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SCHOOL OF MECHANICAL ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

SMEA1404 – THERMAL ENGINEERING

Unit -1 - UNIT 1 GAS AND VAPOUR POWER CYCLES – SMEA1404

Syllabus

Air Standard Cycle analysis - Otto, Diesel, Dual, Brayton cycle, Simple Rankine, Reheat and Regeneration cycles performance.

DEFINITION OF A CYCLE

A **cycle** isdefined as a repeated series of operations occurring in a certain order. It may be repeated by repeating the processes in the same order. The cycle may be of imaginary perfectengine or actual engine. The former is called **ideal cycle** and the latter **actual cycle.** In idealcycle all accidental heat losses are prevented and the working substance is assumed to behave

like a perfect working substance.

AIR STANDARD EFFICIENCY

To compare the effects of different cycles, it is of paramount importance that the effect of the calorific value of the fuel is altogether eliminated and this can be achieved by considering air

(which is assumed to behave as a perfect gas) as the working substance in the engine cylinder. Theefficiency of engine using air as the working medium is known as an "**Air standard efficiency**".This efficiency is oftenly called **ideal efficiency.**

The actual efficiency of a cycle is always less than the air-standard efficiency of that cycleunder ideal conditions. This is taken into account by introducing a new term "**Relative efficiency**"which is defined as :

$$\eta_{\text{relative}} = \frac{\text{Actual thermal efficiency}}{\text{Air standard efficiency}}$$

The analysis of all air standard cycles is based upon the following assumptions :

Assumptions :

1. The gas in the engine cylinder is a perfect gas i.e., it obeys the gas laws and has constant specific heats.
2. The physical constants of the gas in the cylinder are the same as those of air at moderate temperatures i.e., the molecular weight of cylinder gas is 29.
   c_p= 1.005 kJ/kg-K, c_v= 0.718 kJ/kg-K.
3. The compression and expansion processes are adiabatic and they take place without internal friction, i.e., these processes are isentropic.
4. No chemical reaction takes place in the cylinder. Heat is supplied or rejected by bringing a hot body or a cold body in contact with cylinder at appropriate points during the process.
5. The cycle is considered closed with the same 'air' always remaining in the cylinder to repeat the cycle.

THE CARNOT CYCLE

This cycle has the *highest possible efficiency* and consists of four simple operations namely,

(a) Isothermal expansion
(b) Adiabatic expansion
(c) Isothermal compression
(d) Adiabatic compression.

The condition of the Carnot cycle may be imagined to occur in the following way:

One kg of a air is enclosed in the cylinder which (except at the end) is made of perfect nonconducting material. A source of heat '*H*' is supposed to provide unlimited quantity of heat, nonconducting cover '*C*' and a sump '*S*' which is of infinite capacity so that its temperature remains unchanged irrespective of the fact how much heat is supplied to it. The temperature of source *H* isT_1 and the same is of the working substance. The working substance while rejecting heat to sump'*S*' has the temperature. T_2*i.e.,* the same as that of sump *S*.

Following are the *four stages* of the Carnot cycle. Refer Fig. 1.1

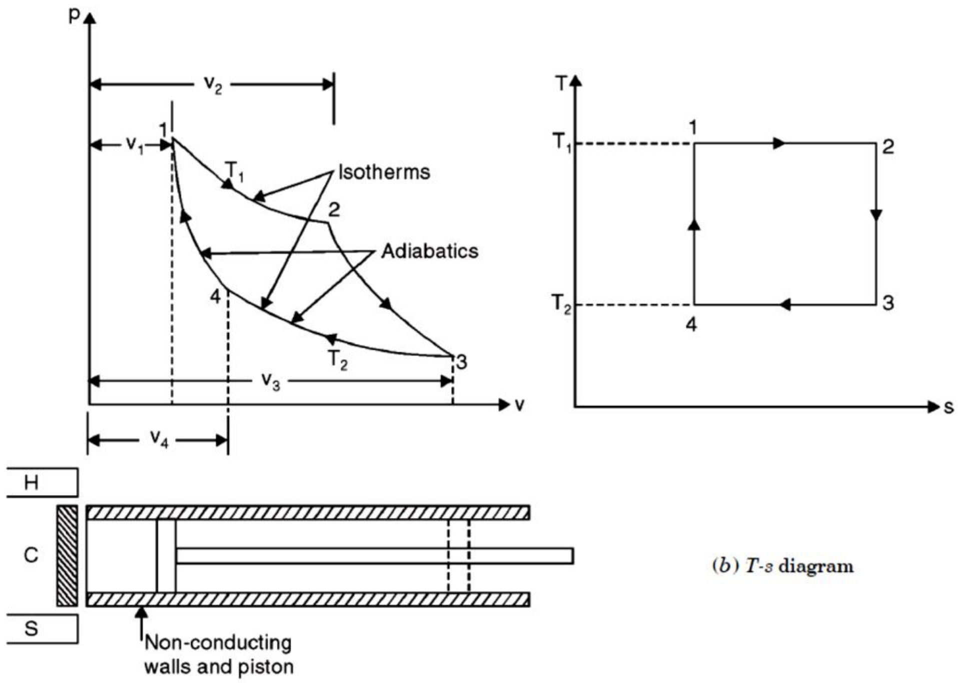


Fig. 1.1. Carnot Cycle

Stage (1). Line 1-2 [Fig. 1.1] represents the isothermal expansion which takes place attemperature *T*1 when source of heat *H* is applied to the end of cylinder. Heat supplied in this caseis given by RT_1 log*e r* and where *r* is the ratio of expansion.

