms – a part is in the form of brake output, a part into exhaust, and the rest is taken by cooling water and the lubricating oil. • The break-up of the total energy input into these different parts is called the heat balance. • The main components in a heat balance are brake output, coolant losses, heat going to exhaust, radiation and other losses. • Preparation of heat balance sheet gives us an idea about the amount of energy wasted in various

<http://www.francisxavier.ac.in>

parts and allows us to think of methods to reduce the losses so incurred.

Exhaust Smoke and Other Emissions: Smoke and other exhaust emissions such as oxides of nitrogen, unburned hydrocarbons, etc. are nuisance for the public environment. With increasing emphasis on air pollution control all efforts are being made to keep them as minimum as it could be. Smoke is an indication of incomplete combustion. It limits the output of an engine if air pollution control is the consideration.

Emission Formation Mechanisms: (S.I) This section discusses the formation of HC, CO, Nox, CO₂, and aldehydes and explains the effects of design parameters.

Hydrocarbon Emissions:

HC emissions are various compounds of hydrogen, carbon, and sometimes oxygen. They are burned or partially burned fuel and/or oil. HC emissions contribute to photochemical smog, ozone, and eye irritation. There are several formation mechanisms for HC, and it is convenient to think about ways HC can avoid combustion and ways HC can be removed; we will discuss each below. Of course, most of the HC input is fuel, and most of it is burned during “normal” combustion. However, some HC avoids oxidation during this process. The processes by which fuel compounds escape burning during normal S.I. combustion are:

1. Fuel vapor-air mixture is compressed into the combustion chamber crevice volumes.
2. Fuel compounds are absorbed into oil layers on the cylinder liner.
3. Fuel is absorbed by and/or contained within deposits on the piston head and piston crown.
4. Quench layers on the combustion chamber wall are left as the flame extinguishes close to the walls.
5. Fuel vapor-air mixture can be left unburned if the flame extinguishes before reaching the walls.
6. Liquid fuel within the cylinder may not evaporate and mix with sufficient air to burn prior to the end of combustion.
7. The mixture may leak through the exhaust valve seat.

(ii) Carbon Monoxide

Formation of CO is well established. Under some conditions, there is not enough O₂ available for complete oxidation and some of the carbon in the fuel ends up as CO. The amount of CO, for a range of fuel composition and C/H ratios, is a function of the relative air-fuel ratio. Even when enough oxygen is present, high peak temperatures can cause dissociation – chemical combustion reactions in which carbon dioxide and water vapor separate into CO, H₂, and O₂. Conversion of CO to CO₂ is governed by reaction $\text{CO} + \text{OH} \leftrightarrow \text{CO}_2 + \text{H}$ Dissociated CO may freeze during

<http://www.francisxavier.ac.in>

the expansion stroke. **(iii) Oxides of Nitrogen** Nox is a generic term for the compounds NO and NO₂. Both are present to some degree in the exhaust, and NO oxidizes to NO₂ in the atmosphere. Nox contributes to acid rain and photochemical smog; it is also thought to cause respiratory health problems at atmospheric concentrations found in some parts of the world. To understand Nox formation, we must recognize several factors that affect Nox equilibrium. Remember that all chemical reactions proceed toward equilibrium at some reaction rate. Equilibrium NO (which comprises most of the Nox formation) is formed at a rate that varies strongly with temperature and equivalence ratio. **(iv) Carbon Dioxide** While not normally considered a pollutant, CO₂ may contribute to the greenhouse effect. Proposals to reduce CO₂ emissions have been made. CO₂ controls strongly influence fuel economy requirements. **(v) Aldehydes** Aldehydes are the result of partial oxidation of alcohols. They are not usually present in significant quantities in gasoline-fueled engines, but they are an issue when alcohol fuels are used. Aldehydes are thought to cause lung problems. So far, little information of engine calibration effects on aldehyde formation is available.

2.13 Emission Formation in C.I. Engine:

For many years, diesel engines have had a reputation of giving poor performance and producing black smoke, an unpleasant odor, and considerable noise. However, it would find it difficult to distinguish today's modern diesel car from its gasoline counterpart. For diesel engines the emphasis is to reduce emissions of Nox and particulates, where these emissions are typically higher than those from equivalent port injected gasoline engines equipped with three-way catalysts. Catalyst of diesel exhaust remains a problem insofar as research has not yet been able to come up with an effective converter that eliminates both particulate matter (PM) and oxide of nitrogen (Nox).

2.14 Principle C.I. Engine Exhaust Constituents: For many years, diesel engines have had a reputation of giving poor performance and producing black smoke, an unpleasant odor, and considerable noise. However, it would find it difficult to distinguish today's modern diesel car from its gasoline counterpart. Concerning CO and HC emissions, diesel engines have an inherent advantages, therefore the emphasis is to reduce emissions of Nox and particulates, where these emissions are typically higher than those from equivalent port injected gasoline engines equipped

<http://www.francixavier.ac.in>

with three-way catalysts. Catalyst of diesel exhaust remains a problem insofar as research has not yet been able to come up with an effective converter that eliminates both particulate matter (PM) and oxide of nitrogen (Nox). In the same manner as with SI engines, the air/fuel ratio of the diesel engine has a significant impact on the level of pollutant concentrations but this parameter is not freely available for minimizing pollution. **Problems:** To determine Brake power, Indicated Power, Frictional Power, Brake Thermal Efficiency, Indicated Thermal Efficiency, Mechanical Efficiency, Relative Efficiency, Volumetric Efficiency, Brake Specific Fuel Consumption, Indicated Specific Fuel Consumption, Indicated mean effective pressure, Brake mean effective pressure.

2.15 Sample problems:

1. Following data relates to 4 cylinder, single stroke petrol engine. A/F ratio by weight 16:1. Calorific value of the fuel= 45200 KJ/kg, mechanical efficiency=82%. Air standard efficiency=52%, relative efficiency=70%, volumetric efficiency=78%, L/D=1.25, suction condition=1 bar, 25°C. Speed=2400 rpm and power at brakes=72kW. Calculate

1. Compression ratio
2. Indicated Thermal Efficiency
3. Brake specific fuel consumption
4. Bore and Stroke.

2. A six cylinder, 4 stroke SI engine having a piston displacement of 700cm³ per cylinder developed 78Kw at 3200 rpm and consumed 27 kg of petrol per hour. The calorific value of the fuel is 44MJ/kg. Estimate 1. The volumetric efficiency of the engine if the air-fuel ratio is 12 and intake air is at 0.9bar, 32°C. 2. Brake thermal efficiency and brake torque. For air R=0.287 KJ/kgK.

2.16 Solved Problems:

1. A trial carried out in a four stroke single cylinder gas engine gave the following results. Cylinder dia=300mm, Engine stroke=500mm, Clearance volume=6750cc, Explosions per minute=100 $p_{min} = 765 \text{ KN/m}^2$ Net work load on the brake=190kg Brake dia=1.5m Rope dia=25mm, Speed of the engine=240rpm, Gas used=30 $\text{m}^3/\text{kg hr}$, Calorific value of gas=20515 KJ/m^3 . Determine compression ratio, mechanical efficiency, indicated thermal efficiency, air standard efficiency, relative efficiency, assume $\gamma = 1.4$

GIVEN DATA:-

Dia of cylinder (d)=300mm=0.3m

Engine stroke(l)=500mm=0.5m

Clearance volume(v_c)=6750/100³= $6.75 \times 10^{-3} \text{ m}^3$

Explosions per minute(n)=100/minute=1.67/sec

$P_{min}=765 \text{ KN/m}^2$

Brake drum dia(D_1)=1.5m

Rope dia(d_1)=0.025m

Work load on the brake(w)=190kg=1.86KN

TO FIND:-

Compression ratio (r)

Mechanical efficiency (η_{mech})

Indicated thermal efficiency (η_{it})

Air standard efficiency (η_{air})

Relative efficiency (η_{rel})

<http://www.francixavier.ac.in>

SOLUTION:-

(1).Compression Ratio (r):-

$$\begin{aligned}
 r &= \left(\frac{v_s}{v_c} \right) + 1 \\
 &= \left(\frac{1 \times a}{v_c} \right) + 1 \\
 &= \frac{0.5 \times \left(\frac{\pi}{4} \right) 0.3^2}{0.75 \times 10^{-3}} + 1 \\
 &= 5.23 + 1
 \end{aligned}$$

$$(r) = 6.23$$

(2).Air Standard Efficiency (η_{air}):-

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

$$= 1 - \left(\frac{1}{6.23^{1.4-1}} \right)$$

$$= 51.89\%$$

(3). Indicated Thermal Efficiency (η_{it}):-

$$(\eta_{it}) = \frac{IP}{P_C \times C_V}$$

Here, indicated power (IP) = $p_{mi} \times l \times a \times n \times k$

$$= 765 \times 0.5 \times 0.0706 \times 1.67 \times 1$$

$$= 45.09 \text{ KW}$$

Therefore,

$$\eta_{it} = \frac{45.09}{\left(\frac{30}{3600} \right) \times 22515}$$

$$= 24.03\%$$

(4). Relative Efficiency (η_{rel}):-

$$(\eta_{rel}) = \frac{\eta_{it}}{\eta_{air}}$$

$$= \frac{24.03}{51.89}$$

$$= 46.30\%$$

(5). Mechanical Efficiency (η_{mech}):-

$$(\eta_{mech}) = \frac{\eta_{BT}}{\eta_{it}}$$

$$= \frac{18.99}{24.03}$$

$$= 79.02\%$$

2. The following observations are recorded during a test on a four-stroke petrol engine, F.C = 3000 of fuel in 12sec, speed of the engine is 2500rpm, B.P = 20KW, Air intake orifice diameter = 35mm, Pressure across the orifice = 140mm of water coefficient of discharge of orifice = 0.6, piston diameter = 150mm, stroke length = 100mm, Density of the fuel = 0.85gm/cc, $r=6.5$, C_v of fuel = 42000KJ/Kg, Barometric pressure = 760mm of Hg, Room temperature = 24°C

Determine:

- (i) Volumetric efficiency on the air basis alone
- (ii) Air-fuel ratio
- (iii) The brake mean effective pressure
- (iv) The relative efficiency on the brake thermal efficiency

Given data:

$$\text{Fuel consumption} = 30\text{cc in 12sec} = \frac{30}{12} \times 3600 \left(\frac{\text{cc}}{\text{hr}} \right)$$

$$\text{Speed (N)} = 2500/60 \text{ rps}$$

$$\text{Brake power} = 20\text{KW}$$

$$\text{Orifice diameter (d}_o\text{)} = 0.035\text{m}$$

$$\text{Pressure across the orifice (P}_o\text{)} = 140\text{mm of water}$$

$$\text{Coefficient of discharge (C}_d\text{)} = 0.6$$

$$\text{Piston diameter (d)} = 150\text{mm} = 0.15\text{m}$$

$$\text{Stroke length (l)} = 0.1\text{m}$$

$$\text{Density of fuel (}\rho\text{)} = 0.85\text{gm/cc}$$

$$\text{Compression ratio (r)} = 6.5$$

$$\text{Room temperature (T}_a\text{)} = 297\text{K}$$

$$\text{Barometric pressure} = 760\text{mm of Hg} = 101.325\text{KK/m}^2 = 10.34\text{m of water}$$

To find:

- (i) Volumetric efficiency on the air basis alone
- (ii) Air-fuel ratio

- (iii) The brake mean effective pressure
- (iv) The relative efficiency on the brake thermal efficiency

Solution:

$$10.34\text{m of water} = 101.325\text{KN/m}^2$$

$$\text{Pressure head} = \frac{P_o}{\rho \times g}$$

$$P_o = 0.14\text{m of water}$$

$$= \frac{101.325}{10.34} \times 0.14$$

$$P_o = 1372\text{N/m}^2$$

$$\text{Density of gas } (\rho) = P/RT$$

$$= \frac{101.325}{0.287 \times 297}$$

$$\rho = 1.1887\text{Kg/m}^3$$

$$\text{Pressure head (h)} = \frac{1372}{1.1887 \times 9.81}$$

$$h = 117.6557\text{m}$$

$$Q_{\text{air}} = C_d \times a \times \sqrt{2gh}$$

$$= 0.6 \times \frac{\pi}{4} (0.035)^2 \sqrt{2 \times 9.81 \times 117.6557}$$

$$= 0.02774 \text{ m}^3/\text{sec}$$

$$\text{No. of. Suction strokes per second} = \frac{N}{2} = \frac{2500}{60 \times 2} = 20.8333$$

$$\text{Air consumptions per stroke} = \frac{0.02774}{20.8333}$$

$$= 0.001332\text{m}^3$$

$$\text{Stroke volume (Vs)} = \frac{\pi}{4} \times (0.15)^2 \times 0.1 = 0.001767 \text{ m}^3$$

$$\text{Volumetric efficiency } (\eta_{\text{vol}}) = \frac{0.001332}{0.001767} \times 100\%$$

$$\eta_{\text{vol}} = 75.382\%$$

$$\begin{aligned}\text{volume of air consumed } V_{\text{air}} &= Q_{\text{air}} = 0.02774 \text{ m}^3/\text{sec} \\ &= 0.02774 \times 3600 \text{ m}^3/\text{hr}\end{aligned}$$

$$\begin{aligned}\text{Mass of air consumed } (m_a) &= V_a \times \rho_a = 99.864 \times 1.1887 \\ &= 118.71 \text{ Kg/hr}\end{aligned}$$

Fuel consumption = 9000cc/hr

$$\text{Mass of the fuel consumed } (m_f) = 9000 \times 0.85 = 7.65 \text{ Kg/hr}$$

$$\begin{aligned}\text{Air fuel ratio} &= \frac{m_a}{m_f} = \frac{118.71}{7.65} = 15.518 : 1\end{aligned}$$

$$\text{Brake power (B.P)} = 20 \text{ KW} = P_{\text{mb}} \times l \times a \times n \times k$$

$$\begin{aligned}P_{\text{mb}} &= \frac{20}{0.001767 \times 20.833 \times 1} \\ &= 543.294 \text{ KN/m}^2\end{aligned}$$

$$\begin{aligned}\text{Air standard efficiency } (\eta_{\text{air}}) &= 1 - \frac{1}{(r)^{\gamma-1}} \\ &= 1 - \frac{1}{(6.5)^{1.4-1}} \\ &= 52.703\%\end{aligned}$$

$$\text{Brake thermal efficiency } (\eta_{\text{BT}}) = \frac{BP \times 3600}{F.C \times C.V} = 22.4\%$$

$$\text{Relative efficiency on brake thermal efficiency basis } (\eta_{\text{rel}}) = \eta_{\text{BT}} / \eta_{\text{air}}$$

$$= 0.22409 / 0.52703$$

$$\eta_{\text{rel}} = 42.52\%$$

2.17 TWO MARK UNIVERSITY QUESTIONS:

1. Classify IC engine according to cycle of lubrication system and field of application.

Types of lubrication system

2. List the various components of IC engines.

3. Name the basic thermodynamic cycles of the two types of internal combustion reciprocating engines.
4. Mention the important requires of liner material.
5. State the purpose of providing piston in IC engines.
6. Define the terms as applied to reciprocating I.C. engines "Mean effective pressure" and "Compression ratio".
7. What is meant by highest useful compression ratio?
8. What are the types of piston rings?
9. What is the use of connecting rod?
10. What is the use of flywheel?

2.18 UNIVERSITY ESSAY QUESTIONS

1. Explain full pressure lubrication system I.C Engine. (16)
2. Explain the water cooling system in I.C Engine. (16)
3. Explain the 2 types of Ignition system In 5.1 Engine. (16)
4. Draw and explain the valve timing diagram of 4 strokes Diesel Engine. (16)
5. Draw and explain the port timing diagram of 2stroke Petrol Engine. (16)
6. Explain with neat sketch the exhaust gas analysis. (16)
7. The following results refer to a test on a petrol engine Indicated power = 30 Kw, Brake power = 26 Kw, Engine speed = 1000 rpm Fuel brake power/ hour = 0.35 kg Calorific value of fuel = 43900kj/kg .Calculate the indicated Thermal efficiency, the brake Thermal efficiency and Mechanical efficiency (16)
8. A four cylinder 2 stroke cycle petrol engine develops 23.5 kw brake power at 2500 rpm. The mean effective pressure on each piston in 8. 5 bar and mechanical efficiency in 85% Calculate the diameter and stroke of each cylinder assuming the length of stroke equal to 1.5 times the diameter of cylinder. (16)
9. The following data to a particular twin cylinder two stroke diesel engine. Bore 15 cm stroke. 20 cm. speed 400 rpm. Indicated mean effective pressure 4 bar, dead weight on the brake drum 650 N. spring balance reading 25 N Diameter of the brake drum 1 m .Fuel consumption 0.075 kg/min and calorific value of the fuel is 44500 KJ/kg. Determine 1. Indicated Power 2. Brake Power 3. Mechanical efficiency 4. Indicated thermal efficiency and 5. Brake thermal efficiency (16)

ME1251 THERMAL ENGINEERING

UNIT III

NOZZLES, TURBINES & STEAM POWER CYCLES

CONTENTS

- 3.1 Flow of steam through nozzles:
- 3.2 Continuity and steady flow energy equations
- 3.3 Types of Nozzles
 - 3.3.1 Convergent Nozzle
 - 3.3.2 Divergent Nozzle
 - 3.3.4 Convergent-Divergent Nozzle
- 3.4 Supersaturated flow or Meta stable flow in Nozzles
- 3.5 Mass of steam discharged through nozzle
- 3.6 Steam Turbines
 - 3.6.1 Impulse Turbines
 - 3.6.2 Reaction Turbines
- 3.7 Compounding of impulse turbine
 - 3.7.1. Velocity Compounding
 - 3.7. 2. Pressure Compounding
 - 3.7. 3. Pressure-Velocity Compounding
- 3.8 Velocity diagram of an impulse turbine
- 3.9 Velocity diagram of the velocity compounded turbines
- 3.10 Governing of Steam Turbine
 - 3.10.1. Throttle Governing
 - 3.10.2. Nozzle Governing
- 3.11 Solved Problems
- 3.12 Two Marks University Questions
- 3.13 University Essay Questions

TECHNICAL TERMS:

1. **Wet steam:** The steam which contains some water particles in superposition.
2. **Dry steam / dry saturated steam:**

When whole mass of steam is converted into steam then it is called as dry steam.
3. **Super heated steam:** When the dry steam is further heated at constant pressure, the temperature increases the above saturation temperature. The steam has obtained is called super heated steam.
4. **Degree of super heat:** The difference between the temperature of saturated steam and saturated temperature is called degree of superheat.
5. **Nozzle:** It is a duct of varying cross sectional area in which the velocity increases with the corresponding drop in pressure.
6. **Coefficient of nozzle:** It is the ratio of actual enthalpy drop to isentropic enthalpy drop.
7. **Critical pressure ratio:** There is only one value of ratio (P_2/P_1) which produces maximum discharge from the nozzle . then the ratio is called critical pressure ratio.
8. **Degree of reaction:** It is defined as the ratio of isentropic heat drop in the moving blade to isentropic heat drop in the entire stages of the reaction turbine.
9. **Compounding:** It is the method of absorbing the jet velocity in stages when the steam flows over moving blades. (i) Velocity compounding (ii) Pressure compounding and (iii) Velocity-pressure compounding
10. **Enthalpy:** It is the combination of the internal energy and the flow energy.
11. **Entropy:** It is the function of quantity of heat with respect to the temperature.
12. **Convergent nozzle:** The cross sectional area of the duct decreases from inlet to the outlet side then it is called as convergent nozzle.

- 13. Divergent nozzle:** The crosssectional area of the duct increases from inlet to the outlet then it is called as divergent nozzle.

FRANCIS XAVIER ENGINEERING COLLEGE

UNIT-III

NOZZLES, TURBINES & STEAM POWER CYCLES

3.1 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.

3.2 Continuity and steady flow energy equations

Through a certain section of the nozzle: $m \cdot v = A \cdot C$ m is the mass flow rate, v is the specific volume, A is the cross-sectional area and C is the velocity. For steady flow of steam through a certain apparatus, principle of conservation of energy states:

$$h_1 + C_1^2/2 + gz_1 + q = h_2 + C_2^2/2 + gz_2 + w$$

For nozzles, changes in potential energies are negligible, $w = 0$ and $q \approx 0$. $H_1 + C_1^2/2 = h_2 + C_2^2/2$

3.3 Types of Nozzles:

1. Convergent Nozzle
2. Divergent Nozzle
3. Convergent-Divergent Nozzle

3.3.1 Convergent Nozzle:

A typical convergent nozzle is shown in fig. 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.

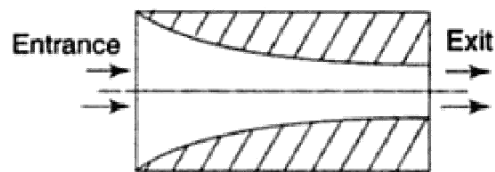


Fig 3.1 Convergent Nozzle:

3.3.2 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.

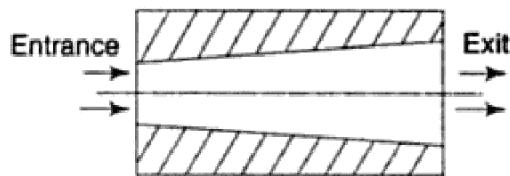


Fig 3.2 Divergent Nozzle:

3.3.4 Convergent-Divergent Nozzle:

In this case, the cross sectional area first decreases from its entrance to throat, and then increases from throat to exit. It is widely used in many type of steam turbines.

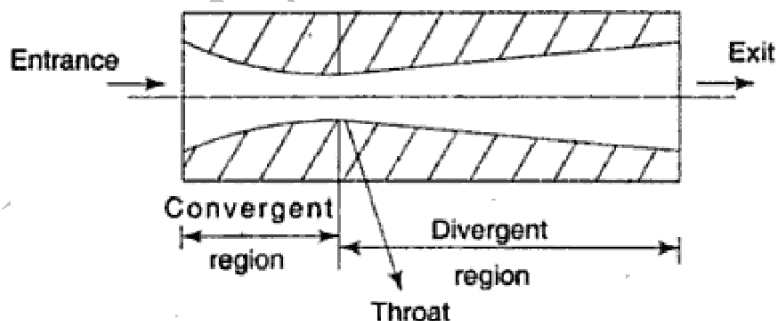


Fig. 3.3 Convergent-Divergent Nozzle

3.4 Supersaturated flow or Meta stable 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 super saturation, 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 super saturation phenomenon is shown on the h-s chart below:

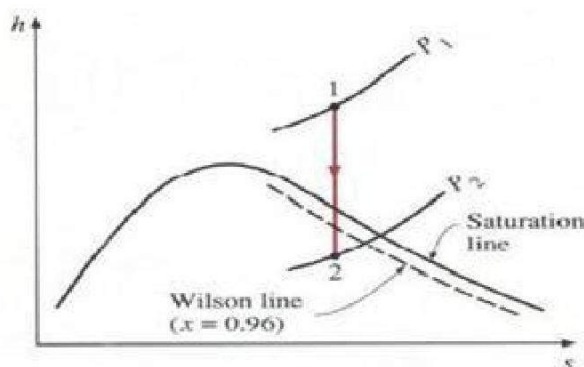


Fig 3.4 The h-s diagram for the isentropic expansion of steam in a nozzle.

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.

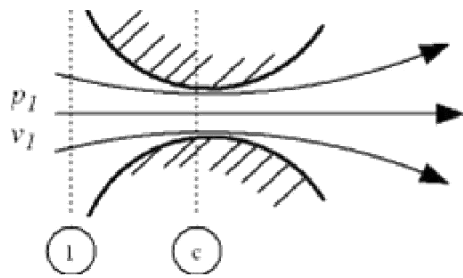


Fig 3.5 Critical flow nozzles

The ratio between the critical pressure and the initial pressure for a nozzle can be expressed as $P_c / p_1 = (2 / (n + 1))^{n / (n - 1)}$

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 super-heated: $n = 1.30$
- 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%. The velocity of steam will be then

$$V_2 = 44.72\sqrt{K(H_1 - H_2)}$$

Where, k is the co-efficient which allows for friction loss. It is also known as nozzle efficiency.

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

$$V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n} \text{ m/s}$$

3.5 Mass of steam discharged through 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.

3.6 STEAM TURBINES: Normally the turbines are classified into types,

1. Impulse Turbine
2. Reaction Turbine

Impulse and Reaction Turbines:

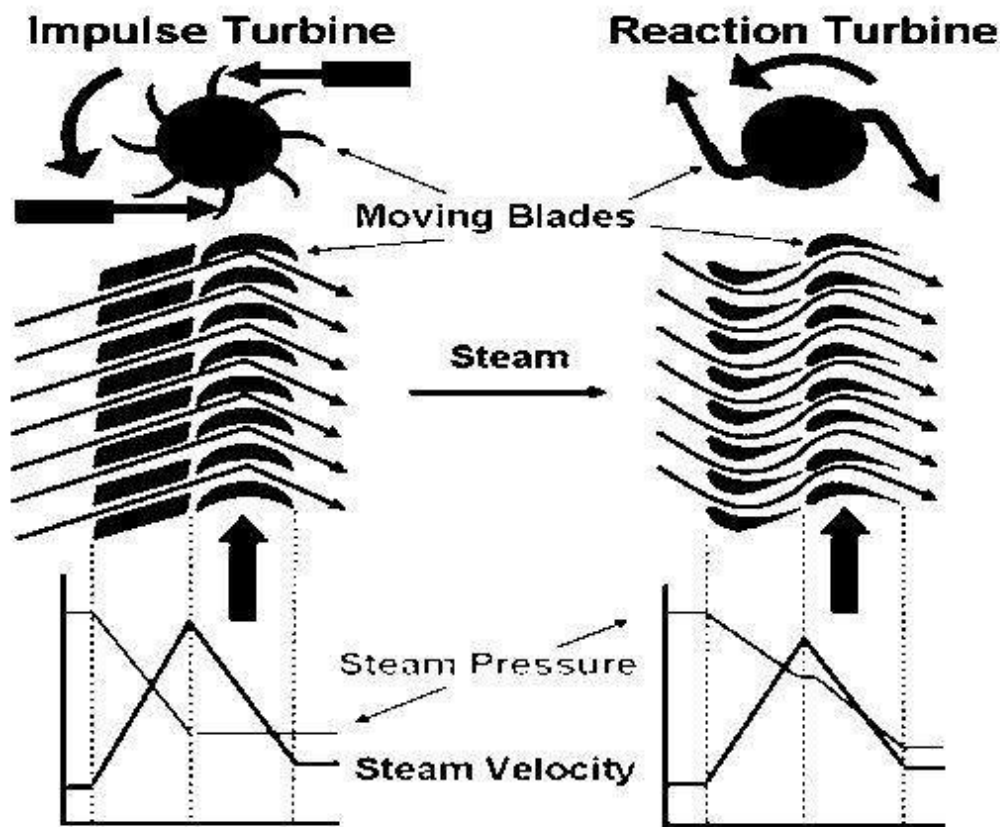


Fig 3.6 impulse turbine and reaction turbine pressure and velocity diagram

3.6.1 Impulse Turbines:

The steam jets are directed at the turbines bucket shaped rotor blades where the pressure exerted by the jets causes the rotor to rotate and the velocity of the steam to reduce as it imparts its kinetic energy to the blades. The blades in turn change the direction of flow of the steam however its pressure remains constant as it passes through the rotor blades since the cross section

of the chamber between the blades is constant. Impulse turbines are therefore also known as constant pressure turbines. The next series of fixed blades reverses the direction of the steam before it passes to the second row of moving blades.

3.6.2 Reaction Turbines

The rotor blades of the reaction turbine are shaped more like aero foils, arranged such that the cross section of the chambers formed between the fixed blades diminishes from the inlet side towards the exhaust side of the blades. The chambers between the rotor blades essentially form nozzles so that as the steam progresses through the chambers its velocity increases while at the same time its pressure decreases, just as in the nozzles formed by the fixed blades. Thus the pressure decreases in both the fixed and moving blades. As the steam emerges in a jet from between the rotor blades, it creates a reactive force on the blades which in turn creates the turning moment on the turbine rotor, just as in Hero's steam engine. (Newton's Third Law – For every action there is an equal and opposite reaction).

3.7 Compounding of impulse turbine:

- This is done to reduce the rotational speed of the impulse turbine to practical limits. (A rotor speed of 30,000 rpm is possible, which is pretty high for practical uses.) - Compounding is achieved by using more than one set of nozzles, blades, rotors, in a series, keyed to a common shaft; so that either the steam pressure or the jet velocity is absorbed by the turbine in stages. - Three main types of compounded impulse turbines are: a) Pressure compounded, b) velocity compounded and c) pressure and velocity compounded impulse turbines.

3.7.1. Velocity Compounding:

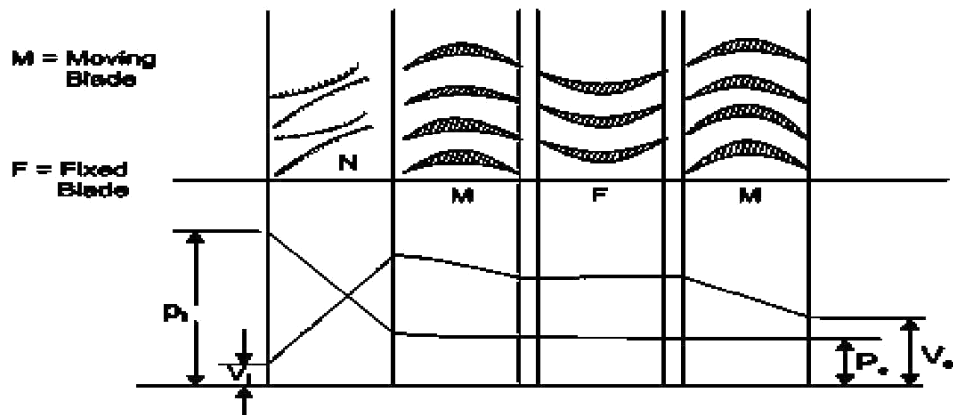


Fig 3.7 Velocity Compounding diagram

P_i = Inlet Pressure, P_e = Exit Pressure, V_i = Inlet Velocity, V_e = Exit Velocity.

The velocity-compounded impulse turbine was first proposed by C.G. Curtis to solve the problems of a single-stage impulse turbine for use with high pressure and temperature steam. The Curtis stage turbine, as it came to be called, is composed of one stage of nozzles as the single-stage turbine, followed by two rows of moving blades instead of one. These two rows are separated by one row of fixed blades attached to the turbine stator, which has the function of redirecting the steam leaving the first row of moving blades to the second row of moving blades. A Curtis stage impulse turbine is shown in Fig. with schematic pressure and absolute steam-velocity changes through the stage. In the Curtis stage, the total enthalpy drop and hence pressure drop occur in the nozzles so that the pressure remains constant in all three rows of blades.

3.7.2. Pressure Compounding:

-This involves splitting up of the whole pressure drop from the steam chest pressure to the condenser pressure into a series of smaller pressure drops across several stages of impulse turbine. -The nozzles are fitted into a diaphragm locked in the casing. This diaphragm separates one wheel chamber from another. All rotors are mounted on the same shaft and the blades are attached on the rotor.

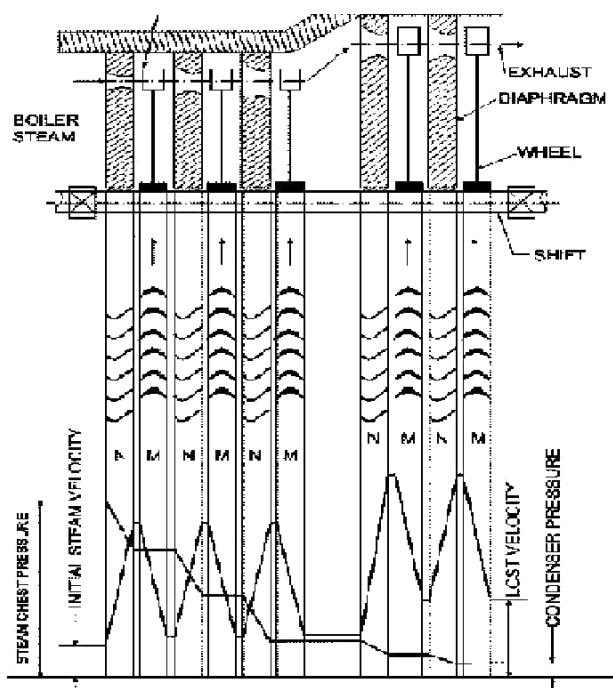


Fig 3.8 Pressure Compounding diagram

3.7.3. Pressure-Velocity Compounding

This is a combination of pressure and velocity compounding. A two-row velocity compounded turbine is found to be more efficient than the three-row type. In a two-step pressure velocity compounded turbine, the first pressure drop occurs in the first set of nozzles, the resulting gain in the kinetic energy is absorbed successively in two rows of moving blades before the second pressure drop occurs in the second set of nozzles. Since the kinetic energy gained in each step is absorbed completely before the next pressure drop, the turbine is pressure compounded and as well as velocity compounded. The kinetic energy gained due to the second pressure drop in the second set of nozzles is absorbed successively in the two rows of moving blades.

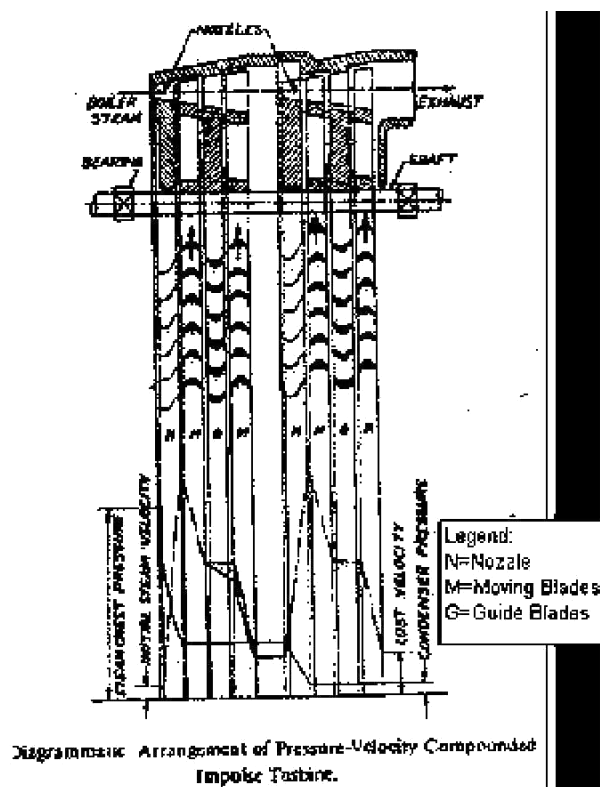


Fig 3.9 Pressure-Velocity Compounding diagram

The pressure velocity compounded steam turbine is comparatively simple in construction and is much more compact than the pressure compounded turbine.

3.8 Velocity diagram of an impulse turbine:

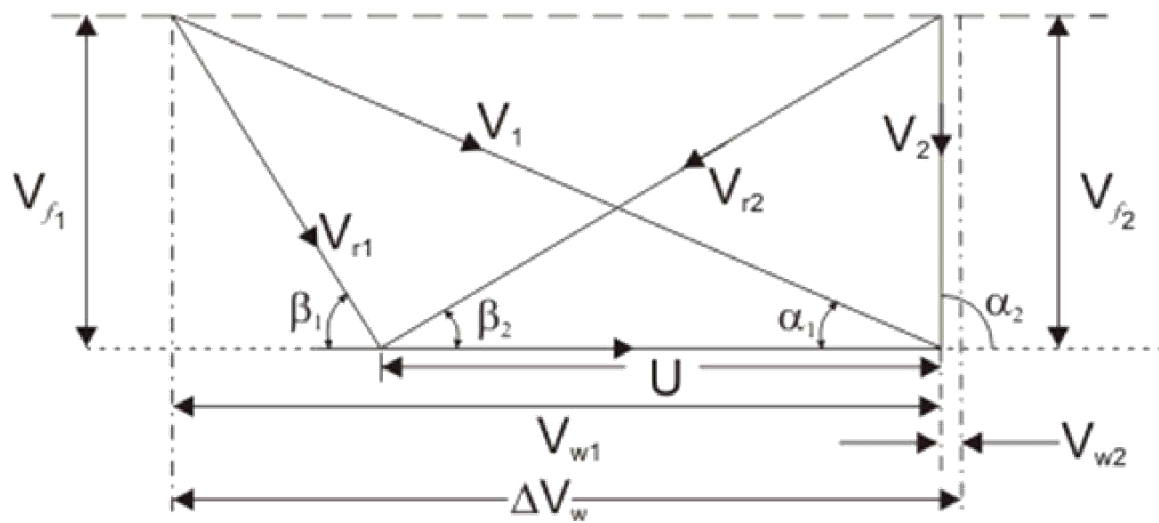


Fig 3.10 Velocity diagram of an impulse turbine

V_1 and V_2 = Inlet and outlet absolute velocity

V_{r1} and V_{r2} = Inlet and outlet relative velocity (Velocity relative to the rotor blades.)

U = mean blade speed

α_1 = nozzle angle, α_2 = absolute fluid angle at outlet

It is to be mentioned that all angles are with respect to the tangential velocity (in the direction of U)

β_1 and β_2 = Inlet and outlet blade angles

V_{w1} and V_{w2} = Tangential or whirl component of absolute velocity at inlet and outlet

V_{f1} and V_{f2} = Axial component of velocity at inlet and outlet

Tangential force on a blade,

$$F_u = \dot{m} (V_{w1} - V_{w2})$$

(mass flow rate X change in velocity in tangential direction)

or,

$$F_u = \dot{m} \Delta V_w$$

FRANCIS XAVIER

$$\text{Power developed} = \dot{m} U \Delta V_w$$

Blade efficiency or Diagram efficiency or Utilization factor is given by

$$\eta_b = \frac{\dot{m} \cdot U \cdot \Delta V_w}{m(V_1^2/2)} = \frac{\text{Workdone}}{\text{K.E. supplied}}$$

Or,

$$\begin{aligned} \eta_b &= \frac{2U\Delta V_w}{V_1^2} \\ \text{stage efficiency } \eta_s &= \frac{\text{Work done by the rotor}}{\text{Isentropic enthalpy drop}} \\ \eta_s &= \frac{\dot{m} U \Delta V_w}{\dot{m}(\Delta H)_{isen}} = \frac{\dot{m} U \Delta V_w}{\dot{m} \left(\frac{V_1^2}{2} \right)} \cdot \frac{\dot{m}(V_1^2/2)}{\dot{m}(\Delta H)_{isen}} \\ \text{or,} \quad \eta_s &= \eta_b \times \eta_n \quad [\eta_n = \text{Nozzle efficiency}] \end{aligned}$$

Optimum blade speed of a single stage turbine

$$\begin{aligned} \Delta V_w &= V_{r1} \cos \beta_1 + V_{r2} \cos \beta_2 \\ &= V_{r1} \cos \beta_1 + \left(1 + \frac{V_{r2}}{V_{r1}} \cdot \frac{\cos \beta_2}{\cos \beta_1} \right) \\ &= (V_1 \cos \alpha_1 - U) + (1 + kc) \end{aligned}$$

where, $k = (V_{r2}/V_{r1}) = \text{friction coefficient}$

$$c = (\cos \beta_2 / \cos \beta_1)$$

$$\eta_b = \frac{2U\Delta V_w}{V_1^2} = 2 \frac{U}{V_1} \left(\cos \alpha_1 - \frac{U}{V_1} \right) (1 + kc)$$

 3.19

$$\rho = \frac{U}{V_1} = \frac{\text{Blade speed}}{\text{Fluid velocity at the blade inlet}} = \text{Blade speed ratio}$$

$$\eta_b \text{ is maximum when } \frac{d\eta_b}{d\rho} = 0 \quad \text{also} \quad \frac{d^2\eta_b}{d\rho} = -4(1+kc)$$

$$\text{or,} \quad \frac{d}{d\rho} \{ 2(\rho \cos \alpha_1 - \rho^2)(1+kc) \} = 0$$

$$\text{or,} \quad \rho = \frac{\cos \alpha_1}{2}$$

α_1 is of the order of 18° to 22°

$$\text{Now,} \quad (\rho)_{opt} = \left(\frac{U}{V_1} \right)_{opt} = \frac{\cos \alpha_1}{2} \quad (\text{For single stage impulse turbine})$$

\therefore The maximum value of blade efficiency

$$\begin{aligned} (\eta_b)_{max} &= 2(\rho \cos \alpha_1 - \rho^2)(1+kc) \\ &= \frac{\cos^2 \alpha_1}{2}(1+kc) \end{aligned}$$

For equiangular blades,

$$(\eta_b)_{max} = \frac{\cos^2 \alpha_1}{2}(1+k)$$

If the friction over blade surface is neglected

$$(\eta_b)_{max} = \cos^2 \alpha_1$$

The fixed blades are used to guide the outlet steam/gas from the previous stage in such a manner so as to smooth entry at the next stage is ensured.

K, the blade velocity coefficient may be different in each row of blades

$$\text{Work done} = \dot{m} \cdot U (\Delta V_{w1} + \Delta V_{w2})$$

$$\text{End thrust} = \dot{m} (\Delta V_{f1} + \Delta V_{f2})$$

The optimum velocity ratio will depend on number of stages and is given by $P_{opt} = \frac{\cos \alpha_{11}}{2n}$

3.9 Velocity diagram of the velocity compounded turbines:

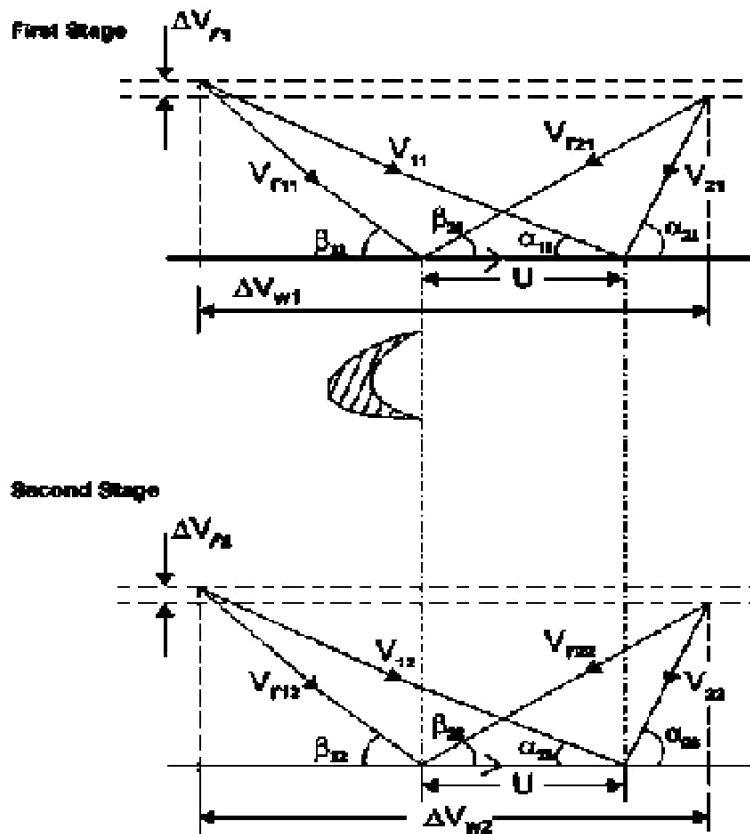


Fig 3.11 Velocity diagram of the velocity compounded turbines

Reaction Turbine:

A **reaction turbine**, therefore, is one that is constructed of rows of fixed and rows of moving blades. The fixed blades act as nozzles. The moving blades move as a result of the impulse of steam received (caused by change in momentum) and also as a result of expansion and acceleration of the steam relative to them. In other words, they also act as nozzles. The enthalpy drop per stage of one row fixed and one row moving blades is divided among them, often equally. Thus a blade with a 50 percent degree of reaction, or a 50 percent reaction stage, is one in which half the enthalpy drop of the stage occurs in the fixed blades and half in the moving blades. The pressure drops will not be equal, however. They are greater for the fixed blades and greater for the high-pressure than the low-pressure stages. The moving blades of a reaction turbine are easily distinguishable from those of an impulse turbine in that they are not symmetrical and, because they act partly as nozzles, have a shape similar to that of the fixed blades, although curved in the opposite direction. The schematic pressure line in figure shows that pressure continuously drops through all rows of blades, fixed and moving. The absolute steam velocity changes within each stage as shown and repeats from stage to stage. The second figure shows a typical velocity diagram for the reaction stage.