Stage (2). Line 2-3 represents the application of non-conducting cover to the end of the cylinder. This is followed by the adiabatic expansion and the temperature falls from T_1 to T_2.

Stage (3). Line 3-4 represents the isothermal compression which takes place when sump'*S*' is applied to the end of cylinder. Heat is rejected during this operation whose value is given by RT_2 log*e r* where *r* is the ratio of compression.

Stage (4). Line 4-1 represents repeated application of non-conducting cover and adiabatic compression due to which temperature increases from T_2 to T_1.

It may be noted that ratio of expansion during isotherm 1-2 and ratio of compression during isotherm 3-4 must be equal to get a closed cycle.

Fig. 1.1 represents the Carnot cycle on *T-s* coordinates.

Now according to law of conservation of energy,

Heat supplied = Work done + Heat rejected

Work done = Heat supplied – Heat rejected

$= RT_1 \cdot \log_e r - RT_2 \log_e r$

$$\text{Efficiency of cycle} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{R \log_e r\,(T_1 - T_2)}{RT_1 \cdot \log_e r}$$

$$= \frac{T_1 - T_2}{T_1}$$

From this equation, it is quite obvious that if temperature T_2 decreases efficiency increasesand it becomes 100% if T_2 becomes absolute zero which, of course is impossible to attain. Furthermore it is not possible to produce an engine that should work on Carnot's cycle as it wouldnecessitate the piston to travel very slowly during first portion of the forward stroke (isothermalexpansion) and to travel more quickly during the remainder of the stroke (adiabatic expansion)which however is not practicable

CONSTANT VOLUME OR OTTO CYCLE

This cycle is so named as it was conceived by 'Otto'. On this cycle, petrol, gas and many types of oil engines work. It is the standard of comparison for internal combustion engines.

Figs. 1.2. (a) and (b) shows the theoretical p-V diagram and T-s diagrams of this cycle respectively.

- The point 1 represents that cylinder is full of air with volume V_1, pressure P_1 and absolute temperature T_1.
- Line 1-2 represents the adiabatic compression of air due to which P_1, V_1 and T_1 change to P_2, V_2 and T_2 respectively.
- Line 2-3 shows the supply of heat to the air at constant volume so that P_2 and T_2 change to P_3 and T_3 (V_3 being the same as V_2).
- Line 3-4 represents the adiabatic expansion of the air. During expansion P_3, V_3 and T_3 change to a final value of P_4, V_4 or V_1 and T_4, respectively.
- Line 4-1 shows the rejection of heat by air at constant volume till original state (point 1) reaches.

Consider 1 kg of air (working substance):

Heat supplied at constant volume $= c_v(T_3 - T_2)$.

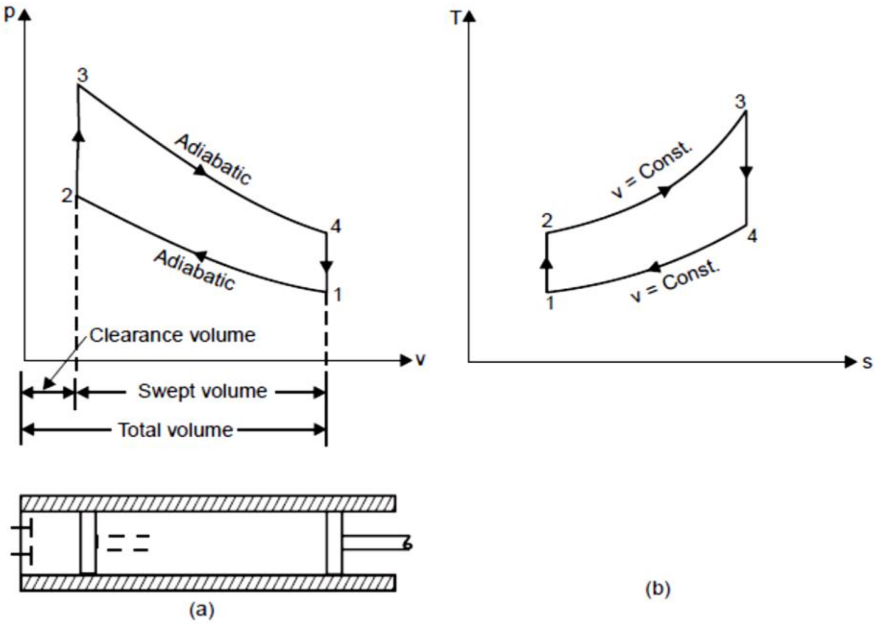
Heat rejected at constant volume $= c_v\,(T_4 - T_1)$.