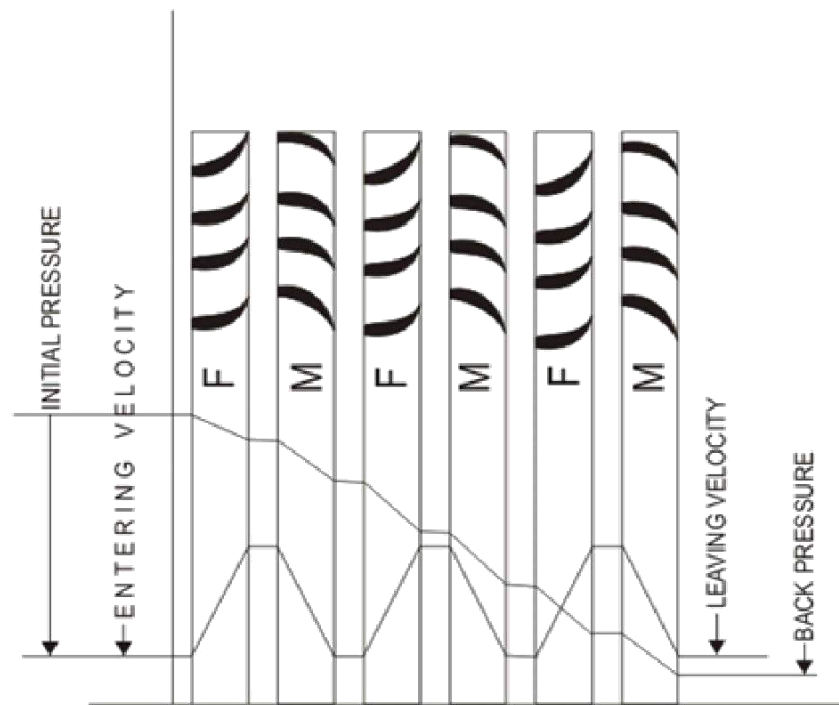


Fig 3.12 reaction turbine pressure & velocity diagram

Pressure and enthalpy drop both in the fixed blade or **stator** and in the moving blade or **Rotor**

$$\text{Degree of Reaction} = \frac{\text{Enthalpy drop in Rotor}}{\text{Enthalpy drop in Stage}}$$

or,

$$R = \frac{h_1 - h_2}{h_0 - h_1}$$

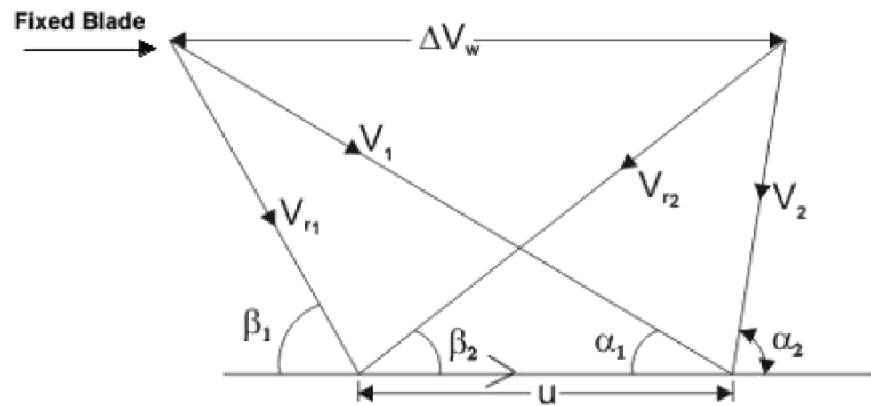


Fig 3.13 velocity diagram

A very widely used design has half degree of reaction or 50% reaction and this is known as Parson's Turbine. This consists of symmetrical stator and rotor blades.

The velocity triangles are symmetrical and we have

$$\alpha_1 = \beta_2 \quad , \quad \beta_1 = \alpha_2$$

$$V_1 = V_{r2} \quad , \quad V_{r1} = V_2$$

Energy input per stage (unit mass flow per second)

$$E = \frac{V_1^2}{2} + \frac{V_{r2}^2 - V_{r1}^2}{2}$$

$$E = V_1^2 - \frac{V_{r1}^2}{2}$$

$$E = V_1^2 - \frac{V_1^2}{2} - \frac{U^2}{2} + \frac{2V_1U \cos \alpha_1}{2}$$

$$E = (V_1^2 - U^2 + 2V_1U \cos \alpha_1)/2$$

From the inlet velocity triangle we have,

$$V_{r1}^2 = V_1^2 - U^2 - 2V_1U \cos \alpha_1$$

Work done (for unit mass flow per second) = $W = U \Delta V_w$

$$= U(2V_1 \cos \alpha_1 - U)$$

Therefore, the blade efficiency

$$\eta_b = \frac{2U(2V_1 \cos \alpha_1 - U)}{V_1^2 - U^2 + 2V_1U \cos \alpha_1}$$

3.10 Governing of Steam Turbine: The method of maintaining the turbine speed constant irrespective of the load is known as governing of turbines. The device used for governing of turbines is called Governor. There are 3 types of governors in steam turbine,

1. Throttle governing
2. Nozzle governing
3. By-pass governing

3.10.1. Throttle Governing:

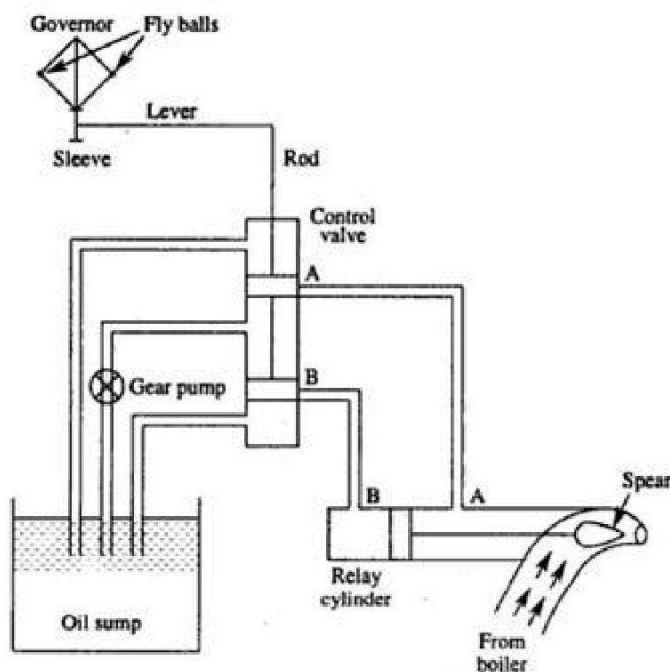


Fig 3.14 Throttle Governing

Let us consider an instant when the load on the turbine increases, as a result the speed of the turbine decreases. The fly balls of the governor will come down. The fly balls bring down the sleeve. The downward movement of the sleeve will raise the control valve rod. The mouth of the pipe AA will open. Now the oil under pressure will rush from the control valve to right side of piston in the relay cylinder through the pipe AA. This will move the piston and spear towards the

left which will open more area of nozzle. As a result steam flow rate into the turbine increases, which in turn brings the speed of the turbine to the normal range.

3.10.2. Nozzle Governing:

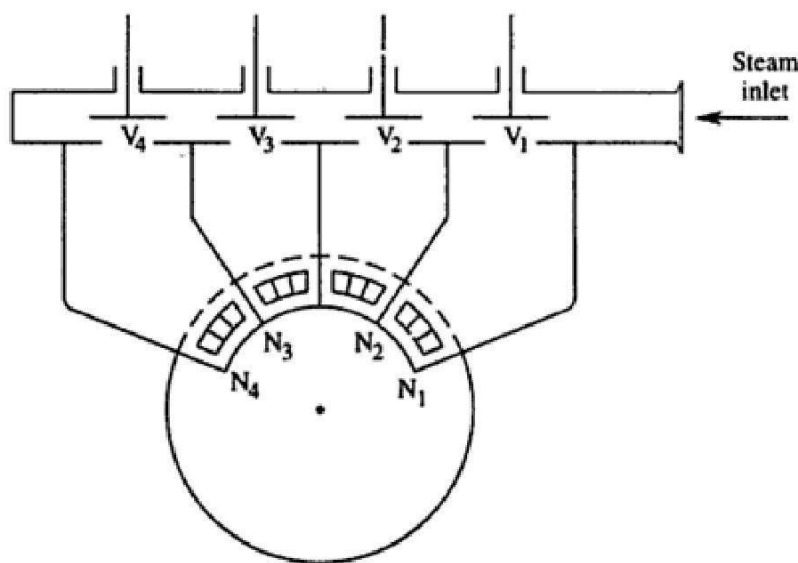


Fig 3.15 Nozzle Governing

A dynamic arrangement of nozzle control governing is shown in fig.

In this nozzles are grouped in 3 to 5 or more groups and each group of nozzle is supplied steam controlled by valves. The arc of admission is limited to 180° or less. The nozzle controlled governing is restricted to the first stage of the turbine, the nozzle area in other stages remaining constant. It is suitable for the simple turbine and for larger units which have an impulse stage followed by an impulse reaction turbine.

3.11 Solved Problems:

1. A convergent divergent adiabatic steam nozzle is supplied with steam at 10 bar and 250°C. the discharge pressure is 1.2 bar. assuming that the nozzle efficiency is 100% and initial velocity of steam is 50 m/s. find the discharge velocity.

Given Data:-

Initial pressure(p_1)=10bar

Initial temperature(T_1)=250°C

Exit pressure(p_2)=1.2 bar

Nozzle efficiency(η_{nozzle})=100%

Initial velocity of steam(v_1)=50m/s

To Find:-

Discharge velocity (v_2)

Solution:-

From steam table, For 10 bar, 250°C,

$$h_1 = 2943 \text{ KJ/kg}$$

$$s_1 = 6.926 \text{ KJ/kgk}$$

From steam table, For 1.2 bar,

$$h_{f2} = 439.3 \text{ KJ/kg} ; h_{fg2} = 2244.1 \text{ KJ/kg};$$

$$s_{f2} = 1.361 \text{ KJ/kg K} ; s_{fg2} = 5.937 \text{ KJ/kgK}.$$

Since $s_1 = s_2$,

$$S_1 = S_{f2} + X_2 S_{fg2}$$

$$6.926 = 1.361 + x_2(5.937)$$

$$X_2 = 0.9373$$

We know that,

$$h_2 = h_{f2} + X_2 h_{fg2}$$

$$= 439.3 + (0.9373)2244.1$$

$$h_2 = 2542 \text{ KJ/Kg}$$

$$\begin{aligned} \text{Exit velocity } (V_2) &= \sqrt{2000(2943 - 2542) + 50^2} \\ &= \sqrt{804456.6} \\ &= 896.91 \text{ m/s.} \end{aligned}$$

2. Dry saturated steam at 6.5 bar with negligible velocity expands isentropically in a convergent divergent nozzle to 1.4 bar and dryness fraction 0.956. Determine the final velocity of steam from the nozzle if 13% heat is loss in friction. Find the % reduction in the final velocity.

Given data:

Initial pressure (P1) = 6.5 bar

Exit pressure (P2) = 1.4 bar

Dryness fraction (X2) = 0.956

Heat loss = 13%

To Find:

The percent reduction in final velocity

Solution:

From steam table for initial pressure P1 = 6.5bar, take

$$\text{values } h_1 = h_f = 2758.8 \text{ KJ/Kg}$$

Similarly, at 1.4 bar,

$$h_{fg2} = 2231.9 \text{ KJ/Kg}$$

$$h_{f2} = 458.4 \text{ KJ/Kg}$$

$$= h_{f2} + X_2 h_{fg2}$$

$$= 458.4 + (0.956)$$

$$2231.6 h_2 = 2592.1 \text{ KJ/Kg}$$

$$\text{Final velocity (V2)} = \sqrt{2000(h_1 - h_2)}$$

$$= \sqrt{2000(2758.8 - 2592.1)}$$

$$V_2 = 577.39 \text{ m/s}$$

Here heat drop is 13% = 0.13

Nozzle efficiency (η) = 1 - 0.13 = 0.87

Velocity of steam by considering the nozzle

$$\text{efficiency, } V_2 = \sqrt{2000(h_1 - h_2) \times \eta}$$

$$V_2 = \sqrt{2000(2758.8 - 2592.1) \times 0.87}$$

$$V_2 = 538.55 \text{ m/s}$$

$$\begin{aligned} \text{\% reduction in final velocity} &= \frac{577.39 - 538.55}{577.39} \times 100 \% \\ &= 6.72\% \end{aligned}$$

3. A convergent divergent nozzle receives steam at 7bar and 200°C and it expands isentropically into a space of 3bar neglecting the inlet velocity calculate the exit area required for a mass flow of 0.1Kg/sec . when the flow is in equilibrium through all and super saturated with $PV^{1.3} = C$.

Given Data:

$$\text{Initial pressure (P}_1\text{)} = 7\text{bar} = 7 \times 10^5 \text{ N/m}^2$$

$$\text{Initial temperature (T}_1\text{)} = 200^\circ\text{C}$$

$$\text{Pressure (P}_2\text{)} = 3\text{bar} = 3 \times 10^5 \text{ N/m}^2$$

$$\text{Mass flow rate (m)} = 0.1 \text{ Kg/sec}$$

$$PV^{1.3} = C$$

To Find:

Area of the nozzle at exit

Solution:

From steam table for $P_1 = 7\text{bar}$ and $T_1 = 200^\circ\text{C}$

$$V_1 = 0.2999$$

$$h_1 = 2844.2$$

$$S_1 = 6.886$$

Similarly for $P_2 = 3\text{bar}$

$$V_{f2} = 0.001074 \quad V_{g2} = 0.60553$$

$$h_{f2} = 561.5 \quad h_{fg2} = 2163.2$$

$$S_{f2} = 1.672 \quad S_{fg2} = 5.319$$

We know that, $S_1 = S_2 = S_t$

$$S_1 = S_{f2} + X_2 S_{fg2}$$

$$6.886 = 1.672 + X_2 (5.319)$$

$$X_2 = 0.98$$

Similarly,

$$h_2 = h_{f2} + X_2 h_{fg2}$$

$$h_2 = 561.5 + 0.98 (2163.2)$$

$$h_2 = 2681.99$$

(i) Flow is in equilibrium through all:

$$V_2 = \sqrt{2000 (h_1 - h_2)}$$

$$V_2 = \sqrt{2000 (2844.2 - 2681.99)}$$

$$V_2 = 569.56$$

$$v_2 = X_2 \times v_{g2}$$

$$= 0.98 \times 0.60553 = 0.5934$$

$$m = \frac{[(A)_2 \times V_2]}{v_2}$$

$$A_2 = \frac{[m \times V_2]}{v_2} = \frac{0.5934 \times 0.1}{569.56}$$

$$A_2 = 1.041 \times 10^{-4} m^2$$

(ii) For saturated flow:

$$v_2 = \sqrt{\frac{2n}{n-1} (P_1 v_1) \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}$$

$$v_2 = \sqrt{\frac{2(1.3)}{1.3-1} (7 \times 10^5 \times 0.2999) \left(1 - \frac{3 \times 10^5}{7 \times 10^5}\right)^{\frac{1.3-1}{1.3}}}$$

$$V_2 = 568.69 \text{ m/s}$$

specific volume of steam at exit. For super saturated flow,

$$P_1 v_1^n = P_2 v_2^n$$

$$\left(\frac{v_2}{v_1}\right)^n = \frac{P_1}{P_2}$$

$$v_2 = \left(\frac{7}{3}\right)^{\frac{1}{1.3}} \times 0.2999$$

$$v_2 = 0.5754$$

$$A_2 = \frac{(m \times V_2)}{v_2}$$

$$= \frac{0.1 \times 0.5754}{568.69}$$

$$A_2 = 1.011 \times 10^{-4} \text{ m}^2$$

3.12. TWO MARKS UNIVERSITY QUESTIONS:**Part-A (2 Marks)**

1. What are the various types of nozzles and their functions?
2. Define nozzle efficiency and critical pressure ratio.
3. Explain the phenomenon of super saturated expansion in steam nozzle. Or what is metastable flow?
4. State the function of fixed blades.
5. Classify steam turbines.
6. How does impulse work?
7. What is meant by carry over loss?
8. State the function of moving blades...."
9. What is the fundamental difference between the operation of impulse and reaction steam turbines?
10. What are the different methods of governing steam turbines?
11. How is throttle governing done?
12. Where nozzle control governing is used?
13. Whereby - pass governing is more suitable?
14. What are the different losses in steam turbines?

2.13. UNIVERSITY ESSAY QUESTIONS:

PART- B (16Marks)

1. An impulse turbine having a set of 16 nozzles receives steam at 20 bar, 400°C . The pressure of steam at exit is 12 bar. If the total discharge is 260 Kg/min and nozzle efficiency is 90%. Find the cross sectional area of each nozzle, if the steam has velocity of 80m/s at entry to the nozzle, find the percentage increase in discharge. (16)
2. Dry saturated steam at a pressure of 8 bar enters the convergent divergent nozzle and leaves it at a pressure 1.5 bar. If the flow is isentropic and if the corresponding index of expansion is 1.133, find the ratio of 0.3 are at exit and throat for max. discharge. (16)
3. Steam enters a group of nozzles of a steam turbine at 12 bar and 2200°C and leaves at 1.2 bar. The steam turbine develops 220 Kw with a specific steam consumption of 13.5 Kg/ Kw. Hr. If the diameter of nozzle at throat is 7mm. Calculate the number of nozzle (16)
4. Derive an expression for critical pressure ratio in terms of the index of expansion (16)
5. Explain the method of governing in steam turbine. (16)
6. Explain various type of compounding in Turbine (16)
7. A 50% reaction turbine running at 400 rpm has the exit angle of blades as 20° and the velocity of steam relative to the blade at the exit is 1.35 times mean speed of the blade. The steam flow rate is 8.33 kg/s and at a particular stage the specific volume is $1.38\text{m}^3/\text{kg}$. Calculate, suitable blade height, assuming the rotor mean diameter 12 times the blade height, and diagram work. (16)
8. The blade angle of a single ring of an impulse turbine is 300m/s and the nozzle angle is 200 . The isentropic heat drop is 473kJ/kg and nozzle efficiency is 85%. Given the blade velocity coefficient is 0.7 and the blades are symmetrical, Draw the velocity diagram and

calculate for a mass flow of 1 kg/s i) axial thrust on balding ii) steam consumption per BP hour if the mechanical efficiency is 90% iii) blade efficiency and stage efficiency. (16)

FRANCIS XAVIER ENGINEERING COLLEGE

ME1251 THERMAL ENGINEERING

UNIT IV

AIR COMPRESSORS

CONTENTS

TECHNICAL TERMS

4.1 Classification of compressors

4.2 Positive Displacement compressors

4.2.1 Double acting compressor

4.2.2 Diaphragm Compressors

4.3 Rotary compressors

4.3.1 Lobe compressor

4.3.2 Liquid ring compressor

4.3.3 Vane Type compressor:

4.3.4 Screw Type compressor

4.3.5 Scroll Type Compressor

4.4 Non-Positive displacement compressors

4.4.1 Centrifugal Compressor

4.4.2 Axial Compressor

4.4.3 Roots Blower Compressor

4.5 Multistage Compression

4.5.1 Advantages of Multi-stage compression

4.6 Work done in a single stage reciprocating compressor without clearance volume

4.6.1 Work done in a single stage reciprocating compressor with clearance volume

4.7 Volumetric Efficiency

4.7.1 Mathematical analysis of multistage compressor is done with following assumptions

4.8 Solved Problems

4.9 Two Marks University Questions

4.10 University Essay Questions

TECHNICAL TERMS

1. Volumetric Efficiency of the Compressor

It is the ratio of actual volume of air drawn in the compressor to the stroke volume of the compressor.

2. Mechanical efficiency

It is the ratio of indicated power to shaft power or brake power of motor.

3. Isentropic efficiency

It is the ratio of the isentropic power to the brake power required to drive the compressor.

4. Centrifugal compressor

The flow of air is perpendicular to the axis of compressor

5. Axial flow compressor

The flow of air is parallel to the axis of compressor

6. Compression:

The process of increasing the pressure of air, gas and vapour by reducing its volume is called as compression.

7. Single acting compressor:

The suction, compression and the delivery of air takes on the one side of piston

8. Double acting compressor:

The suction, compression and the delivery of air takes place on both sides of the piston.

9. Multi stage compressor:

The compression of air from initial pressure to the final pressure is carried out in more than one cylinder.

10. Application of compressed air:

Pneumatic brakes, drills, jacks, lifts, spray of paintings, shop cleaning, injecting the fuel in diesel engine, supercharging, refrigeration and in air conditioning systems.

11. Inter cooler:

It is a simple heat exchanger, exchanges the heat of compressed air from low pressure compressor to circulating water before the air enters to high pressure compressor. The purpose of intercooling is to minimize the work of compression.

12. Isentropic efficiency:

It is the ratio of isentropic power to the brake power required to drive the compressor.

13. Clearance ratio:

It is the ratio of clearance volume to the swept volume or stroke volume is called as clearance ratio.

14. Isothermal efficiency:

It is the ratio between isothermal work to the actual work of the compressor.

15. Compression ratio:

The ratio between total volume and the clearance volume of the cylinder is called compression ratio.

16. Perfect intercooling:

When the temperature of the air leaving the intercooler is equal to the original atmospheric air temperature, then the inter cooling is called perfect intercooling.

UNIT-IV

AIR COMPRESSORS

4.1 Classification of compressors:

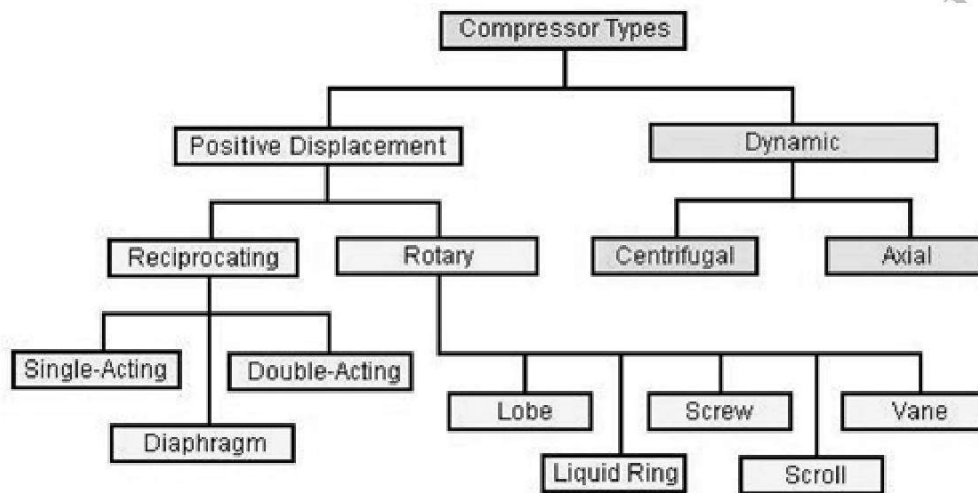


Fig 1.

The compressors are also classified based on other aspects like

1. Number of stages (single-stage, 2-stage and multi-stage),
2. Cooling method and medium (Air cooled, water cooled and oil-cooled),
3. Drive types (Engine driven, Motor driven, Turbine driven, Belt, chain, gear or direct coupling drives),
4. Lubrication method (Splash lubricated or forced lubrication or oil-free compressors).
5. Service Pressure (Low, Medium, High)

4.2 Positive Displacement compressors: Reciprocating Compressor: Single-Acting

Reciprocating compressor:

These are usually reciprocating compressors, which has piston working on air only in one direction. The other end of the piston is often free or open which does not perform any work. The

air is compressed only on the top part of the piston. The bottom of the piston is open to crankcase and not utilized for the compression of air.

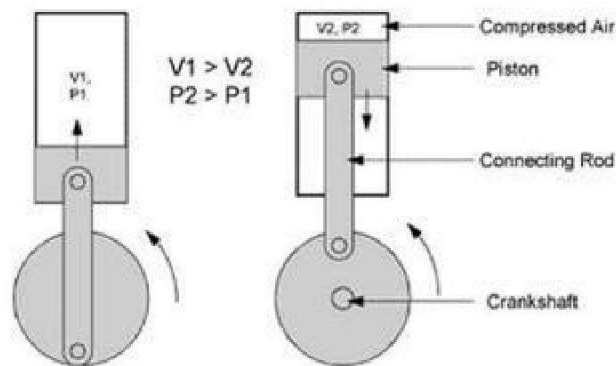


Fig 2 Piston & Cylinder Arrangement

4.2.1 Double acting compressor:

These compressors are having two sets of suction/intake and delivery valves on both sides of the piston. As the piston moves up and down, both sides of the piston is utilized in compressing the air. The intake and delivery valves operate corresponding to the stroke of the compressor. The compressed air delivery is comparatively continuous when compared to a single-acting air compressor. Thus both sides of the pistons are effectively used in compressing the air.

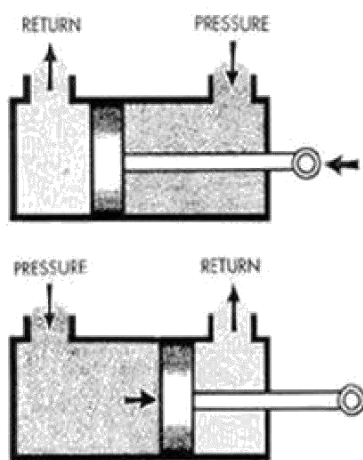


Fig 3

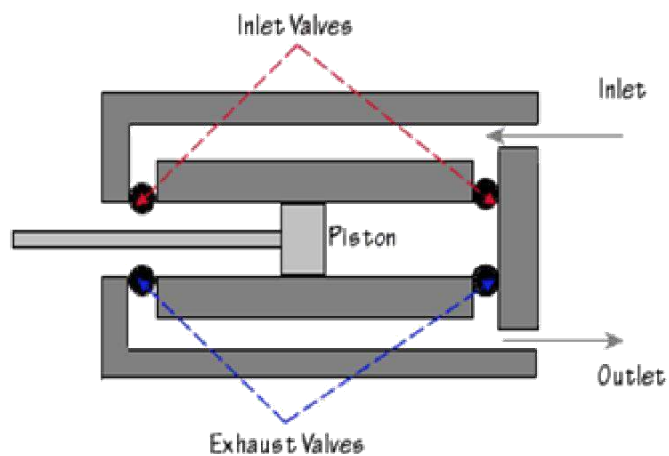


Fig 4 Double Acting Compressor

4.2.2 Diaphragm Compressors: In the diaphragm compressor, the piston pushes against a diaphragm, so the air does not come in contact with the reciprocating parts. This type compressor is preferred for food preparation, pharmaceutical, and chemical industries, because no effluent from the compressor enters the fluid.

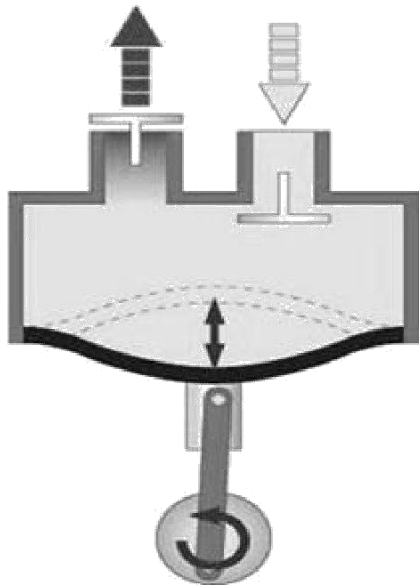


Fig 5 Diaphragm Compressors

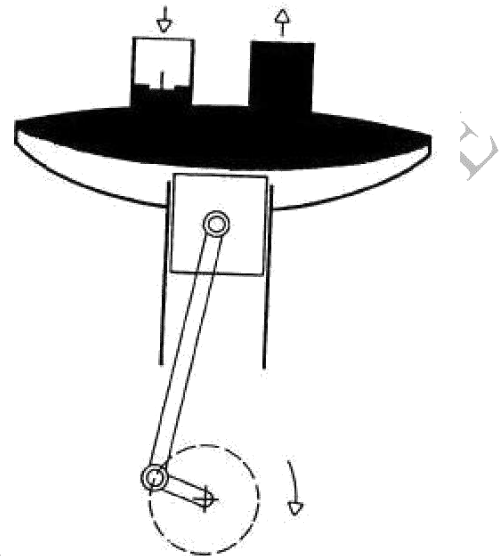


Fig 6 Diaphragm Compressors

4.3 Rotary compressors:

4.3.1 Lobe compressor:

The Lobe type air compressor is very simpler type with no complicated moving parts. There are single or twin lobes attached to the drive shaft driven by the prime mover. The lobes are displaced by 90 degrees. Thus if one of the lobes is in horizontal position, the other at that particular instant will be in vertical position. Thus the air gets trapped in between these lobes and as they rotate they get compressed and delivered to the delivery line.

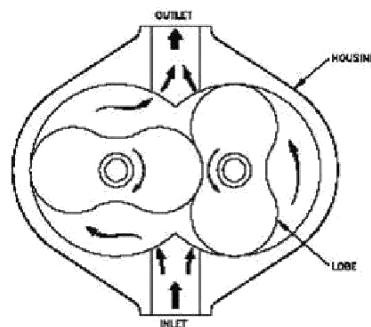


Fig 7 Lobe compressor

4.3.2 Liquid ring compressor:

Liquid ring compressors require a liquid to create a seal. For medical applications, liquid ring compressors are always sealed with water but not oil. An impeller, which is offset so the impeller is not in the center of the pump housing, rotates and traps pockets of air in the space between the impeller fins and the compressor housing. The impeller is typically made of brass. As the impeller turns, there is a pocket of air that is trapped in the space between each of the fins. The trapped air is compressed between the impeller and the pump housing, sealed with the water ring. As the air is compressed, it's then pushed out of the pumps discharge. To avoid possible contaminants the compressor is always getting a supply of fresh sealing water. In a “once through” system, sealing water is drained and used only once, while in a “partial re-circulating” system, some (but never all) of the discharged water is re-circulated.

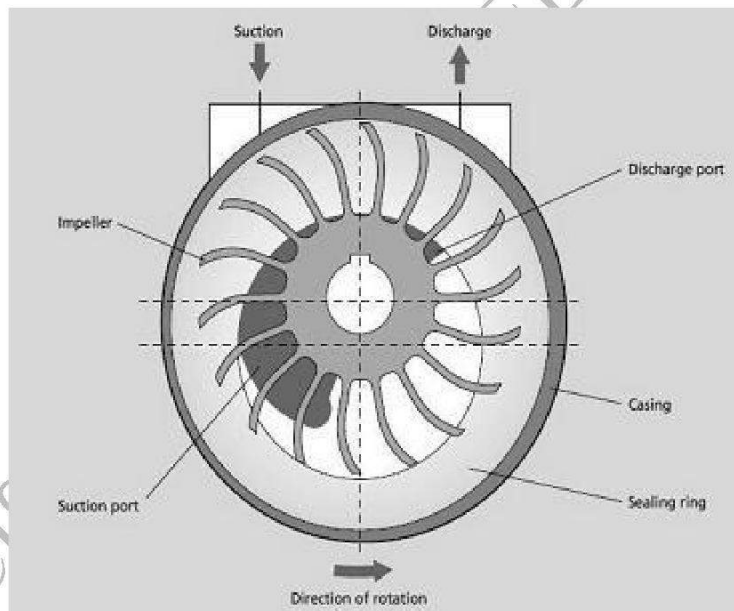


Fig 8 Liquid ring compressor

4.3.3 Vane Type compressor:

The rotary slide vane-type, as illustrated in Figure, has longitudinal vanes, sliding radially in a slotted rotor mounted eccentrically in a cylinder. The centrifugal force carries the sliding vanes against the cylindrical case with the vanes forming a number of individual longitudinal cells in

the eccentric annulus between the case and rotor. The suction port is located where the longitudinal cells are largest. The size of each cell is reduced by the eccentricity of the rotor as the vanes approach the discharge port, thus compressing the air. This type of compressor, looks and functions like a vane type hydraulic pump. An eccentrically mounted rotor turns in a cylindrical housing having an inlet and outlet. Vanes slide back and forth in grooves in the rotor. Air pressure or spring force keeps the tip of these vanes in contact with the housing. Air is trapped in the compartments formed by the vanes and housing and is compressed as the rotor turns.

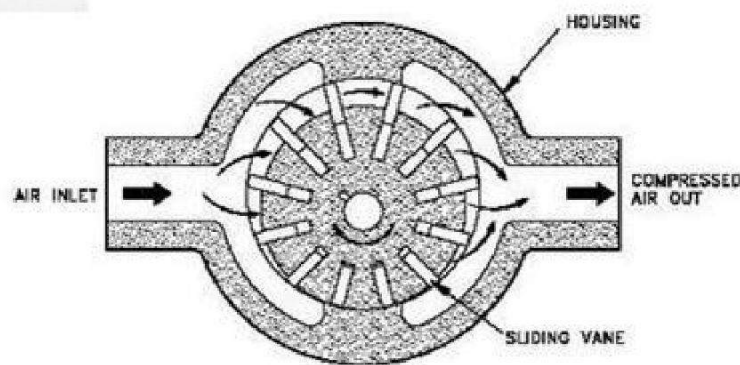


Fig 9 Vane Type compressor

4.3.4 Screw Type compressor:

The screw compressors are efficient in low air pressure requirements. Two screws rotate intermeshing with each other, thus trapping air between the screws and the compressor casing, forming pockets which progressively travel and gets squeezed and delivering it at a higher pressure which opens the delivery valve. The compressed air delivery is continuous and quiet in operation than a reciprocating compressor. Rotary air compressors are positive displacement compressors. The most common rotary air compressor is the single stage helical or spiral lobe oil flooded screw air compressor. These compressors consist of two rotors within a casing where the rotors compress the air internally. There are no valves. These units are basically oil cooled (with air cooled or water cooled oil coolers) where the oil seals the internal clearances. Since the cooling takes place right inside the compressor, the working parts never experience extreme

operating temperatures. The rotary compressor, therefore, is a continuous duty, air cooled or water cooled compressor package.

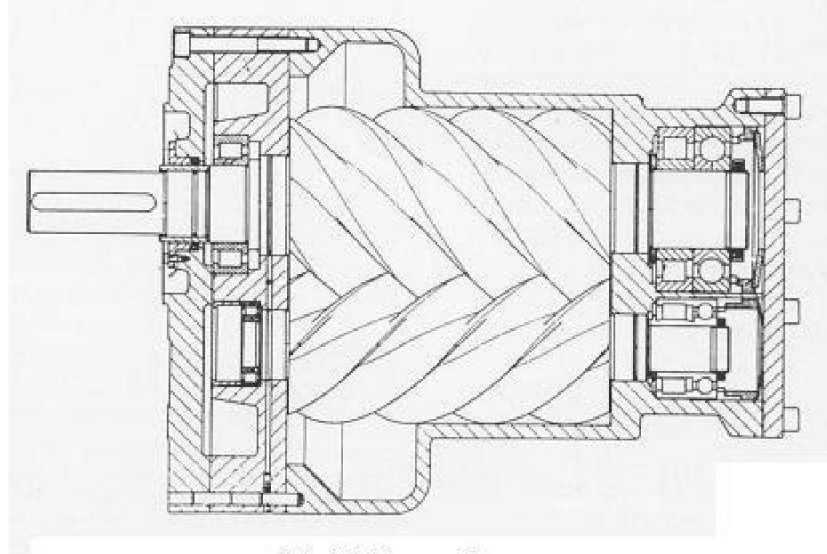


Fig 10 Screw Type compressor

4.3.5 Scroll Type Compressor:

This type of compressor has a very unique design. There are two scrolls that look like loosely rolled up pieces of paper—one rolled inside the other. The orbiting scroll rotates inside of the stationary scroll. The air is forced into progressively smaller chambers towards the center. The compressed air is then discharged through the center of the fixed scroll. No inlet or exhaust valves are needed.

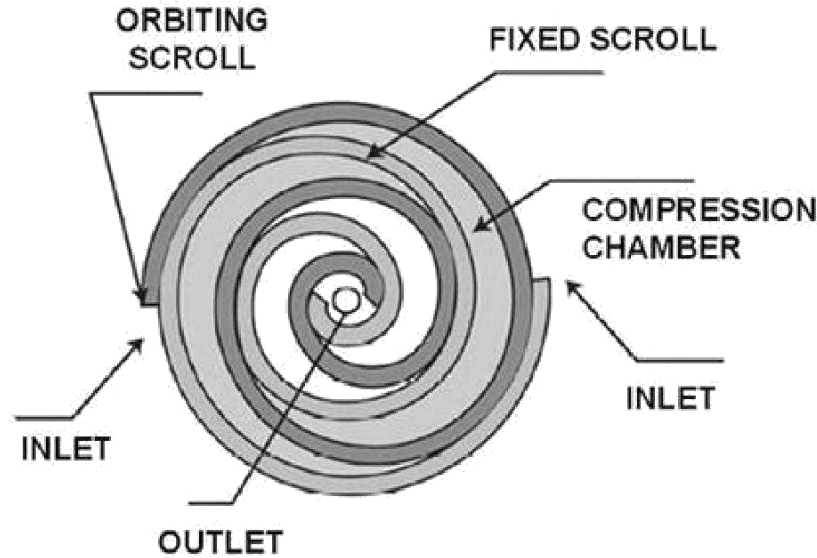


Fig 11 Scroll Type Compressor

4.4 Non-Positive displacement compressors or Dynamic compressor:

4.4.1 Centrifugal Compressor:

The centrifugal air compressor is a **dynamic** compressor which depends on transfer of energy from a **rotating impeller** to the air. Centrifugal compressors produce high-pressure discharge by converting angular momentum imparted by the rotating impeller (dynamic displacement). In order to do this efficiently, centrifugal compressors rotate at higher speeds than the other types of compressors. These types of compressors are also designed for higher capacity because flow through the compressor is continuous. Adjusting the inlet guide vanes is the most common method to control capacity of a centrifugal compressor. By closing the guide vanes, volumetric flows and capacity are reduced.

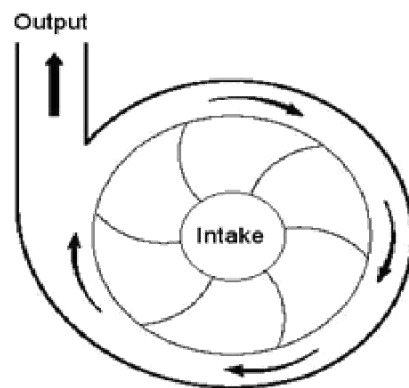


Fig 12 Centrifugal Compressor

The centrifugal air compressor is an oil free compressor by design. The oil lubricated running gear is separated from the air by shaft seals and atmospheric vents. The centrifugal air compressor is a dynamic compressor which depends on a rotating impeller to compress the air. In order to do this efficiently, centrifugal compressors must rotate at higher speeds than the other types of compressors. These types of compressors are designed for higher capacity because flow through the compressor is continuous and oil free by design.

4.4.2 Axial Compressor:

These are similar to centrifugal compressors except the direction of air flow is axial. The blades of the compressor are mounted onto the hub and in turn onto the shaft. As the shaft rotates at a high speed, the ambient air is sucked into the compressor and then gets compressed (high speed of rotation of the blades impart energy to the air) and directed axially for further usage. An axial flow compressor, in its very simple form is called as axial flow fan, which is commonly used for domestic purposes. The pressure built depends on the number of stages. These are commonly used as vent fans in enclosed spaces, blower ducts, etc. One can find its main application in the aerospace industry, where the gas turbines drive the axial flow air compressors.

4.4.3 Roots Blower Compressor:

This type is generally called as blower. The discharge air pressure obtained from this type of machine is very low. The Discharge Pressure of 1 bar can be obtained in Single Stage and pressure of 2.2 bar is obtained from Stage. The discharge pressure achieved by two rotors which

have separate parallel axis and rotate in opposite directions. This is the example of Positive Displacement Compressor in Rotary Type Air Compressor.

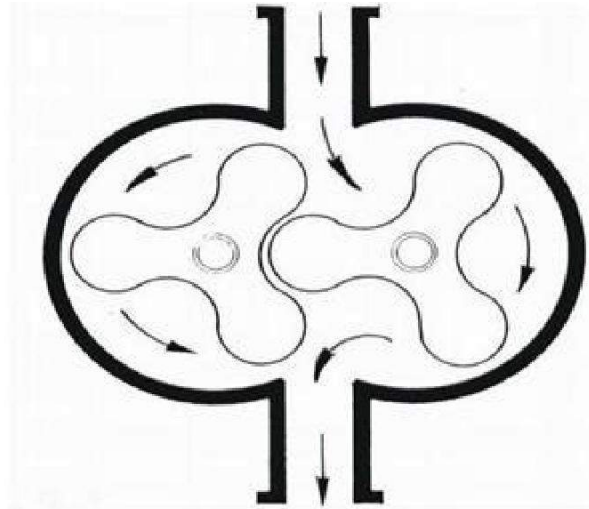


Fig 13 Roots Blower Compressor

4.5 Multistage Compression:

Multistage compression refers to the compression process completed in more than one stage i.e., a part of compression occurs in one cylinder and subsequently compressed air is sent to subsequent cylinders for further compression. In case it is desired to increase the compression ratio of compressor then multi-stage compression becomes inevitable. If we look at the expression for volumetric efficiency then it shows that the volumetric efficiency decreases with increase in pressure ratio. This aspect can also be explained using p-V representation shown in Figure.