But, work done = Heat supplied – Heat rejected

$= c_v\,(T_3 - T_2) - c_v\,(T_4 - T_1)$

$$\therefore \quad \text{Efficiency} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v\,(T_3 - T_2) - c_v\,(T_4 - T_1)}{c_v\,(T_3 - T_2)}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2}$$



Figs. 1.2. (a) the theoretical p-V diagram (b) the theoretical T-s diagram of Otto Cycle

Let compression ratio, $r_c\ (= r) = \dfrac{v_1}{v_2}$

and expansion ratio, $r_e\ (= r) = \dfrac{v_4}{v_3}$

(These *two ratios are same* in this cycle)

As $$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1}$$

Then, $$T_2 = T_1 \cdot (r)^{\gamma - 1}$$

Similarly, $$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma - 1}$$

or $$T_3 = T_4 \cdot (r)^{\gamma - 1}$$

Inserting the values of T_2 and T_3 in equation (*i*), we get

$$\eta_{otto} = 1 - \frac{T_4 - T_1}{T_4 \cdot (r)^{\gamma - 1} - T_1 \cdot (r)^{\gamma - 1}} = 1 - \frac{T_4 - T_1}{r^{\gamma - 1}(T_4 - T_1)}$$

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

This expression is known as the air standard efficiency of the Otto cycle. It is clear from the above expression that efficiency increases with the increase in the value of r, which means we can have maximum efficiency by increasing r to a considerable extent, but due to practical difficulties its value is limited to about 8. The net work done per kg in the Otto cycle can also be expressed in terms of p, v. If p is expressed in bar i.e., 10^5 N/m2, then work done

$$W = \left(\frac{p_3v_3 - p_4v_4}{\gamma - 1} - \frac{p_2v_2 - p_1v_1}{\gamma - 1}\right) \times 10^2 \text{ kJ} \quad ...(13.4)$$

Also $$\frac{p_3}{p_4} = r^\gamma = \frac{p_2}{p_1}$$

$\therefore$ $$\frac{p_3}{p_2} = \frac{p_4}{p_1} = r_p$$

where r_p stands for *pressure ratio.*

and $$v_1 = rv_2 = v_4 = rv_3 \qquad \left[\because \frac{v_1}{v_2} = \frac{v_4}{v_3} = r\right]$$

$\therefore$ $$W = \frac{1}{\gamma - 1}\left[p_4v_4\left(\frac{p_3v_3}{p_4v_4} - 1\right) - p_1v_1\left(\frac{p_2v_2}{p_1v_1} - 1\right)\right]$$

$$= \frac{1}{\gamma - 1}\left[p_4v_4\left(\frac{p_3}{p_4r} - 1\right) - p_1v_1\left(\frac{p_2}{p_1r} - 1\right)\right]$$

$$= \frac{v_1}{\gamma - 1}\left[p_4(r^{\gamma - 1} - 1) - p_1(r^{\gamma - 1} - 1)\right]$$

$$= \frac{v_1}{\gamma - 1}\left[(r^{\gamma - 1} - 1)(p_4 - p_1)\right]$$

$$= \frac{p_1v_1}{\gamma - 1}\left[(r^{\gamma - 1} - 1)(r_p - 1)\right] \quad ..$$

Mean effective pressure ($\mathbf{p_m}$) is given by :

$$p_m = \left[\left(\frac{p_3v_3 - p_4v_4}{\gamma - 1} - \frac{p_2v_2 - p_1v_1}{\gamma - 1}\right) \div (v_1 - v_2)\right] \text{ bar}$$

Also $$p_m = \frac{\left[\frac{p_1v_1}{\gamma - 1}(r^{\gamma - 1} - 1)(r_p - 1)\right]}{(v_1 - v_2)}$$

$$= \frac{\frac{p_1v_1}{\gamma - 1}[(r^{\gamma - 1} - 1)(r_p - 1)}{v_1 - \frac{v_1}{r}}$$

$$= \frac{\frac{p_1v_1}{\gamma - 1}[(r^{\gamma - 1} - 1)(r_p - 1)]}{v_1\left(\frac{r - 1}{r}\right)}$$

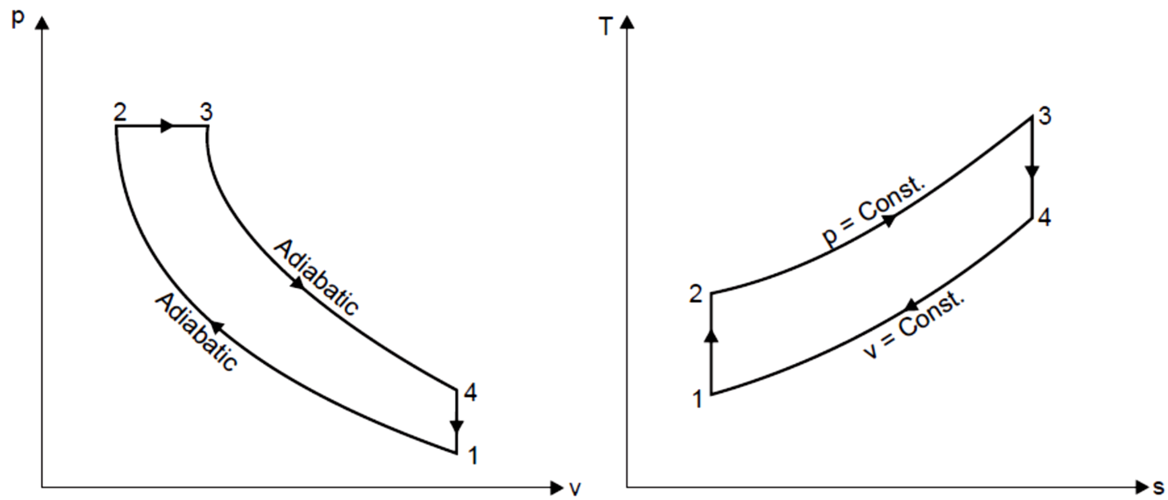
i.e., $$\mathbf{p_m} = \frac{p_1r[(r^{\gamma - 1} - 1)(r_p - 1)]}{(\gamma - 1)(r - 1)}$$

MEP may be thought of as the average pressure acting on a piston during different portions of its cycle.It is the ratio of the work done to stoke volume of the cycle

CONSTANT PRESSURE OR DIESEL CYCLE

This cycle was introduced by Dr. R. Diesel in 1897. It differs from Otto cycle in that heat is supplied at constant pressure instead of at constant volume. Fig. 1.3. (a and b) shows the p-v and T-s diagrams of this cycle respectively.

This cycle comprises of the following operations:



Figs. 1.3. (a) The theoretical p-V diagram (b) The theoretical T-s diagram of Diesel Cycle

(i) 1-2 Adiabatic compression.
(ii) 2-3 Addition of heat at constant pressure.
(iii) 3-4 Adiabatic expansion.
(iv) 4-1 Rejection of heat at constant volume.

- Point 1 represents that the cylinder is full of air. Let P_1, V_1 and T_1 be the corresponding pressure, volume and absolute temperature.
- The piston then compresses the air adiabatically (i.e.,pV^r = constant) till the values become P_2, V_2 and T_2 respectively (at the end of the stroke) at point 2. Heat is then added from a hot body at a constant pressure.
- During this addition of heat let volume increases from V_2 to V_3 and temperature T_2 to T_3, corresponding to point 3.
- This point (3) is called the point of cut-off. The air then expands adiabatically to the conditions P_4, V_4 and T_4 respectively corresponding to point 4.
- Finally, the air rejects the heat to the cold body at constant volume till the point 1 where it returns to its original state.

Consider 1 kg of air.

Heat supplied at constant pressure $= c_p(T_3 - T_2)$

Heat rejected at constant volume $= c_v(T_4 - T_1)$

Work done $=$ Heat supplied $-$ heat rejected

$$= c_p(T_3 - T_2) - c_v(T_4 - T_1)$$

$$\therefore \quad \eta_{\text{diesel}} = \frac{\text{Work done}}{\text{Heat supplied}}$$

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

$$= 1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)} \quad \ldots(i) \quad \left[\because \frac{c_p}{c_v} = \gamma\right]$$

Let compression ratio, $r = \dfrac{v_1}{v_2}$, and cut-off ratio, $\rho = \dfrac{v_3}{v_2}$ *i.e.*, $\dfrac{\text{Volume at cut-off}}{\text{Clearance volume}}$

Now, during *adiabatic compression 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} = (r)^{\gamma - 1} \quad \text{or} \quad T_2 = T_1 \,.\, (r)^{\gamma - 1}$$

During *constant pressure process 2-3,*

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = \rho \quad \text{or} \quad T_3 = \rho \,.\, T_2 = \rho \,.\, T_1 \,.\, (r)^{\gamma - 1}$$

During *adiabatic expansion 3-4*

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma - 1}$$

$$= \left(\frac{r}{\rho}\right)^{\gamma - 1} \quad \left(\because \frac{v_4}{v_3} = \frac{v_1}{v_3} = \frac{v_1}{v_2} \times \frac{v_2}{v_3} = \frac{r}{\rho}\right)$$

$$\therefore \quad T_4 = \frac{T_3}{\left(\frac{r}{\rho}\right)^{\gamma - 1}} = \frac{\rho \,.\, T_1 (r)^{\gamma - 1}}{\left(\frac{r}{\rho}\right)^{\gamma - 1}} = T_1 \,.\, \rho^{\gamma}$$

By inserting values of T_2, T_3 and T_4 in eqn. (i), we get

$$\eta_{\text{diesel}} = 1 - \frac{(T_1 \,.\, \rho^{\gamma} - T_1)}{\gamma\,(\rho \,.\, T_1 \,.\, (r)^{\gamma - 1} - T_1 \,.\, (r)^{\gamma - 1})} = 1 - \frac{(\rho^{\gamma} - 1)}{\gamma (r)^{\gamma - 1} (\rho - 1)}$$

or

$$\eta_{\text{diesel}} = 1 - \frac{1}{\gamma (r)^{\gamma - 1}} \left[\frac{\rho^{\gamma} - 1}{\rho - 1}\right] \quad \ldots(13.7)$$

It may be observed that eqn. (13.7) for efficiency of diesel cycle is different from that of the Otto cycle only in bracketed factor. This factor is always greater than unity, because r > 1. Hence for a given compression ratio, the Otto cycle is more efficient.