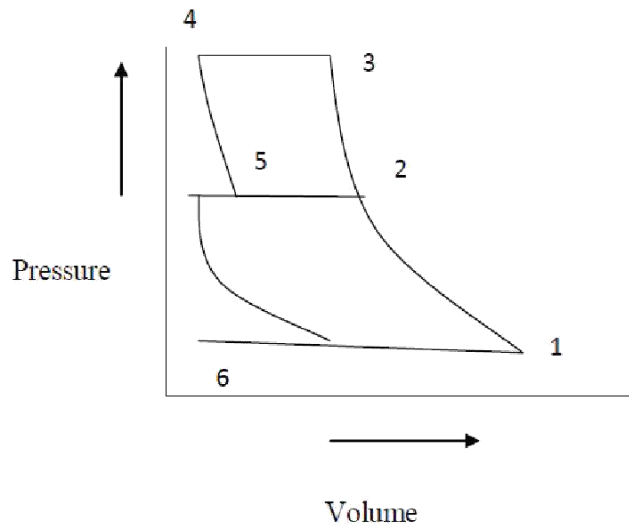


Fig 14 P-V Diagram

A multi-stage compressor is one in which there are several cylinders of different diameters. The intake of air in the first stage gets compressed and then it is passed over a cooler to achieve a temperature very close to ambient air. This cooled air is passed to the intermediate stage where it is again getting compressed and heated. This air is again passed over a cooler to achieve a temperature as close to ambient as possible. Then this compressed air is passed to the final or the third stage of the air compressor where it is compressed to the required pressure and delivered to the air receiver after cooling sufficiently in an after-cooler.

4.5.1 Advantages of Multi-stage compression:

1. The work done in compressing the air is reduced, thus power can be saved
2. Prevents mechanical problems as the air temperature is controlled
3. The suction and delivery valves remain in cleaner condition as the temperature and vaporization of lubricating oil is less
4. The machine is smaller and better balanced
5. Effects from moisture can be handled better, by draining at each stage
6. Compression approaches near isothermal
7. Compression ratio at each stage is lower when compared to a single-stage machine

4.6 Work done in a single stage reciprocating compressor without clearance volume:

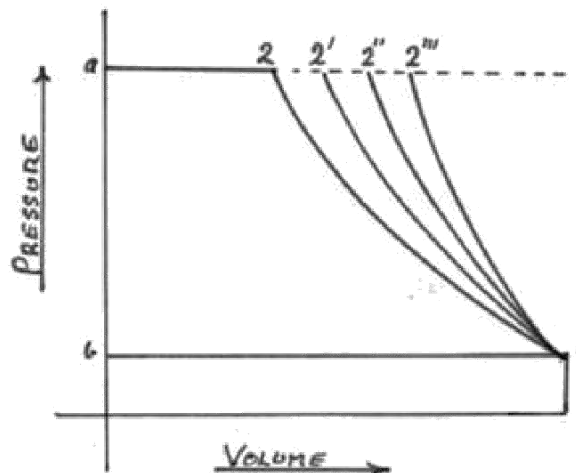


Fig 15 PV Diagram

Air enters compressor at pressure p_1 and is compressed upto p_2 . Compression work requirement can be estimated from the area below the each compression process. Area on p-V diagram shows that work requirement shall be minimum with isothermal process 1-2'''. Work requirement is maximum with process 1-2 ie., adiabatic process. As a designer one shall be interested in a compressor having minimum compression work requirement. Therefore, ideally compression should occur isothermally for minimum work input. In practice it is not possible to have isothermal compression because constancy of temperature during compression can not be realized. Generally, compressors run at substantially high speed while isothermal compression requires compressor to run at very slow speed so that heat evolved during compression is dissipated out and temperature remains constant. Actually due to high speed running of compressor the compression process may be assumed to be near adiabatic or polytrophic process

following law of compression as $Pv^n = C$ with of „n“ varying between 1.25 to 1.35 for air.

Compression process following three processes is also shown on T-s diagram.

It is thus obvious that actual compression process should be compared with isothermal compression process. A mathematical parameter called isothermal efficiency is defined for quantifying the degree of deviation of actual compression process from ideal compression process. Isothermal efficiency is defined by the ratio is isothermal work and actual indicated work

in reciprocating compressor.

$$\text{Isothermal efficiency} = \frac{\text{Isothermal work}}{\text{Actual indicated Work}}$$

Practically, compression process is attempted to be closed to isothermal process by air/water cooling, spraying cold water during compression process. In case of multistage compression process the compression in different stages is accompanied by intercooling in between the stages. $P_2 V_2$.

Mathematically, for the compression work following polytropic process, $PV^n=C$. Assuming negligible clearance volume the cycle work done.

$W_c = \text{Area on p-V diagram}$

$$\begin{aligned} W_c &= \left[p_2 V_2 + \left(\frac{p_2 V_2 - p_1 V_1}{n-1} \right) \right] - p_1 V_1 \\ &= \left[\left(\frac{n}{n-1} \right) [p_2 V_2 - p_1 V_1] \right] \\ &= \left(\frac{n}{n-1} \right) (p_1 V_1) \left[\frac{p_2 V_2}{p_1 V_1} - 1 \right] \\ &= \left(\frac{n}{n-1} \right) (p_1 V_1) \left[\left(\frac{p_2}{p_1} \right)^{\left(\frac{n-1}{n} \right)} - 1 \right] \\ &= \left(\frac{n}{n-1} \right) (mRT_1) \left[\left(\frac{p_2}{p_1} \right)^{\left(\frac{n-1}{n} \right)} - 1 \right] \\ &= \left(\frac{n}{n-1} \right) (mR)(T_2 - T_1) \end{aligned}$$

In case of compressor having isothermal compression process, $n = 1$, i.e., $p_1 V_1 = p_2 V_2$

$$W_{iso} = p_2 V_2 + p_1 V_1 \ln r - p_1 V_1$$

$$W_{iso} = p_1 V_1 \ln r, \quad \text{where, } r = \frac{V_1}{V_2}$$

In case of compressor having adiabatic compression process,

$$W_{adiabatic} = \left(\frac{\gamma}{\gamma - 1} \right) (mR)(T_2 - T_1) \quad (\text{Or})$$

$$W_{adiabatic} = (mC_p)(T_2 - T_1)$$

$$W_{adiabatic} = (m)(h_2 - h_1)$$

$$\eta_{iso} = \frac{p_1 V_1 \ln r}{\left(\frac{n}{n-1} \right) (p_1 V_1) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]}$$

The isothermal efficiency of a compressor should be close to 100% which means that actual compression should occur following a process close to isothermal process. For this the mechanism be derived to maintain constant temperature during compression process. Different arrangements which can be used are:

(i) Faster heat dissipation from inside of compressor to outside by use of fins over cylinder. Fins facilitate quick heat transfer from air being compressed to atmosphere so that temperature rise during compression can be minimized.

(ii) Water jacket may be provided around compressor cylinder so that heat can be picked by cooling water circulating through water jacket. Cooling water circulation around compressor regulates rise in temperature to great extent.

(iii) The water may also be injected at the end of compression process in order to cool the air being compressed. This water injection near the end of compression process requires special arrangement in compressor and also the air gets mixed with water and needs to be separated out

before being used. Water injection also contaminates the lubricant film inner surface of cylinder and may initiate corrosion etc, the water injection is not popularly used.

(iv) In case of multistage compression in different compressors operating serially, the air leaving one compressor may be cooled up to ambient state or somewhat high temperature before being injected into subsequent compressor. This cooling of fluid being compressed between two consecutive compressors is called inter cooling and is frequently used in case of multistage compressors.

4.6.1 Work done in a single stage reciprocating compressor with clearance volume:

Considering clearance volume: With clearance volume the cycle is represented on Figure. The work done for compression of air polytropically can be given by the area enclosed in cycle 1-2-3-4. Clearance volume in compressors varies from 1.5% to 35% depending upon type of compressor.

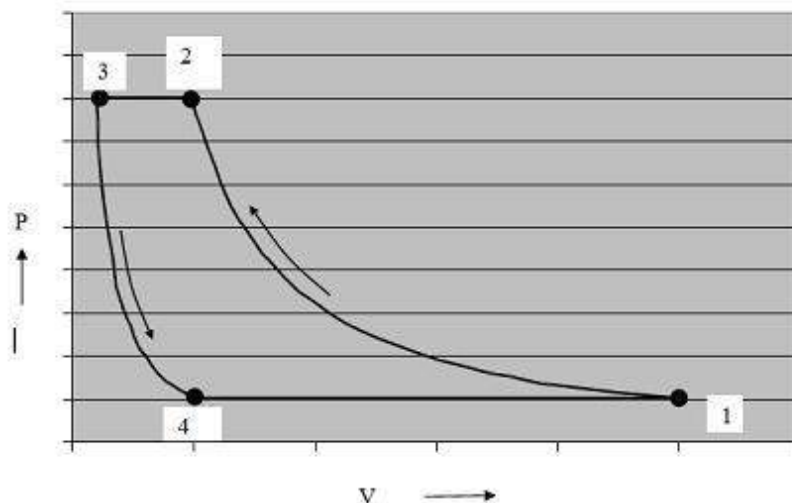


Fig16 PV Diagram

$$W_{c,with CV} = \text{Area 1234}$$

$$= \left(\frac{n}{n-1} \right) (p_1 V_1) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] - \left(\frac{n}{n-1} \right) (p_4 V_4) \left[\left(\frac{p_3}{p_4} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$\text{Here } P_1 = P_4, P_2 = P_3$$

$$= \left(\frac{n}{n-1} \right) (p_1 V_1) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] - \left(\frac{n}{n-1} \right) (p_1 V_4) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \left(\frac{n}{n-1} \right) (p_1) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] (V_1 - V_4)$$

In the cylinder of reciprocating compressor (V1-V4) shall be the actual volume of air delivered per cycle. $V_d = V_1 - V_4$. This (V1 – V4) is actually the volume of air inhaled in the cycle and

delivered subsequently.

$$W_{c,with CV} = \left(\frac{n}{n-1} \right) (p_1 V_d) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

If air is considered to behave as perfect gas then pressure, temperature, volume and mass can be inter related using perfect gas equation. The mass at state 1 may be given as m_1 mass at state 2 shall be m_1 , but at state 3 after delivery mass reduces to m_2 and at state 4 it shall be m_2 .

$$\text{So, at state 1, } p_1 V_1 = m_1 R T_1$$

$$\text{at state 2, } p_2 V_2 = m_1 R T_2$$

$$\text{at state 3, } p_3 V_3 = m_2 R T_3 \text{ or } p_2 V_3 = m_2 R T_3$$

$$\text{at state 4, } p_4 V_4 = m_2 R T_4 \text{ or } p_1 V_4 = m_2 R T_4$$

Ideally there shall be no change in temperature during suction and delivery
i.e., $T_4 = T_1$ and $T_2 = T_3$ from earlier equation

$$W_{c,withCV} = \left(\frac{n}{n-1} \right) (p_1) \left[\left(\frac{p_2}{p_1} \right)^{\frac{(n-1)}{n}} - 1 \right] (V_1 - V_4)$$

Temperature and pressure can be related as,

$$\left(\frac{p_2}{p_1} \right)^{\frac{(n-1)}{n}} = \frac{T_2}{T_1} \quad \text{and} \quad \left(\frac{p_4}{p_3} \right)^{\frac{(n-1)}{n}} = \frac{T_4}{T_3} \quad \Rightarrow \quad \left(\frac{p_1}{p_2} \right)^{\frac{(n-1)}{n}} = \frac{T_4}{T_3}$$

Substituting

$$W_{c,withCV} = \left(\frac{n}{n-1} \right) (m_1 R T_1 - m_2 R T_4) \left[\frac{T_2}{T_1} - 1 \right]$$

Substituting for constancy of temperature during suction and deliver

$$W_{c,withCV} = \left(\frac{n}{n-1} \right) (m_1 R T_1 - m_2 R T_1) \left[\frac{T_2 - T_1}{T_1} \right]$$

Or

$$W_{c,withCV} = \left(\frac{n}{n-1} \right) (m_1 - m_2) R (T_2 - T_1)$$

Thus (m₁-m₂) denotes the mass of air sucked or delivered. For unit mass of air delivered the work done per kg of air can be given as,

$$W_{c,withCV} = \left(\frac{n}{n-1} \right) R (T_2 - T_1) \quad \text{per kg of air}$$

Thus from above expressions it is obvious that the clearance volume reduces the effective swept volume i.e., the mass of air handled but the work done per kg of air delivered remains unaffected. From the cycle work estimated as above the theoretical power required for running compressor shall be,

For single acting compressor running with N rpm, power input required, assuming clearance volume.

$$Power_{required} = \left[\left(\frac{n}{n-1} \right) \left[\left(\frac{p_2}{p_1} \right)^{\left(\frac{n-1}{n} \right)} - 1 \right] p_1 (V_1 - V_4) \right] (N)$$

For double acting compressor, Power

$$Power_{required} = \left[\left(\frac{n}{n-1} \right) \left[\left(\frac{p_2}{p_1} \right)^{\left(\frac{n-1}{n} \right)} - 1 \right] p_1 (V_1 - V_4) \right] (2N)$$

4.7 Volumetric Efficiency:

Volumetric efficiency of compressor is the measure of the deviation from volume handling capacity of compressor. Mathematically, the volumetric efficiency is given by the ratio of actual volume of air sucked and swept volume of cylinder. Ideally the volume of air sucked should be equal to the swept volume of cylinder, but it is not so in actual case. Practically the volumetric efficiency lies between 60 to 90%. Volumetric efficiency can be overall volumetric efficiency and absolute volumetric efficiency as given below.

$$\text{Overall volumetric efficiency} = \frac{\text{Volume of free air sucked in cylinder}}{\text{Swept volume of LP cylinder}}$$

$$(\text{Volumetric efficiency})_{\text{freeaircondition}} = \frac{\text{Volume of free air sucked in cylinder}}{(\text{Swept volume of LP cylinder})_{\text{freeaircondition}}}$$

Here free air condition refers to the standard conditions. Free air condition may be taken as 1 atm or 1.01325 bar and 15°C or 288K. consideration for free air is necessary as otherwise the different compressors can not be compared using volumetric efficiency because specific volume or density of air varies with altitude. It may be seen that a compressor at datum level (sea level) shall deliver large mass than the same compressor at high altitude.

This concept is used for giving the capacity of compressor in terms of „free air delivery“ (FAD).

“Free air delivery is the volume of air delivered being reduced to free air conditions”. In case of air the free air delivery can be obtained using perfect gas equation as,

$$\frac{p_a V_a}{T_a} = \frac{p_1 (V_1 - V_4)}{T_1} = \frac{p_2 (V_2 - V_3)}{T_2}$$

Where subscript a or pa, Va, Ta denote properties at free air conditions

$$V_a = \frac{p_1 T_a}{p_a} \frac{p_1 (V_1 - V_4)}{T_1} = \text{FAD per cycle}$$

This volume Va gives „free air delivered“ per cycle by the compressor. Absolute volumetric efficiency can be defined, using NTP conditions in place of free air conditions,

$$\eta_{vol} = \frac{\text{FAD}}{\text{Swept volume}} = \frac{V_a}{(V_1 - V_2)} = \frac{p_1 T_a (V_1 - V_4)}{p_a T_1 (V_1 - V_3)}$$

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1} \right) \left[\frac{(V_s + V_c) - V_4}{V_s} \right]$$

Here Vs is the swept volume = V1 – V3 and Vc is the clearance volume = V3

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1} \right) \left[1 + \left(\frac{V_c}{V_s} \right) - \left(\frac{V_4}{V_s} \right) \right]$$

$$\text{Here } \frac{V_4}{V_s} = \frac{V_4}{V_c} \cdot \frac{V_c}{V_s} = \left(\frac{V_4}{V_3} \cdot \frac{V_c}{V_s} \right)$$

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1} \right) \left[1 + C - C \left(\frac{V_4}{V_3} \right) \right]$$

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1} \right) \left[1 + C - C \left(\frac{p_2}{p_1} \right)^{\frac{1}{n}} \right]$$

Volumetric efficiency depends on ambient pressure and temperature, suction pressure and temperature, ratio of clearance to swept volume, and pressure limits. Volumetric efficiency increases with decrease in pressure ratio in compressor.

4.7.1 Mathematical analysis of multistage compressor is done with following assumptions:

- (i) Compression in all the stages is done following same index of compression and there is no pressure drop in suction and delivery pressures in each stage. Suction and delivery pressure remains constant in the stages.
- (ii) There is perfect inter cooling between compression stages.
- (iii) Mass handled in different stages is same i.e., mass of air in LP and HP stages are same.
- (iv) Air behaves as perfect gas during compression.

From combined p-V diagram the compressor work requirement can be given as,

$$\text{Work requirement in LP cylinder, } W_{LP} = \left(\frac{n}{n-1} \right) P_1 V_1 \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right\}$$

$$\text{Work requirement in HP cylinder, } W_{HP} = \left(\frac{n}{n-1} \right) P_2 V_2 \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right\}$$

For perfect intercooling, $p_1 V_1 = p_2 V_2$ and

$$W_{HP} = \left(\frac{n}{n-1} \right) P_2 V_2 \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right\}$$

Therefore, total work requirement, $W_c = W_{LP} + W_{HP}$, for perfect inter cooling

$$W_c = \left(\frac{n}{n-1} \right) \left[P_1 V_1 \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right\} + P_2 V_2 \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right\} \right]$$

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

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

Minimum work required in two stage compressor:

Minimum work required in two stage compressor can be given by

$$W_{c,min} = \left(\frac{n}{n-1} \right) P_1 V_1 \cdot 2 \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right\}$$

For i number of stages, minimum work,

$$W_{c,min} = i \cdot \left(\frac{n}{n-1} \right) P_1 V_1 \left\{ \left(\frac{P_{i+1}}{P_i} \right)^{\frac{n-1}{n}} - 1 \right\}$$

It also shows that for optimum pressure ratio the work required in different stages remains same for the assumptions made for present analysis. Due to pressure ratio being equal in all stages the temperature ratios and maximum temperature in each stage remains same for perfect inter cooling. If the actual volume sucked during suction stroke is V_1 , V_2 , and V_3 . . . For different stages they by perfect gas law, $P_1 V_1 = RT_1$, $P_2 V_2 = RT_2$, $P_3 V_3 = RT_3$ For perfect inter cooling $P_1 V_1 = RT_1$, $P_2 V_2 = RT_1$, $P_3 V_3 = RT_1$ $P_1 V_1 = P_2 V_2 = RT_2$, $P_3 V_3 =$

If the volumetric efficiency of respective stages in $\eta_{V_1}, \eta_{V_2}, \eta_{V_3}, \dots$

Then theoretical volume of cylinder1, $V_{1,th} = \frac{V_1}{\eta_{V1}}; V_1 = \eta_{V1} \cdot V_{1,th}$

Cylinder 2, $V_{2,th} = \frac{V_2}{\eta_{V2}}; V_2 = \eta_{V2} \cdot V_{2,th}$

Cylinder 3, $V_{3,th} = \frac{V_3}{\eta_{V3}}; V_3 = \eta_{V3} \cdot V_{3,th}$

Substituting,

$$P_1 \cdot \eta_{V1} \cdot V_{1,th} = P_2 \cdot \eta_{V2} \cdot V_{2,th} = P_3 \cdot \eta_{V3} \cdot V_{3,th} = \dots$$

Theoretical volumes of cylinder can be given using geometrical dimensions of cylinder as diameters $D_1, D_2, D_3 \dots$ and stroke lengths $L_1, L_2, L_3 \dots$

$$\text{Or } V_{1,th} = \frac{\pi}{4} \cdot D_1^2 \cdot L_1$$

$$V_{2,th} = \frac{\pi}{4} \cdot D_2^2 \cdot L_2$$

$$V_{3,th} = \frac{\pi}{4} \cdot D_3^2 \cdot L_3$$

$$\text{Or } P_1 \cdot \eta_{V1} \cdot \frac{\pi}{4} \cdot D_1^2 \cdot L_1 = P_2 \cdot \eta_{V2} \cdot \frac{\pi}{4} \cdot D_2^2 \cdot L_2$$

$$= P_3 \cdot \eta_{V3} \cdot \frac{\pi}{4} \cdot D_3^2 \cdot L_3 = \dots$$

$$P_1 \cdot \eta_{V1} \cdot \frac{\pi}{4} \cdot D_1^2 \cdot L_1 = P_2 \cdot \eta_{V2} \cdot \frac{\pi}{4} \cdot D_2^2 \cdot L_2$$

$$= P_3 \cdot \eta_{V3} \cdot \frac{\pi}{4} \cdot D_3^2 \cdot L_3 = \dots$$

If the volumetric efficiency is same for all cylinders, i.e. $\eta_{V1} = \eta_{V2} = \eta_{V3} = \dots$ and stroke for all cylinder is same i.e. $L_1 = L_2 = L_3 = \dots$

Then, $D_1^3 P_1 = D_2^3 P_2 = D_3^3 P_3 = \dots$

These generic relations may be used for getting the ratio of diameters of cylinders of multistage compression.

Energy balance: Energy balance may be applied on the different components constituting multistage compression.