The *net work* for diesel cycle can be expressed in terms of pv as follows :

$$W = p_2(v_3 - v_2) + \frac{p_3v_3 - p_4v_4}{\gamma - 1} - \frac{p_2v_2 - p_1v_1}{\gamma - 1}$$

$$= p_2(\rho v_2 - v_2) + \frac{p_3\rho v_2 - p_4 r v_2}{\gamma - 1} - \frac{p_2v_2 - p_1 r v_2}{\gamma - 1}$$

$$\left[\begin{array}{l} \because \ \dfrac{v_3}{v_2} = \rho \quad \therefore \ v_3 = \rho v_2 \ \text{and} \ \dfrac{v_1}{v_2} = r \quad \therefore \ v_1 = r v_2 \\ \text{But } v_4 = v_1 \quad \therefore \quad v_4 = r v_2 \end{array}\right]$$

$$= p_2v_2(\rho - 1) + \frac{p_3\rho v_2 - p_4 r v_2}{\gamma - 1} - \frac{p_2v_2 - p_1 r v_2}{\gamma - 1}$$

$$= \frac{v_2[p_2(\rho - 1)(\gamma - 1) + p_3\rho - p_4 r - (p_2 - p_1 r)]}{\gamma - 1}$$

$$= \frac{v_2\left[p_2(\rho - 1)(\gamma - 1) + p_3\left(\rho - \dfrac{p_4 r}{p_3}\right) - p_2\left(1 - \dfrac{p_1 r}{p_2}\right)\right]}{\gamma - 1}$$

$$= \frac{p_2v_2[(\rho - 1)(\gamma - 1) + \rho - \rho^{\gamma} . r^{1-\gamma} - (1 - r^{1-\gamma})}{\gamma - 1}$$

$$\left[\because \ \frac{p_4}{p_3} = \left(\frac{v_3}{v_4}\right)^{\gamma} = \left(\frac{\rho}{r}\right)^{\gamma} = \rho^{\gamma} r^{-\gamma}\right]$$

$$= \frac{p_1v_1r^{\gamma-1}[(\rho - 1)(\gamma - 1) + \rho - \rho^{\gamma}r^{1-\gamma} - (1 - r^{1-\gamma})}{\gamma - 1}$$

$$\left[\because \ \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} \ \text{or} \quad p_2 = p_1 . r^{\gamma} \ \text{and} \ \frac{v_1}{v_2} = r \quad \text{or} \quad v_2 = v_1 r^{-1}\right]$$

$$= \frac{p_1v_1r^{\gamma-1}[\gamma(\rho - 1) - r^{1-\gamma}(\rho^{\gamma} - 1)]}{(\gamma - 1)} \qquad \text{...(13.8)}$$

Mean effective pressure p_m is given by :

$$p_m = \frac{p_1v_1r^{\gamma-1}[\gamma(\rho - 1) - r^{1-\gamma}(\rho^{\gamma} - 1)]}{(\gamma - 1)v_1\left(\dfrac{r - 1}{r}\right)}$$

or

$$\mathbf{p_m} = \frac{p_1r^{\gamma}[\gamma(\rho - 1) - r^{1-\gamma}(\rho^{\gamma} - 1)]}{(\gamma - 1)(r - 1)} . \qquad \text{...(13.9)}$$

DUAL COMBUSTION CYCLE

This cycle (also called the limited pressure cycle or mixed cycle) is a combination of Otto and Diesel cycles, in a way, that heat is added partly at constant volume and partly at constant pressure ; the advantage of which is that more time is available to fuel (which is injected into theengine cylinder before the end of compression stroke) for combustion. Because of lagging characteristicsof fuel this cycle is invariably used for diesel and hot spot ignition engines.

The dual combustion cycle (Fig 1.4) consists of the following operations :

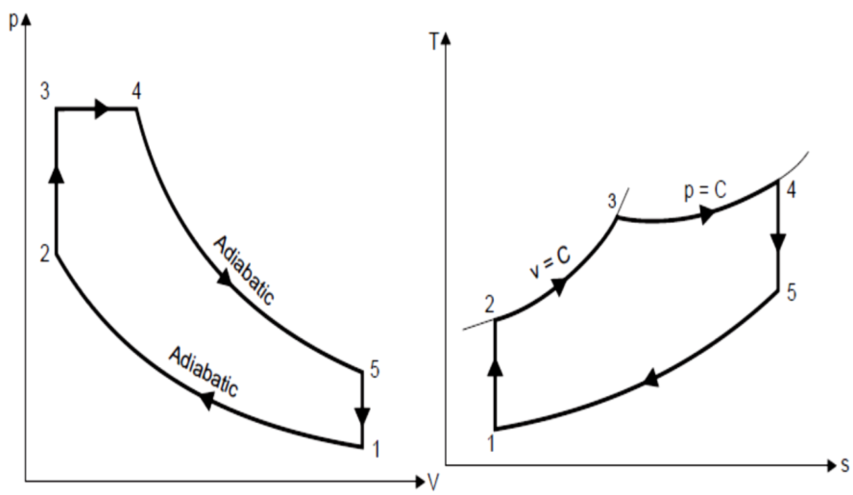
(i) 1-2—Adiabatic compression

(ii) 2-3—Addition of heat at constant volume

(iii) 3-4—Addition of heat at constant pressure

(iv) 4-5—Adiabatic expansion

(v) 5-1—Rejection of heat at constant volume.



Figs. 1.4. (a) The theoretical p-V diagram (b) The theoretical T-s diagram of Dual Cycle

Consider 1 kg of air.