For LP stage the steady flow energy equation can be written as below:

$$m \cdot h_1 + W_{LP} = m \cdot h_2 + Q_{LP}$$

$$Q_{LP} = W_{LP} - m(h_2 - h_1)$$

$$Q_{LP} = W_{LP} - m \cdot C_p (T_2 - T_1)$$

For intercooling between LP and HP stage steady flow energy equation shall be;

$$m \cdot h_2 = m \cdot h_{2'} + Q_{int}$$

$$Q_{int} = m(h_2 - h_{2'})$$

$$Q_{int} = m \cdot C_p (T_2 - T_{2'})$$

For HP stage the steady flow energy equation yields.

$$m \cdot h_{2'} + W_{HP} = m \cdot h_3 + Q_{HP}$$

$$Q_{HP} = W_{HP} + m(h_{2'} - h_3)$$

$$Q_{HP} = W_{HP} + m \cdot C_p (T_{2'} - T_3) = W_{HP} - m \cdot C_p (T_3 - T_{2'})$$

In case of perfect intercooling and optimum pressure ratio, $T_{2'} = T_1$ and $T_3 = T_2$.

Hence for these conditions,

$$Q_{LP} = W_{LP} - m \cdot C_p (T_2 - T_1)$$

$$Q_{int} = m \cdot C_p (T_2 - T_1)$$

$$Q_{HP} = W_{HP} - m \cdot C_p (T_2 - T_1)$$

Total heat rejected during compression shall be the sum of heat rejected during compression and heat extracted in intercooler for perfect inter cooling. Heat rejected during compression for polytropic process

$$= \left(\frac{\gamma - n}{\gamma - 1} \right) \times \text{Work}$$

4.8 Solved Problems

1. A single stage double acting air compressor of 150KW power takes air in at 1 bar & delivers at 6 bar. The compression follows the law $PV^{1.35} = C$. the compressor runs at 160rpm with average piston speed of 150 m/min. Determine the size of the cylinder.

GIVEN DATA

Power (P) = 150KW

Piston speed (2LN) = 150m/min $= \frac{150}{60} = 2.5 \frac{\text{m}}{\text{s}}$

Speed (N) = 160rpm $160/60 = 2.7\text{rps}$

Pressure (P1) = 1bar = 100KN/m²

Pressure (P2) = 6bar = 600KN/m²

$PV^{1.35} = C$, $n = 1.35$

Hence it is a polytropic process.

TO FIND

Size of the cylinder (d)?

SOLUTION

It is given that,

$$2lN = 2.5\text{m/s}$$

$$l = \frac{2.5}{2 \times 2.7}$$

$$l = 0.4629\text{m}$$

$$\text{since } V_1 = V_s = \frac{\pi d^2 l}{4}$$

$$V_1 = V_s = \frac{\pi d^2 (1.4629)}{4}$$

$$V_1 = 0.3635d^2$$

We know that,

$$\text{Power (P)} = 2 \times W \times N$$

(for double acting)

For polytropic process, work done (W) is

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

$$W = \frac{1.35}{1.35-1} (100 \times 0.3635d^2) \left[(6)^{\frac{1.35-1}{1.35}} - 1 \right]$$

$$W = 82.899 d^2$$

$$\text{Power (P)} = 2 \times W \times N$$

$$150 = 2 \times 82.899 d^2 \times 2.7$$

$$d^2 = 0.3350$$

$$d = 0.57M$$

2. A single stage single acting reciprocating air compressor is required to handle 30m^3 of free air per hour measured at 1 bar . the delivery pressure is 6.5 bar and the speed is 450 r.p.m allowing volumetric efficiency of 75%;an isothermal efficiency of 76% and mechanical efficiency of 80% Find the indicated mean effective pressure and the power required the compressor

GIVEN DATA

Volume	$V_1 = 30\text{m}^3$
Pressure	$P_1 = 1 \text{ bar} , P_2 = 6.5 \text{ bar}$
Speed	$N = 450 \text{ r.p.m}$
Volumetric efficiency	$\eta_v = 75\%$
Isothermal efficiency	$\eta_i = 76\%$
Mechanical efficiency	$\eta_m = 80\%$

TO FIND

The indicated mean effective pressure

The power required to drive the compressor

SOLUTION

Indicted Mean Effective Pressure

We know that isothermal work done

$$= 2.3 V_1 P_1 \log \left[\left(\frac{p_2}{p_1} \right) \right]$$

$$= 2.3 \times 10^5 \times 30 \log \left[\left(\frac{6.5}{1} \right) \right]$$

$$= 5609 \times 10^3 \text{ J/h}$$

And indicated work done = $\frac{\text{Isothermal work done}}{\text{Isothermal efficiency}}$

$$= \frac{5609}{0.76} = 7380 \text{ KJ/h}$$

We know that swept volume of the piston

$$V_s = \frac{\text{volume of free air}}{\text{volumetric efficiency}} = \frac{30}{.75}$$

$$= 40 \text{ m}^3/\text{h}$$

Indicated mean effective pressure $p_m = \frac{\text{indicated work done}}{\text{swept volume}} = \frac{7380}{40}$

$$= 184.5 \text{ kJ/m}^3$$

$$= 184.5 \text{ KN/m}^2$$

The power required to drive the compressor

We know that work done by the compressor = $\frac{\text{indicated work done}}{\text{mechanical efficiency}}$

$$= \frac{7380}{.8}$$

$$= 9225 \text{ KJ/h}$$

Therefore the power required to drive the compressor = $\frac{9225}{3600}$

$$=2.56\text{KW}$$

RESULT

Indicated mean effective pressure $p_m = 184.5 \text{ kN/m}^2$

The power required to drive the compressor $= 2.56 \text{ KW}$

3. A two stages, single acting air compressor compresses air to 20bar. The air enters the L.P cylinder at 1bar and 27°C and leaves it at 4.7bar. The air enters the H.P. cylinder at 4.5bar and 27°C . the size of the L.P cylinder is 400mm diameter and 500mm stroke. The clearance volume in both cylinder is 4% of the respective stroke volume. The compressor runs at 200rpm, taking index of compression and expansion in the two cylinders as 1.3, estimate 1. The indicated power required to run the compressor; and 2. The heat rejected in the intercooler per minute.

GIVEN DATA

Pressure (P_4) = 20bar

Pressure (P_1) = 1bar $= 1 \times 10^5 \text{ N/m}^2$

Temperature (T_1) = $27^\circ\text{C} = 27 + 273 = 300\text{K}$

Pressure (P_2) = 4.7bar

Pressure (P_3) = 4.5bar

Temperature (T_3) = $27^\circ\text{C} = 27 + 273 = 300\text{K}$

Diameter (D_1) = 400mm 0.4m

Stroke (L_1) = 500mm = 0.5m

$$K = \frac{v_{c1}}{v_{s1}} = \frac{v_{c3}}{v_{s3}} = 4\% = 0.04$$

$$N = 200\text{rpm} ; n = 1.3$$

TO FIND

Indicated power required to run the compressor

SOLUTION

We know the swept volume of the L.P cylinder

$$\begin{aligned} v_{s1} &= \frac{\pi}{4} (D_1)^2 L_1 = \frac{\pi}{4} (0.4)^2 0.5 \\ &= 0.06284 \text{ m}^3 \end{aligned}$$

And volumetric efficiency,

$$\begin{aligned} \eta_v &= 1 + K - K \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \\ &= 1 + 0.04 - 0.04 \left(\frac{4.7}{1} \right)^{\frac{1}{1.3}} \\ &= 0.9085 \text{ or } 90.85\% \end{aligned}$$

Volume of air sucked by air pressure compressor,

$$\begin{aligned} v_1 &= v_{s1} \times \eta_v = 0.06284 \times 0.9085 = 0.0571 \frac{\text{m}^3}{\text{stroke}} \\ &= 0.0571 \times N_w = 0.0571 \times 200 = 11.42 \text{ m}^3/\text{min} \end{aligned}$$

And volume of air sucked by H.P compressor,

$$v_3 = \frac{P_1 V_1}{P_3} = \frac{1 \times 11.42}{4.5} = 2.54 \frac{m^3}{min}$$

We know that indicated work done by L.P compressor,

$$\begin{aligned} W_L &= \left(\frac{n}{n-1} \right) P_1 v_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \\ &= \left(\frac{1.3}{1.3-1} \right) 1 \times 10^5 \times 11.42 \left[\left(\frac{4.7}{1} \right)^{\frac{1.3-1}{1.3}} - 1 \right] \\ &= 2123.3 \times 10^3 \text{ J/min} = 2123.3 \text{ KJ/min} \end{aligned}$$

And indicated workdone by H.P compressor,

$$\begin{aligned} W_H &= \left(\frac{n}{n-1} \right) P_3 v_3 \left[\left(\frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 1 \right] \\ &= \left(\frac{1.3}{1.3-1} \right) 4.5 \times 10^5 \times 2.54 \left[\left(\frac{4.20}{4.5} \right)^{\frac{1.3-1}{1.3}} - 1 \right] \\ &= 2043.5 \times 10^3 \text{ J/min} = 2034.5 \text{ KJ/min} \end{aligned}$$

Total indicated work done by the compressor,

$$W = W_L + W_H = 2123.3 + 2034.5 = 4157.8 \text{ KJ/min}$$

Indicated power required to run the compressor

$$= 4157.8 / 60 = 69.3 \text{ KW}$$

4.9 TWO MARK UNIVERSITY QUESTIONS:

Part-A (2 Marks)

1. What is meant by single acting compressor?
2. What is meant by double acting compressor?
3. What is meant by single stage compressor?
4. What is meant by multistage compressor?
5. Define isentropic efficiency
6. Define mean effective pressure. How is it related to in power of an I.C engine.
7. What is meant by free air delivered?
8. Explain how flow of air is controlled in a reciprocating compressor?
9. What factors limit the delivery pressure in reciprocating compressor?
10. Name the methods adopted for increasing isothermal efficiency of reciprocating air compressor.
11. Why clearance is necessary and what is its effect on the performance of reciprocating compressor?
12. What is compression ratio?
13. What is meant by inter cooler?

4.10 UNIVERSITY ESSAY QUESTIONS:

Part-B (16 Marks)

1. Drive an expression for the work done by single stage single acting reciprocating air compressor. (16)
2. Drive an expression for the volumetric efficiency of reciprocating air compressors (16)
3. Explain the construction and working of a root blower (16)
4. Explain the construction and working of a centrifugal compressor (16)
5. Explain the construction and working of a sliding vane compressor and axial flow compressor.(16)
6. A single stage single acting air compressor is used to compress air from 1 bar and 22°C to 6 bar according to the law $PV^{1.25} = C$. The compressor runs at 125 rpm and the ratio of stroke length to bore of a cylinder is 1.5. If the power required by the compressor is 20 kW, determine the size of the cylinder. (16)
7. A single stage single acting air compressor is used to compress air from 1.013 bar and 25°C to 7 bar according to law $PV^{1.3} = C$.The bore and stroke of a cylinder are 120mm and 150mm respectively. The compressor runs at 250 rpm .If clearance volume of the cylinder is 5% of stroke volume and the mechanical efficiency of the compressor is 85%, determine volumetric efficiency, power, and mass of air delivered per minute. (16)

8. A two stage single acting air compressor compresses 2m³ air from 1 bar and 20° C to 15 bar. The air from the low pressure compressor is cooled to 25° C in the intercooler. Calculate the minimum power required to run the compressor if the compression follows $PV^{1.25}=C$ and the compressor runs at 400 rpm. (16)

FRANCIS XAVIER ENGINEERING COLLEGE

ME1251 THERMAL ENGINEERING

**UNIT V
REFRIGERATION AND AIR CONDITIONING**

CONTENTS

TECHNICAL TERMS

5.1 Fundamentals of refrigeration

5.2 Common Refrigerants

5.3 Required Properties of Ideal Refrigerant

5.4 Coefficient of Performance (COP)

5.5 Vapour Compression Refrigeration

5.5.1 Schematic of a Basic Vapor Compression Refrigeration System

5.5.2 Alternative Refrigerants for Vapour Compression Systems

5.6 Vapour Absorption Refrigeration

5.7 Comparison between Vapor Compression and Absorption System

5.8 Ton of refrigeration

5.9 Air- Conditioning System

5.9.1. Zoned Systems

5.9.2 Unitary Systems

5.10 Window Air-conditioning System

5.10.1 Blower

5.10.2 Propeller fan or the condenser fan

5.11.3 Fan motor

5.11 Split Air-conditioning System

5.11.1 Evaporator Coil or the Cooling Coil

5.11.2 Air Filter

5.11.3 Cooling Fan or Blower

5.11.4 Drain Pipe

5.11.5 Louvers or Fins

5.12 Solved Problems

5.13 Two Marks University Questions

5.14 University Essay Questions

TECHNICAL TERMS

1. SPECIFIC HEAT

It is the ratio between the quantities of heat required to change the temperature of 1 pound of any substance 1°F, as compared to the quantity of heat required to change 1 pound of water 1°F. Specific heat is equal to the number of Btu required to raise the temperature of 1 pound of a substance 1°F. For example, the specific heat of milk is .92, which means that 92 Btu will be needed to raise 100 pounds of milk 1° F. The specific heat of water is 1, by adoption as a standard, and specific heat of another substance (solid, liquid, or gas) is determined experimentally by comparing it to water. Specific heat also expresses the heat-holding capacity of a substance compared to that of water.

2. SENSIBLE HEAT

Heat that is added to, or subtracted from, a substance that changes its temperature but not its physical state is called sensible heat. It is the heat that can be indicated on a thermometer. This is the heat human senses also can react to, at least within certain ranges. For example, if a person put their finger into a cup of water, the senses readily tell that person whether it is cold, cool, tepid, hot, or very hot. Sensible heat is applied to a solid, a liquid, or a gas/vapor as indicated on a thermometer. The term sensible heat does not apply to the process of conversion from one physical state to another.

3. LATENT HEAT

It is the term used for the heat absorbed or given off by a substance while it is changing its physical state. When this occurs, the heat given off or absorbed does NOT cause a temperature change in the substance. In other words, sensible heat is the term for heat that affects the temperature of things; latent heat is the term for heat that affects the physical state of things.

To understand the concept of latent heat, you must realize that many substances may exist as solids, as liquids, or as gases, depending primarily upon the temperatures and pressure to which they are subjected.

4. SUPERHEAT

It is a very important term in the terminology of refrigeration - but it is unfortunately used in different ways. It can be used to describe a process where refrigerant vapour is heated from its

saturated condition to a condition at higher temperature. The term superheat can also be used to describe - or quantify - the end condition of the before-mentioned process.

5. PRESSURE

It is defined as a force per unit area. It is usually measured in pounds per square inch (psi). Pressure may be in one direction, several directions, or in all directions. The ice (solid) exerts pressure downward. The water (fluid) exerts pressure on all wetted surfaces of the container.

Gases exert pressure on all inside surfaces of their containers.

6. VAPORIZATION

It is the process of changing a liquid to vapor, either by evaporation or boiling. When a glass is filled with water, as shown in figure 6-10, and exposed to the rays of the sun for a day or two, you should note that the water level drops gradually. The loss of water is due to evaporation. Evaporation, in this case, takes place only at the surface of the liquid. It is gradual, but the evaporation of the water can be speeded up if additional heat is applied to it. In this case, the boiling of the water takes place throughout the interior of the liquid. Thus the absorption of heat by a liquid causes it to boil and evaporate.

7. CONDENSATION

It is the process of changing a vapor into a liquid. For example, in figure 6-12, a warm atmosphere gives up heat to a cold glass of water, causing moisture to condense out of the air and form on the outside surface of the glass. Thus the removal of heat from a vapor causes the vapor to condense.

8. COP of REFRIGERATION

The COP of a refrigeration system is the ratio of net refrigeration effect to the work required to produce the effect.

9. UNIT OF REFRIGERATION

The capacity of refrigeration is expressed in tonnes of refrigeration (TOR). 1 tones of refrigeration = 210 kJ/min (or) = 3.5 kJ/sec (kW)

A tone of refrigeration is defined as the quantity of heat to be removed in order to form one tone of ice at 0°C in 24 hours.

10. REFRIGERATION EFFECT

The amount of heat extracted in a given time is known as refrigeration effect.

11. EFFECTS OF UNDER COOLING

It increases the refrigeration effect therefore the COP increases. The mass flow rate of the refrigeration is less than that for the simple saturated cycle. The reduced mass flow rate reduces the piston displacement per minute. Power per tones of refrigeration losses due to reduction in mass flow rate. The increased efficiency may be offering some extent by the rise in the condenser pressure. Work input almost remains same. The heat rejection capacity of the condenser increases.

12. EFFECTS OF SUPER HEATING

Super heating increases the net refrigeration effect, but super heating requires more work input therefore super heating reduces the COP.

No moisture contents in the refrigerant therefore no corrosion in the machines part.

13. PROPERTIES OF IDEAL REFRIGERANT

It should have low boiling point and low freezing point.

It must have low specific heat and high latent heat.

It should have high thermal conductivity to reduce the heat transfer in evaporator and condenser.

It should have low specific volume to reduce the size of the compressor.

It should be non-flammable, non-expensive, non-toxic and non-corrosive.

It should have high critical pressure and temperature to avoid large power requirements.

It should give high COP to reduce the running cost of the system.

It must be cheap and must be readily available

14. RSHF

Room sensible heat factor is defined as the ratio of room sensible heat load to the room total heat load.

15. RELATIVE HUMIDITY

It is defined as the ratio of partial pressure of water vapour (p_w) in a mixture to the saturation pressure (p_s) of pure water at the same temperature of mixture.

16. SPECIFIC HUMIDITY

It is defined as the ratio of the mass of water vapour (m_s) in a given volume to the mass of dry air in a given volume (m_a).

17. DEGREE OF SATURATION

It is the ratio of the actual specific humidity and the saturated specific humidity at the same temperature of the mixture.

18. DEW POINT TEMPERATURE

The temperature at which the vapour starts condensing is called dew point temperature. It is also equal to the saturation temperature at the partial pressure of water vapour in the mixture. The dew point temperature is an indication of specific humidity.

19. SENSIBLE HEAT AND LATENT HEAT

Sensible heat is the heat that changes the temperature of the substance when added to it or when abstracted from it. Latent heat is the heat that does not affect the temperature but change of state occurred by adding the heat or by abstracting the heat.

20. PSYCHOMETRIC PROCESSES

1. Sensible heating and sensible cooling, 2. Cooling and dehumidification, 3. Heating and humidification, 4. Mixing of air streams, 5. Chemical dehumidification, 6. Adiabatic evaporative cooling.

21. ADIABATIC MIXING

The process of mixing two or more stream of air without any heat transfer to the surrounding is known as adiabatic mixing. It is happened in air conditioning system.

22. DRY BULB TEMPERATURE (DBT)

The temperature recorded by the thermometer with a dry bulb. The dry bulb thermometer cannot be affected by the moisture present in the air. It is the measure of sensible heat of the air.

23. WET BULB TEMPERATURE (WBT)

It is the temperature recorded by a thermometer whose bulb is covered with cotton wick (wet) saturated with water. The wet bulb temperature may be the measure of enthalpy of air. WBT is the lowest temperature recorded by moistened bulb.

24. DEW POINT DEPRESSION

It is the difference between dry bulb temperature and dew point temperature of air vapour mixture.

UNIT V

REFRIGERATION AND AIR CONDITIONING

5.1 Fundamentals of refrigeration

The first mechanical refrigerators for the production of ice appeared around the year 1860. In 1880 the first ammonia compressors and insulated cold stores were put into use in the USA. Electricity began to play a part at the beginning of this century and mechanical refrigeration plants became common in some fields: e.g. breweries, slaughter-houses, fishery, ice production, for example. After the Second World War the development of small hermetic refrigeration compressors evolved and refrigerators and freezers began to take their place in the home. Today, these appliances are regarded as normal household necessities.

Refrigeration is the process of removing heat from an area or a substance and is usually done by an artificial means of lowering the temperature, such as the use of ice or mechanical refrigeration.

Mechanical Refrigeration is defined as a mechanical system or apparatus so designed and constructed that, through its function, heat is transferred from one substance to another. Since refrigeration deals entirely with the removal or transfer of heat, some knowledge of the nature and effects of heat is necessary for a clear understanding of the subject.

5.2 Common Refrigerants

Today, there are three specific types of refrigerants used in refrigeration and air-conditioning systems:

1. Chlorofluorocarbons or CFCs, such as R-11, R-12, and R-114
2. Hydro chlorofluorocarbons or HCFCs, such as R-22 or R-123
3. Hydro fluorocarbons or HFCs, such as R-134a. All these refrigerants are "halogenated," which means they contain chlorine, fluorine, bromine, astatine, or iodine.

Refrigerants, such as Dichlorodifluoromethane (R-12), Mono chloro difluoromethane (R-22), and Refrigerant 502 (R-502), are called primary refrigerants because each one changes its state upon the application or absorption of heat, and, in this act of change, absorbs and extracts heat from the area or substance.

The primary refrigerant is so termed because it acts directly upon the area or substance, although it may be enclosed within a system. For a primary refrigerant to cool, it must be placed in a closed system in which it can be controlled by the pressure imposed upon it. The refrigerant can then absorb at the temperature ranges desired. If a primary refrigerant were used without being controlled, it would absorb heat from most perishables and freeze them solid.

Secondary Refrigerants are substances, such as air, water, or brine. Though hot refrigerants in themselves, they have been cooled by the primary refrigeration system; they pass over and around the areas and substances to be cooled; and they are returned with their heat load to the primary refrigeration system. Secondary refrigerants pay off where the cooling effect must be moved over a long distance and gastight lines cost too much.

Refrigerants are classified into groups. The National Refrigeration Safety Code catalogs all refrigerants into three groups:

- Group I – safest of the refrigerants, such as R-12, R-22, and R-502
- Group II – toxic and somewhat flammable, such as R-40 (Methyl chloride) and R-764 (Sulfur dioxide)
- Group III – flammable refrigerants, such as R-170 (Ethane) and R-290 (Propane).

R-12 Dichlorodifluoromethane (CCl₂F₂) Dichlorodifluoromethane, commonly referred to as R-12, is colorless and odorless in concentrations of less than 20 percent by volume in air. In higher concentrations, its odor resembles that of carbon tetrachloride. It is nontoxic, noncorrosive, nonflammable, and has a boiling point of -21.7°F (-29°C) at atmospheric pressure.

5.3 Required Properties of Ideal Refrigerant

- 1) The refrigerant should have low boiling point and low freezing point.

- 2) It must have low specific heat and high latent heat. Because high specific heat decreases the refrigerating effect per kg of refrigerant and high latent heat at low temperature increases the refrigerating effect per kg of refrigerant.
- 3) The pressures required to be maintained in the evaporator and condenser should be low enough to reduce the material cost and must be positive to avoid leakage of air into the system.
- 4) It must have high critical pressure and temperature to avoid large power requirements.
- 5) It should have low specific volume to reduce the size of the compressor.
- 6) It must have high thermal conductivity to reduce the area of heat transfer in evaporator and condenser.
- 7) It should be non-flammable, non-explosive, non-toxic and non-corrosive.
- 8) It should not have any bad effects on the stored material or food, when any leak develops in the system.
- 9) It must have high miscibility with lubricating oil and it should not have reacting properly with lubricating oil in the temperature range of the system.
- 10) It should give high COP in the working temperature range. This is necessary to reduce the running cost of the system.

5.4 Coefficient of Performance (COP)

The performance of refrigerators and heat pumps is expressed in terms of coefficient of performance (COP), defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

5.5 Vapour Compression Refrigeration

Heat flows naturally from a hot to a colder body. In refrigeration system the opposite must occur i.e. heat flows from a cold to a hotter body. This is achieved by using a substance called a refrigerant, which absorbs heat and hence boils or evaporates at a low pressure to form a gas. This gas is then compressed to a higher pressure, such that it transfers the heat it has gained

to ambient air or water and turns back (condenses) into a liquid. In this way heat is absorbed, or removed, from a low temperature source and transferred to a higher temperature source.

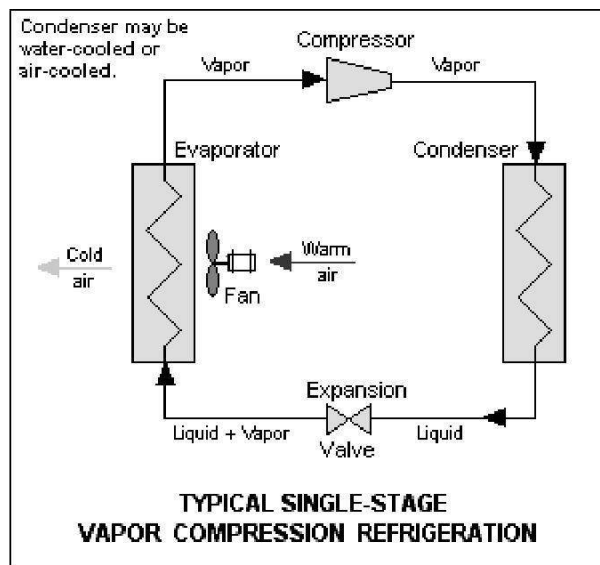


Fig 5.1 Vapour Compression Refrigeration

The refrigeration cycle can be broken down into the following stages

1 - 2 Low pressure liquid refrigerant in the evaporator absorbs heat from its surroundings, usually air, water or some other process liquid. During this process it changes its state from a liquid to a gas, and at the evaporator exit is slightly superheated.

2 - 3 The superheated vapour enters the compressor where its pressure is raised. There will also be a big increase in temperature, because a proportion of the energy input into the compression process is transferred to the refrigerant.

3 - 4 The high pressure superheated gas passes from the compressor into the condenser. The initial part of the cooling process (3 - 3a) desuperheats the gas before it is then turned back into liquid (3a - 3b). The cooling for this process is usually achieved by using air or water. A further reduction in temperature happens in the pipe work and liquid receiver (3b - 4), so that the refrigerant liquid is sub-cooled as it enters the expansion device.

4 - 1 The high-pressure sub-cooled liquid passes through the expansion device, which both reduces its pressure and controls the flow into the evaporator.

Refrigeration effect = $Q_R = m (h_1 - h_4)$

Since process 3-4 is a throttling process, $h_4 = h_3$.

Hence $Q_R = m (h_1 - h_3)$

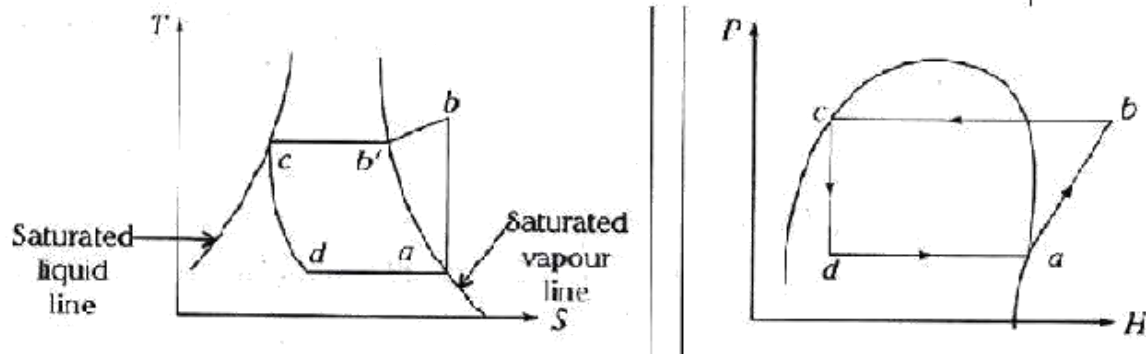
Similarly, by applying steady flow, steady state energy equation to compressor we get

Compressor work input = $W_c = m (h_2 - h_1)$

Hence

$$\text{COP} = \frac{Q_R}{W_c} = \frac{(h_1 - h_4)}{(h_2 - h_1)}$$

FRANCIS XAVIER ENGINEER



Sub cooling or Under cooling:

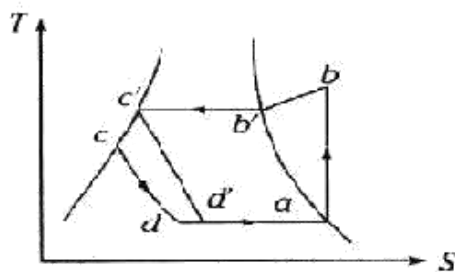
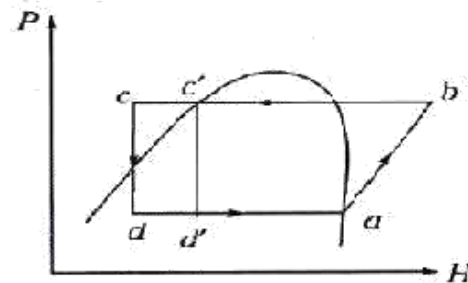


Fig 5.2 TS Diagram



PH Diagram

It can be seen that the condenser has to be capable of rejecting the combined heat inputs of the evaporator and the compressor; i.e. $(1 - 2) + (2 - 3)$ has to be the same as $(3 - 4)$. There is no heat loss or gain through the expansion device.

Super Heating:

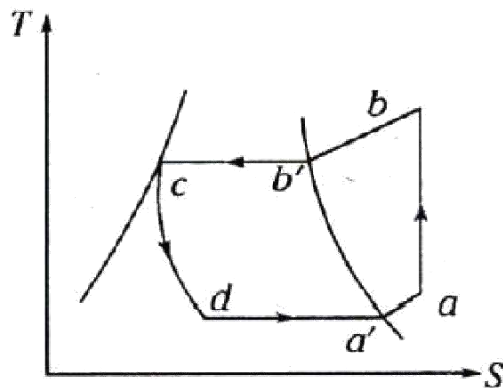


Fig 5.3 TS diagram

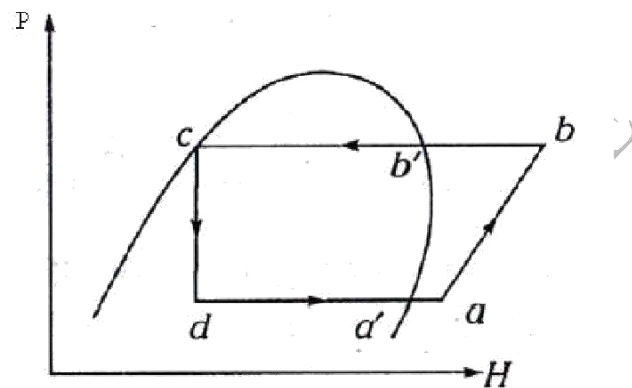


Fig 5.4 PH Diagram

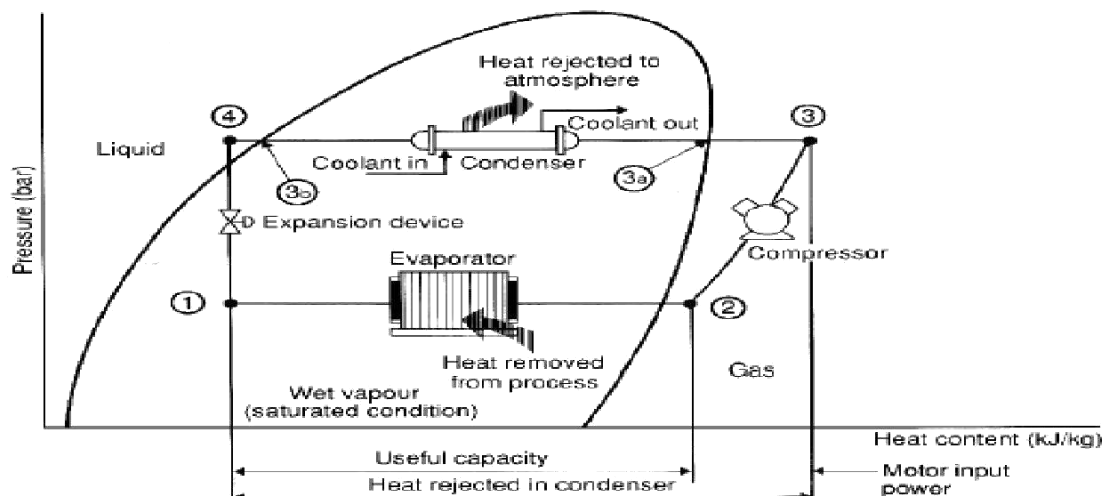


Fig. 5.5 condition of Refrigeration System

5.5.1 Schematic of a Basic Vapor Compression Refrigeration System

Advantages of Vapour compression refrigeration system over air refrigeration system:

- Since the working cycle approaches closer to carnot cycle, the C.O.P is quite high.
- Operational cost of vapour compression system is just above 1/4th of air refrigeration

system.

- Since the heat removed consists of the latent heat of vapour, the amount of liquid circulated is less and as a result the size of the evaporator is smaller.
- Any desired temperature of the evaporator can be achieved just by adjusting the throttle valve.

Disadvantages of Vapour compression refrigeration system over air refrigeration system

- Initial investment is high
- Prevention of leakage of refrigerant is a major problem

5.5.2 Alternative Refrigerants for Vapour Compression Systems

The use of CFCs is now beginning to be phased out due to their damaging impact on the protective tropospheric ozone layer around the earth. The Montreal Protocol of 1987 and the subsequent Copenhagen agreement of 1992 mandate a reduction in the production of ozone depleting Chlorinated Fluorocarbon (CFC) refrigerants in a phased manner, with an eventual stop to all production by the year 1996. In response, the refrigeration industry has developed two alternative refrigerants; one based on Hydrochloro Fluorocarbon (HCFC), and another based on Hydro Fluorocarbon (HFC). The HCFCs have a 2 to 10% ozone depleting potential as compared to CFCs and also, they have an atmospheric lifetime between 2 to 25 years as compared to 100 or more years for CFCs (Brandt, 1992). However, even HCFCs are mandated to be phased out by 2005, and only the chlorine free (zero ozone depletion) HFCs would be acceptable.

Until now, only one HFC based refrigerant, HFC 134a, has been developed. HCFCs are comparatively simpler to produce and the three refrigerants 22, 123, and 124 have been developed. The use of HFCs and HCFCs results in slightly lower efficiencies as compared to CFCs, but this may change with increasing efforts being made to replace CFCs.

5.6 Vapour Absorption Refrigeration

In the absorption refrigeration system, refrigeration effect is produced mainly by the use of energy as heat. In such a system, the refrigerant is usually dissolved in a liquid. A concentrated solution of ammonia is boiled in a vapour generator producing ammonia vapour at high pressure. The high pressure ammonia vapour is fed to a condenser where it is condensed to liquid ammonia by rejecting energy as heat to the surroundings. Then, the liquid ammonia is throttled through a valve to a low pressure. During throttling, ammonia is partially vapourized and its temperature decreases.

This low temperature ammonia is fed to an evaporator where it is vapourized removing energy from the evaporator. Then this low-pressure ammonia vapour is absorbed in the weak solution of ammonia. The resulting strong ammonia solution is pumped back to the vapour generator and the cycle is completed. The COP of the absorption system can be evaluated by considering it as a

combination of a heat pump and a heat engine

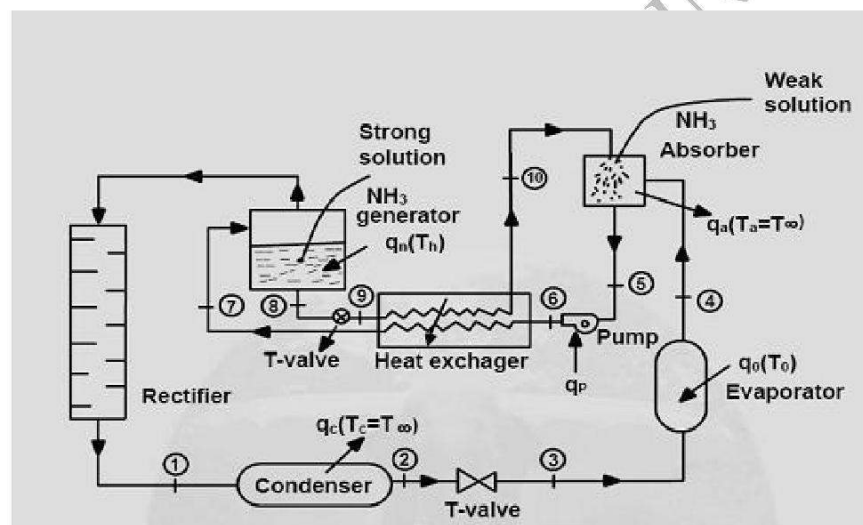


Fig 5.6 Vapour Absorption Refrigeration

5.7 Comparison between Vapor Compression and Absorption System

Table 1 Vapor Compression and Absorption System

Absorption system	Compression System
a) Uses low grade energy like heat. Therefore, may be worked on exhaust systems from I.C engines, etc.	a) Using high-grade energy like mechanical work.
b) Moving parts are only in the pump, which is a small element of the system. Hence operation is smooth.	b) Moving parts are in the compressor. Therefore, more wear, tear and noise.
c) The system can work on lower evaporator pressures also without affecting the COP.	c) The COP decreases considerably with decrease in evaporator pressure.
d) No effect of reducing the load on performance.	d) Performance is adversely affected at partial loads.
e) Liquid traces of refrigerant present in piping at the exit of evaporator	e) Liquid traces in suction line may damage the compressor.

5.8 Ton of refrigeration:

Amount of heat required to melt a Ton of Ice in a 24/h Period

One ton of refrigeration is the heat required to melt 1 ton of ice in 24 hrs. That is, a refrigeration machine rated at 1 ton cools as much in 24 hrs. as 1 ton of ice would by melting in the same period.

The heat required is the product of the latent heat of fusion and the mass in kg. $Q = mH$,

1 ton = 907 kg

Latent heat of fusion: $H = 340 \text{ kJ/kg}$

$Q = 907 \times 340 = 308380 \text{ kJ}$

The power required is then:

$$P = E/t = Q/t = 308380 \text{ kJ}/24 \text{ hr} = 308380/(24 \times 3600) = 3.57 \text{ kw}$$

Note: 1 watt = 1 J/s

So that 1 kw = 1 kJ/s

5.9 Air- Conditioning Systems:

The central air conditioning system is used for cooling big buildings, houses, offices, entire hotels, gyms, movie theaters, factories etc. If the whole building is to be air conditioned, HVAC engineers find that putting individual units in each of the rooms is very expensive initially as well in the long run. The central air conditioning system is comprised of a huge compressor that has the capacity to produce hundreds of tons of air conditioning. Cooling big halls, malls, huge spaces, galleries etc is usually only feasible with central conditioning units.

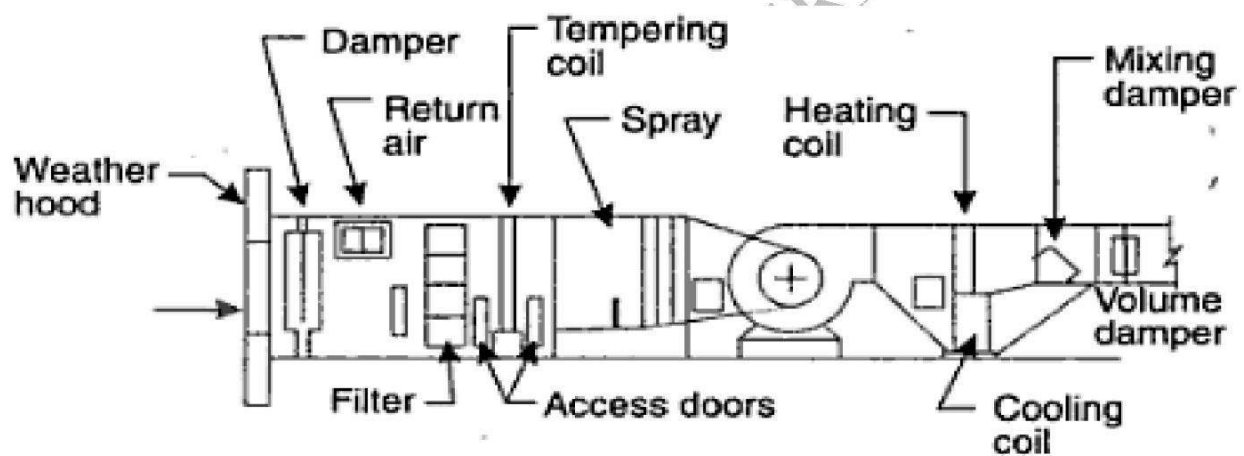


Fig.5.9 Air Condition System

5.9.1 Zoned Systems

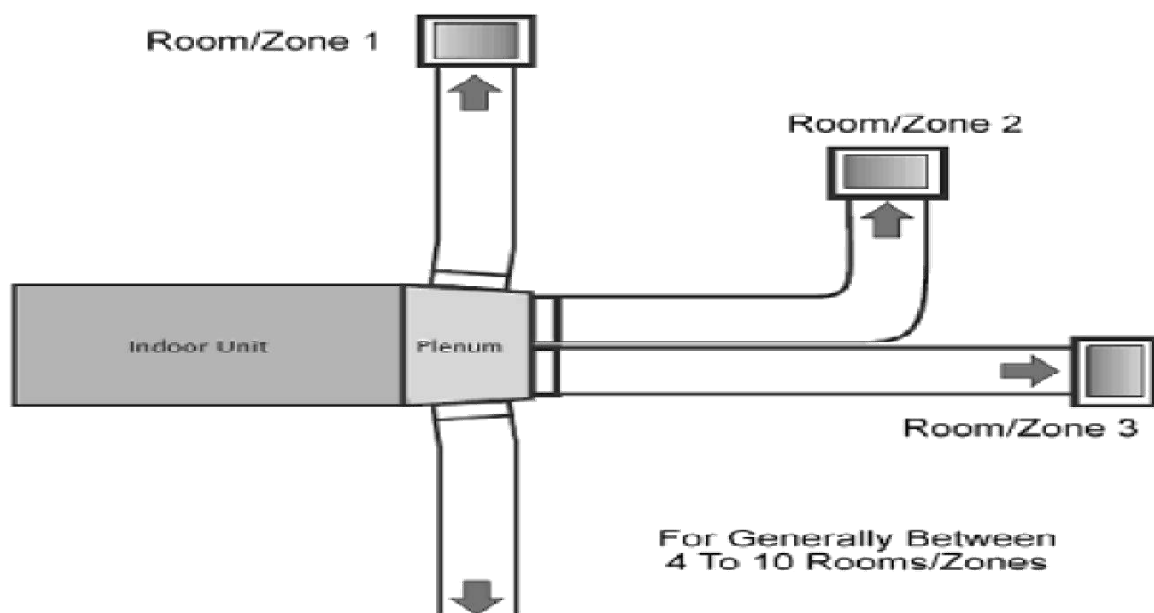


Fig 5.9.1 Zoned System

A zoned air conditioning system using a room air terminal which has the same horizontal dimensions as a floor tile of a raised tile floor such that the terminal may replace one tile in such a floor. The terminal includes a cool air inlet below the floor for drawing in cooling air circulated in the under floor space and a return air inlet in the top surface of the terminal. The cool air and return air is mixed in a mixing chamber and drawn from the mixing chamber by a fan and returned to the room through an outlet vent. The ratio of cool air to return air mixed in the mixing chamber is controlled by a modulating damper which is controlled in response to the temperature of the return air in order to control the room temperature in the region of the terminal in accordance with an adjustable set point. A heater is also provided in the terminal for those occasions where the return air is cooler than the set point.