Total heat supplied = Heat supplied during the operation 2-3

+ heat supplied during the operation 3-4

$$= c_v(T_3 - T_2) + c_p(T_4 - T_3)$$

Heat rejected during operation 5-1 $= c_v(T_5 - T_1)$

Work done = Heat supplied − heat rejected

$$= c_v(T_3 - T_2) + c_p(T_4 - T_3) - c_v(T_5 - T_1)$$

$$\eta_{dual} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v(T_3 - T_2) + c_p(T_4 - T_3) - c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}$$

$$= 1 - \frac{c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}$$

$$= 1 - \frac{c_v(T_5 - T_1)}{(T_3 - T_2) + \gamma(T_4 - T_3)} \qquad ...(i) \qquad \left(\because \ \gamma = \frac{c_p}{c_v}\right)$$

Compression ratio, $r = \dfrac{v_1}{v_2}$

During *adiabatic compression process 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} = (r)^{\gamma - 1} \qquad ...(ii)$$

During *constant volume heating process,*

$$\frac{p_3}{T_3} = \frac{p_2}{T_2}$$

or $$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \beta, \text{ where } \beta \text{ is known as } \textbf{pressure or explosion ratio.}$$

or $$T_2 = \frac{T_3}{\beta} \qquad ...(iii)$$

During *adiabatic expansion process,*

$$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1}$$

$$= \left(\frac{r}{\rho}\right)^{\gamma-1} \qquad ...(iv)$$

$$\left(\because \frac{v_5}{v_4} = \frac{v_1}{v_4} = \frac{v_1}{v_2} \times \frac{v_2}{v_4} = \frac{v_1}{v_2} \times \frac{v_3}{v_4} = \frac{r}{\rho}, \rho \text{ being the cut-off ratio}\right)$$

During *constant pressure heating process,*

$$\frac{v_3}{T_3} = \frac{v_4}{T_4}$$

$$T_4 = T_3 \frac{v_4}{v_3} = \rho\, T_3 \qquad ...(v)$$

Putting the value of T_4 in the eqn. (iv), we get

$$\frac{\rho T_3}{T_5} = \left(\frac{r}{\rho}\right)^{\gamma-1} \quad \text{or} \quad T_5 = \rho \,.\, T_3 \,.\, \left(\frac{\rho}{r}\right)^{\gamma-1}$$

Putting the value of T_2 in eqn. (ii), we get

$$\frac{\frac{T_3}{\beta}}{T_1} = (r)^{\gamma-1}$$

$$T_1 = \frac{T_3}{\beta} \cdot \frac{1}{(r)^{\gamma-1}}$$

Now inserting the values of T_1, T_2, T_4 and T_5 in eqn. (i), we get

$$\eta_{\text{dual}} = 1 - \frac{\left[\rho \,.\, T_3\left(\frac{\rho}{r}\right)^{\gamma-1} - \frac{T_3}{\beta} \cdot \frac{1}{(r)^{\gamma-1}}\right]}{\left[\left(T_3 - \frac{T_3}{\beta}\right) + \gamma(\rho T_3 - T_3)\right]} = 1 - \frac{\frac{1}{(r)^{\gamma-1}}\left(\rho^{\gamma} - \frac{1}{\beta}\right)}{\left(1 - \frac{1}{\beta}\right) + \gamma(\rho - 1)}$$

i.e., $$\eta_{\text{dual}} = 1 - \frac{1}{(r)^{\gamma-1}} \cdot \frac{(\beta \,.\, \rho^{\gamma} - 1)}{[(\beta - 1) + \beta\gamma(\rho - 1)]} \qquad ...(13.10)$$

Work done is given by,

$$W = p_3(v_4 - v_3) + \frac{p_4 v_4 - p_5 v_5}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1}$$

$$= p_3 v_3(\rho - 1) + \frac{(p_4 \rho v_3 - p_5 r v_3) - (p_2 v_3 - p_1 r v_3)}{\gamma - 1}$$

$$= \frac{p_3 v_3(\rho - 1)(\gamma - 1) + p_4 v_3\left(\rho - \frac{p_5}{p_4} r\right) - p_2 v_3\left(1 - \frac{p_1}{p_2} r\right)}{\gamma - 1}$$

Also $\dfrac{p_5}{p_4} = \left(\dfrac{v_4}{v_5}\right)^\gamma = \left(\dfrac{\rho}{r}\right)^\gamma$ and $\dfrac{p_2}{p_1} = \left(\dfrac{v_1}{v_2}\right)^\gamma = r^\gamma$

also, $p_3 = p_4,\ v_2 = v_3,\ v_5 = v_1$

$$\therefore \quad W = \frac{v_3[p_3(\rho-1)(\gamma-1) + p_3(\rho - \rho^\gamma r^{1-\gamma}) - p_2(1 - r^{1-\gamma})]}{(\gamma-1)}$$

$$= \frac{p_2 v_2[\beta(\rho-1)(\gamma-1) + \beta(\rho - \rho^\gamma r^{1-\gamma}) - (1 - r^{1-\gamma})]}{(\gamma-1)}$$

$$= \frac{p_1 (r)^\gamma v_1/r[\beta\gamma(\rho-1) + (\beta-1) - r^{1-\gamma}(\beta\rho^\gamma - 1)]}{\gamma-1}$$

$$= \frac{p_1 v_1 r^{\gamma-1}[\beta\gamma(\rho-1) + (\beta-1) - r^{\gamma-1}(\beta\rho^\gamma - 1)]}{\gamma-1} \quad \ldots(13.11)$$

Mean effective pressure (p_m) is given by,

$$p_m = \frac{W}{v_1 - v_2} = \frac{W}{v_1\left(\dfrac{r-1}{r}\right)} = \frac{p_1 v_1[r^{1-\gamma}\beta\gamma(\rho-1) + (\beta-1) - r^{1-\gamma}(\beta\rho^\gamma - 1)]}{(\gamma-1)v_1\left(\dfrac{r-1}{r}\right)}$$

$$\mathbf{p_m} = \frac{p_1(r)^\gamma[\beta(\rho-1) + (\beta-1) - r^{1-\gamma}(\beta\rho^\gamma - 1)]}{(\gamma-1)(r-1)} \quad \ldots(13.12)$$

COMPARISON OF OTTO, DIESEL AND DUAL COMBUSTION CYCLES

Following are the important variable factors which are used as a basis for comparison of the cycles:

- Compression ratio.
- Maximum pressure
- Heat supplied
- Heat rejected
- Net work

Some of the above mentioned variables are fixed when the performance of Otto, Diesel and dual combustion cycles is to be compared.

Efficiency Versus Compression Ratio

Fig. 1.5 shows the comparison for the air standard efficiencies of the Otto, Diesel and Dual combustion cycles at various compression ratios and with given cut-off ratio for the Diesel and Dual combustion cycles. It is evident from the Fig. 13.26 that the air standard efficiencies increase with the increase in the compression ratio. For a given compression ratio Otto cycle isthe most efficient while the Diesel cycle is the least efficient. ($\eta_{otto} > \eta_{dual} > \eta_{diesel}$).

Note. The maximum compression ratio for the petrol engine is limited by detonation. In their respective ratio ranges, the Diesel cycle is more efficient than the Otto cycle.

For the Same Compression Ratio and the Same Heat Input

A comparison of the cycles (Otto, Diesel and Dual) on the p-v and T-s diagrams for the same compression ratio and heat supplied is shown in the Fig. 1.6.

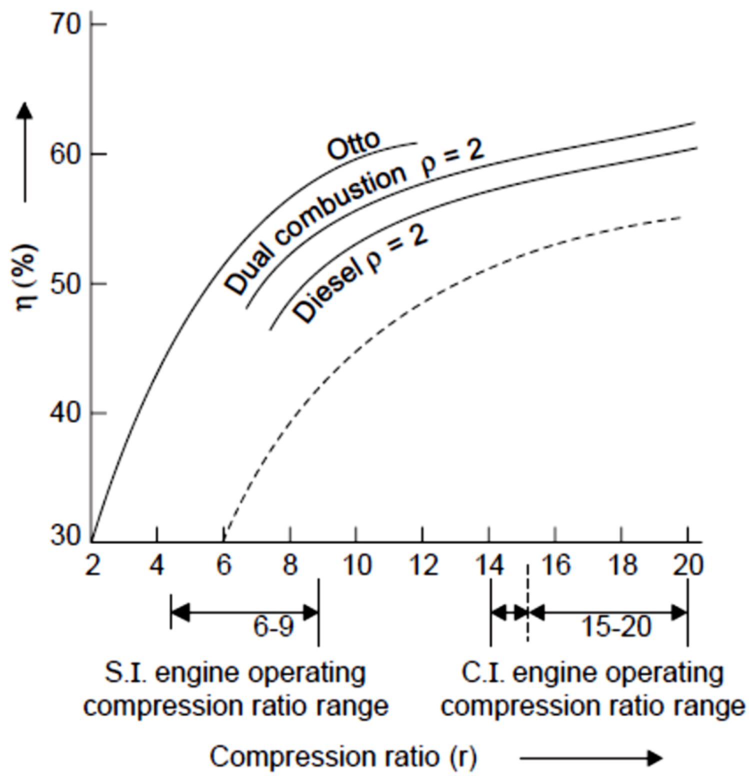


Fig. 1.5 the comparison for the air standard efficiencies of the Otto, Diesel and Dual combustion cycles