5.9.2 Unitary Systems:

A unitary air conditioning system comprises an outdoor unit including a compressor for compressing a refrigerant, an outdoor heat exchanger for heat exchange of the refrigerant and an expander connected to the outdoor heat exchanger, for expanding the refrigerant; a duct installed inside a zone of a building; a central blower unit having a heat exchanger connected to the

outdoor unit through a first refrigerant pipe and a blower for supplying the air heat-exchanged by the heat exchanger to the duct; and an individual blower unit including a heat exchanger connected to the outdoor unit through a second refrigerant pipe and a fan for sending the air heat exchanged by the heat exchanger and disposed in a zone in the building, for individually cooling or heating the zone. Accordingly, cooling or heating operation is performed on each zone of the building, and simultaneously, additional individual heating or cooling operation can be performed on a specific space, so that a cost can be reduced and cooling or heating in the building can be efficiently performed.

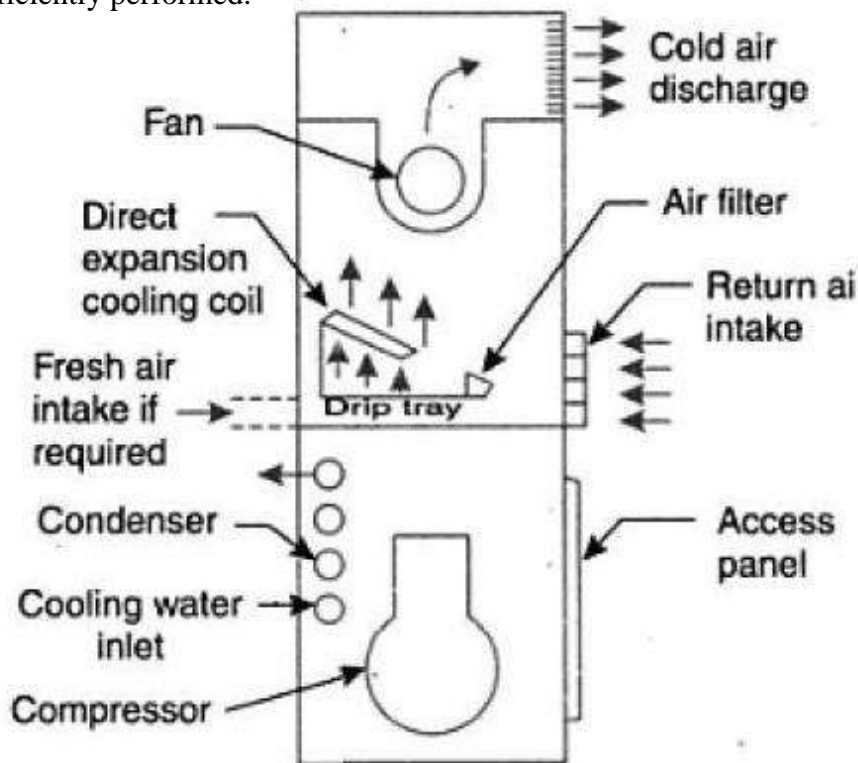


Fig.5.9.2 Unitary Systems

5.10 Window Air-conditioning System:

It is the most commonly used air conditioner for single rooms. In this air conditioner all the components, namely the compressor, condenser, expansion valve or coil, evaporator and cooling coil are enclosed in a single box. This unit is fitted in a slot made in the wall of the room,

or often a window sill.

Window air conditioners are one of the most widely used types of air conditioners because they are the simplest form of the air conditioning systems. Window air conditioner comprises of the rigid base on which all the parts of the window air conditioner are assembled. The base is assembled inside the casing which is fitted into the wall or the window of the room in which the air conditioner is fitted.

The whole assembly of the window air conditioner can be divided into two compartments: the room side, which is also the cooling side and the outdoor side from where the heat absorbed by the room air is liberated to the atmosphere. The room side and outdoor side are separated from each other by an insulated partition enclosed inside the window air conditioner assembly.

In the front of the window air conditioner on the room side there is beautifully decorated front panel on which the supply and return air grills are fitted (the whole front panel itself is commonly called as front grill). The louvers fitted in the supply air grills are adjustable so as to supply the air in desired direction. There is also one opening in the grill that allows access to the Control panel or operating panel in front of the window air conditioner.

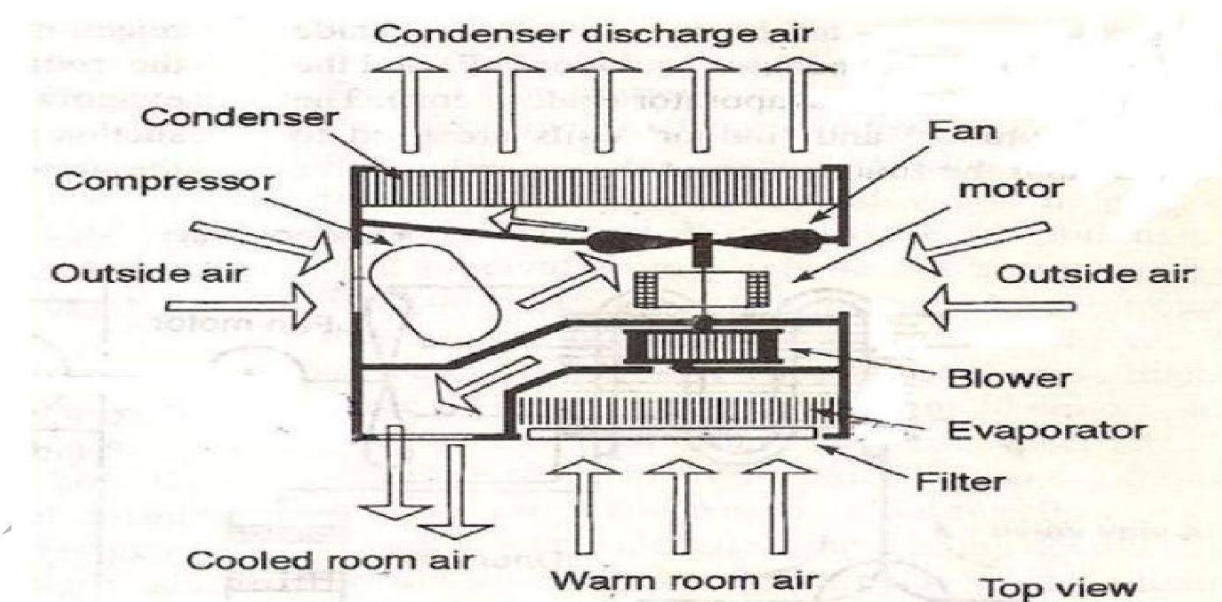


Fig.5.10 Window air conditioning system

5.10.1 Blower:

This is the small blower that is fitted behind the evaporator or cooling coil inside the assembly of the window air conditioner system. The blower sucks the air from the room which first passes over the air filter and gets filtered. The air then passes over the cooling coil and gets chilled. The blower then blows this filtered and chilled air, which passes through the supply air compartment inside the window air conditioner assembly. This air is then delivered into the room from the supply air grill of the front panel.

5.10.2 Propeller fan or the condenser fan:

The condenser fan is the forced draft type of propeller fan that sucks the atmospheric air and blows it over the condenser. The hot refrigerant inside the condenser gives up the heat to the atmospheric air and its temperature reduces.

5.10.3 Fan motor:

The motor inside the window air conditioner assembly is located between the condenser and the evaporator coil. It has double shaft on one side of which the blower is fitted and on the other side the condenser fan is fitted. This makes the whole assembly of the blower, the condenser fan and the motor highly compact.

5.11 Split Air-conditioning System:

The split air conditioner comprises of two parts: the outdoor unit and the indoor unit. The outdoor unit, fitted outside the room, houses components like the compressor, condenser and expansion valve. The indoor unit comprises the evaporator or cooling coil and the cooling fan. For this unit you don't have to make any slot in the wall of the room. Further, the present day split units have aesthetic looks and add to the beauty of the room. The split air conditioner can be used to cool one or two rooms.

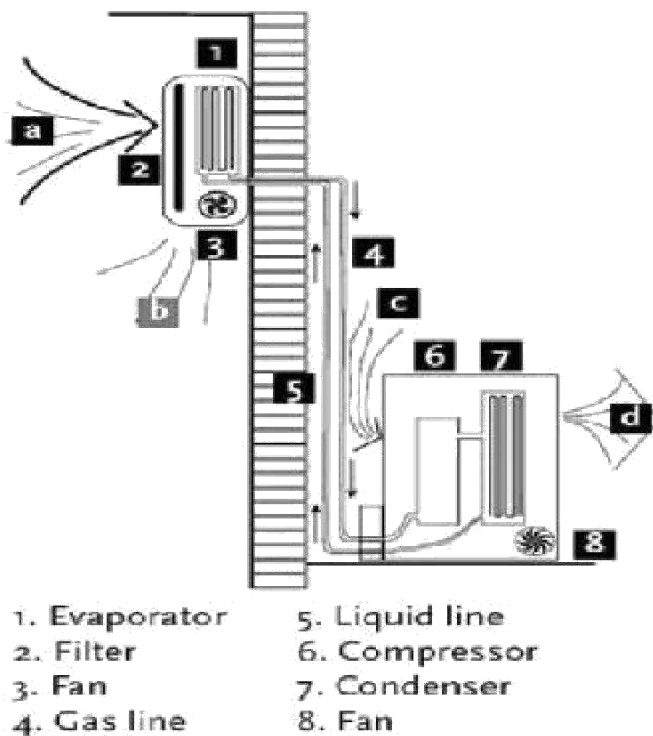


Fig.5.11 Split Air-conditioning System

5.11.1 Evaporator Coil or the Cooling Coil:

The cooling coil is a copper coil made of number turns of the copper tubing with one or more rows depending on the capacity of the air conditioning system. The cooling coil is covered with the aluminum fins so that the maximum amount of heat can be transferred from the coil to the air inside the room.

5.11.2 Air Filter:

The air filter is very important part of the indoor unit. It removes all the dirt particles from the room air and helps supplying clean air to the room. The air filter in the wall mounted type of the indoor unit is placed just before the cooling coil. When the blower sucks the hot room air, it is first passed through the air filter and then through the cooling coil.

5.11.3 Cooling Fan or Blower:

Inside the indoor unit there is also a long blower that sucks the room air or the atmospheric air. It is an induced type of blower and while it sucks the room air it is passed over the cooling coil and the filter due to which the temperature of the air reduces and all the dirt from it is removed. The blower sucks the hot and unclean air from the room and supplies cool and clean air back. The shaft of the blower rotates inside the bushes and it is connected to a small multiple speed motor, thus the speed of the blower can be changed. When the fan speed is changed with the remote it is the speed of the blower that changes.

5.11.4 Drain Pipe:

Due to the low temperature refrigerant inside the cooling coil, its temperature is very low, usually much below the dew point temperature of the room air. When the room air is passed over the cooling coil due to the suction force of the blower, the temperature of the air becomes very low and reaches levels below its dew point temperature. Due to this the water vapor present in the air gets condensed and dew or water drops are formed on the surface of the cooling coil. These water drops fall off the cooling coil and are collected in a small space inside the indoor unit. To remove the water from this space the drain pipe is connected from this space extending to some external place outside the room where water can be disposed off. Thus the drain pipe helps removing dew water collected inside the indoor unit.

5.11.5 Louvers or Fins:

The cool air supplied by the blower is passed into the room through louvers. The louvers help changing the angle or direction in which the air needs to be supplied into the room as per the requirements. With louvers one easily changes the direction in which the maximum amount of the cooled air has to be passed. There are two types of louvers: horizontal and vertical. The horizontal louvers are connected to a small motor and their position can be set by the remote control. One can set a fixed position for the horizontal louvers so that chilled air is passed in a

particular direction only or one can keep it in rotation mode so that the fresh air is supplied throughout the room. The vertical louvers are operated manually and one can easily change their position as per the requirements. The horizontal louvers control flow of air in upper and downward directions of the room, while vertical louvers control movement of air in left and right directions.

5.12 SOLVED PROBLEMS

1. A sling psychrometer gives reading of 25°C dry bulb temperature 15°C wet bulb temperature. The barometer indicates 760 mm of Hg assuming partial pressure of the vapour as 10 mm of Hg. Determine 1. Specific humidity 2. Saturation ratio.

Given Data:

Dry bulb temperature $t_d = 25^{\circ}\text{C}$ Wet
bulb temperature $t_w = 15^{\circ}\text{C}$ Barometer
pressure $p_b = 760\text{ mm of Hg}$
Partial pressure $p_v = 10\text{ mm of Hg}$

To Find:

Specific humidity
Saturation ratio.

Solution:

Specific humidity:

We know that Specific humidity

$$W = \frac{.622 p_v}{p_b - p_v} = \frac{.622 \times 10}{760 - 10}$$

$$= 0.0083 \text{ kg/kg of dry air}$$

Saturation ratio:

From steam table corresponding to dry bulb temperature $t_d = 25^{\circ}\text{C}$

We find the partial pressure $p_s = 0.03166 \text{ bar}$

$$= \frac{0.03166}{0.00133}$$

$$= 23.8 \text{ mm of Hg}$$

We know that Saturation ratio.

$$\mu = \frac{pv(pb - ps)}{ps(pb - pv)}$$

$$= \frac{10(760 - 23.8)}{23.8(760 - 10)} = 0.41$$

RESULT:

1. Specific humidity = 0.0083 kg/kg of dry air
2. Saturation ratio. = **0.41**

2. A two stages, single acting air compressor compresses air to 20bar. The air enters the L.P cylinder at 1bar and 27°C and leaves it at 4.7bar. the air enters the H.P. cylinder at 4.5bar and 27°C. the size of the L.P cylinder is 400mm diameter and 500mm stroke. The clearance volume In both cylinder is 4% of the respective stroke volume. The compressor runs at 200rpm, taking index of compression and expansion in the two cylinders as 1.3, estimate 1. The indicated power required to run the compressor; and 2. The heat rejected

in the intercooler per minute.

GIVEN DATA:

Pressure (P4) = 20bar

Pressure (P1) = 1bar = $1 \times 10^5 \text{ N/m}^2$

Temperature (T1) = 27°C = 27+273 = 300K

Pressure (P2) = 4.7bar

Pressure (P3) = 4.5bar

Temperature (T3) = 27°C = 27+273 = 300K

Diameter (D1) = 400mm 0.4m

$$\text{Stroke (L1)} = 500\text{mm} = 0.5\text{m}$$

$$K = \frac{v_{c1}}{v_{s1}} = \frac{v_{c3}}{v_{s3}} = 4\% = 0.04$$

$$N = 200\text{rpm} ; n = 1.3$$

To Find:

Indicated power required to run the compressor

Solution :

We know the swept volume of the L.P cylinder

$$v_{s1} = \frac{\pi}{4} (D_1)^2 L_1 = \frac{\pi}{4} (0.4)^2 0.5$$

$$= 0.06284 \text{ m}^3$$

And volumetric efficiency,

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

$$= 1 + 0.04 - 0.04 \left(\frac{4.7}{1} \right)^{\frac{1}{1.3}}$$

$$= 0.9085 \text{ or } 90.85\%$$

Volume of air sucked by air pressure compressor,

$$v_1 = v_{s1} \times \eta_v = 0.06284 \times 0.9085 = 0.0571 \frac{\text{m}^3}{\text{stroke}}$$

$$= 0.0571 \times N_w = 0.0571 \times 200 = 11.42 \text{ m}^3/\text{min}$$

And volume of air sucked by H.P compressor,

$$v_3 = \frac{P_1 V_1}{P_3} = \frac{1 \times 11.42}{4.5} = 2.54 \frac{\text{m}^3}{\text{min}}$$

We know that indicated work done by L.P compressor,

$$\begin{aligned} W_L &= \left(\frac{n}{n-1} \right) P_1 v_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \\ &= \left(\frac{1.3}{1.3-1} \right) 1 \times 10^5 \times 11.42 \left[\left(\frac{4.7}{1} \right)^{\frac{1.3-1}{1.3}} - 1 \right] \\ &= 2123.3 \times 10^3 \text{ J/min} = 2123.3 \text{ KJ/min} \end{aligned}$$

And indicated workdone by H.P compressor,

$$\begin{aligned} W_H &= \left(\frac{n}{n-1} \right) P_3 v_3 \left[\left(\frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 1 \right] \\ &= \left(\frac{1.3}{1.3-1} \right) 4.5 \times 10^5 \times 2.54 \left[\left(\frac{4.20}{4.5} \right)^{\frac{1.3-1}{1.3}} - 1 \right] \\ &= 2043.5 \times 10^3 \text{ J/min} = 2034.5 \text{ KJ/min} \end{aligned}$$

Total indicated work done by the compressor,

$$W = W_L + W_H = 2123.3 + 2034.5 = 4157.8 \text{ KJ/min}$$

Indicated power required to run the compressor

$$= 4157.8 / 60 = 69.3 \text{KW}$$

3. In an oil gas turbine installation , air is taken as 1 bar and 30°C . The air is compressed to 4bar and then heated by burning the oil to a temperature of 500°C . If the air flows at the rate of 90Kg/min . Find the power developed by the plant take γ for air as 1.4 C_p as 1KJ/KgK . If 2.4Kg of oil having calorific value of 40,000 KJ/Kg if burned in the combustion chamber per minute. Find the overall efficiency of the plant.

Given Data:

Pressure ($P_4 = P_3$) = 1bar

Pressure ($P_1 = P_2$) = 4bar

Temperature (T_2) = $500^{\circ}\text{C} = 500 + 273 = 773\text{K}$ Mass

flow rate of air (m_a) = 90Kg/min = 1.5Kg/sec Mass

flow rate of fuel (m_f) = 2.4Kg/min = 0.04Kg/sec

Temperature (T_4) = $30^{\circ}\text{C} = 30 + 273 = 303\text{K}$

$\gamma = 1.4$; $C_p = 1\text{KJ/KgK}$; $C_v = 40,000 \text{ KJ/Kg}$

To Find:

Power developed by the plant

Performance of the gas turbine

Overall efficiency of the plant

Solution:

Power developed by the plant:

Let T_1, T_3 = temperature of air at points 1 and 3

We know that isentropic expansion 2-3,

$$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 0.673$$

$$T_3 = T_2 \times 0.673 = 773 \times 0.673 = 520\text{K}$$

Similarly for isentropic compression 4-1:

$$\frac{T_4}{T_1} = \left(\frac{P_4}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 0.673$$

$$T_1 = T_4 / 0.673 = 303 / 0.673 = 450\text{K}$$

Performance of the gas turbine:

We know that work developed by the turbine,

$$W_T = m C_p (T_2 - T_3) = 1.5 \times 1(773 - 520) = 379.5\text{KJ/s}$$

And work developed by the compressor,

$$W_c = m C_p (T_1 - T_4) = 1.5 \times 1(450 - 303) = 220.5\text{KJ/s}$$

Net work or power of the turbine,

$$P = W_T - W_c = 379.5 - 220.5 = 159\text{KJ/s} = 159\text{KW}$$

Overall efficiency of the plant:

We know that the heat supplied per second

$$= m_f \times C = 0.04 \times 40,000 = 1600 \text{ KJ/s}$$

Therefore, overall efficiency of the plant,

$$\eta_o = 159/1600 = 0.099 \text{ or } 9.99\%$$

FRANCIS XAVIER ENGINEERING COLLEGE

5.13 TWO MARK UNIVERSITY QUESTIONS:

Part-A (2 Marks)

1. Name four important properties of a good refrigerant
2. What is the difference between air conditioning and refrigeration?
3. What is the function of the throttling valve in vapour compression refrigeration system?
4. In a vapour compression refrigeration system, where the highest temperature will occur?
5. The vapour absorption system can use low-grade heat energy in the generator. Is true or false?
6. Name any four commonly used refrigerants.
7. Explain unit of Refrigeration.
8. Why throttle valve is used in place of expansion cylinder for vapour compression refrigerant machine.
9. What are the effects of super heat and subcooling on the vapour compression cycle?
10. What are the properties of good refrigerant?
11. How are air-conditioning systems classified?
12. How does humidity affect human comfort?
13. What are the various sources of heat gain of an air-conditioned space?
14. What do you mean by the term infiltration in heat load calculations?

5.14 UNIVERSITY ESSAY QUESTIONS:

Part-B (16 Marks)

1. Draw neat sketch of simple vapor compression refrigeration system and explain. (16)
2. Explain with sketch the working principle of aqua Ammonia refrigeration system. (16)
3. Explain with sketch the working principle of water-Lithium bromide refrigeration system. (16)
4. Briefly explain the cooling load calculation in air conditioning system. (16)
5. Explain winter, summer, and year round Alc system. (16)
6. Explain unitary Alc and central Alc system. (16)
7. Explain any four psychometric processes with sketch. (16)
8. A refrigeration system of 10.5 tonnes capacity at an evaporator temperature of -12°C and a condenser temperature of 27°C is needed in a food storage locker. The refrigerant Ammonia is sub cooled by 6°C before entering the expansion valve. The compression in the compressor is of adiabatic type. Find 1. Condition of vapor at outlet of the compressor. 2. Condition of vapor at the entrance of the Evaporator 3. COP & power required. (16)
9. A sling psychrometer in a lab test recorded the following readings DBT= 35°C , WBT= 25°C Calculate the following 1. Specific humidity 2. Relative humidity 3. Vapor density in air 4. Dew point temperature 5. Enthalpy of mixing per kg of air .take atmospheric pressure=1.0132 bar. (16)

FRANCIS XAVIER ENGINEERING COLLEGE