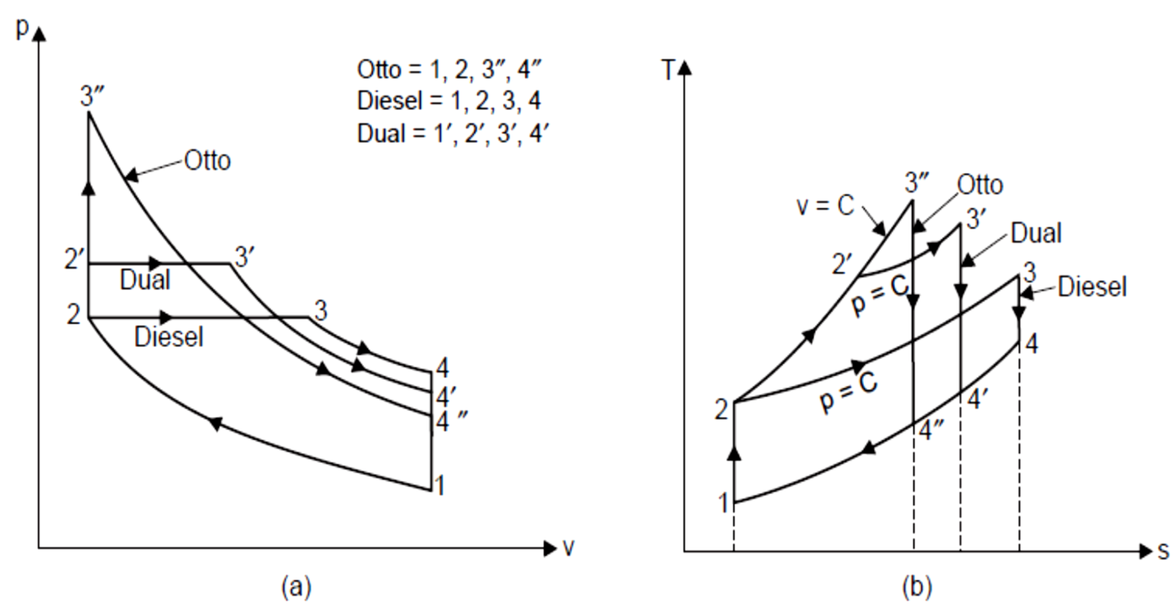


Fig. 1.6. A comparison of the cycles (Otto, Diesel and Dual) on the p-v and T-s diagrams

We know that, $$\eta = 1 - \frac{\text{Heat rejected}}{\text{Heat supplied}} \quad \text{...(13.13)}$$

Since all the cycles reject their heat at the same specific volume, process line from state 4 to 1, the quantity of heat rejected from each cycle is represented by the appropriate area under theline 4 to 1 on the T-s diagram. As is evident from the eqn. (13.13) the cycle which has the least heat rejected will have the highest efficiency. Thus, Otto cycle is the most efficient and Diesel cycle is the least efficient of the three cycles.

$$\eta_{otto} > \eta_{dual} > \eta_{diesel}$$

For Constant Maximum Pressure and Heat Supplied

Fig. 1.7. shows the Otto and Diesel cycles on p-v and T-s diagrams for constant maximum pressure and heat input respectively.

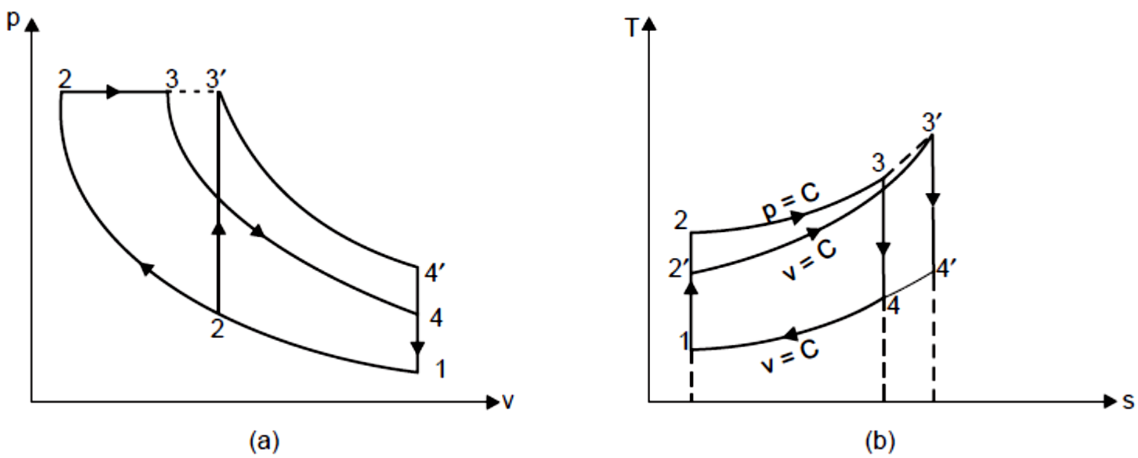


Fig. 1.7. the Otto and Diesel cycles on p-v and T-s diagrams for constant maximum pressure and heat input

— For the maximum pressure the points 3 and 3′ must lie on a constant pressure line.

— On T-s diagram the heat rejected from the Diesel cycle is represented by the area under the line 4 to 1 and this area is less than the Otto cycle area under the curve 4′ to 1 ; hence the Diesel cycle is more efficient than the Otto cycle for the condition of maximumpressure and heat supplied.

GAS TURBINE CYCLE—BRAYTON CYCLE

Ideal Brayton Cycle

Brayton cycle is a constant pressure cycle for a perfect gas. It is also called Joule cycle. The heat transfers are achieved in reversible constant pressure heat exchangers. An ideal gasturbine plant would perform the processes that make up a Brayton cycle. The cycle is shown in theFig. 1.8 (a) and it is represented on p-v and T-s diagrams as shown in Figs. 1.8 (b) and (c).

The various operations are as follows :

Operation 1-2. The air is compressed isentropically from the lower pressure p_1 to the upper pressure p_2, the temperature rising from T_1 to T_2. No heat flow occurs.

Operation 2-3. Heat flows into the system increasing the volume from V_2 to V_3 and temperature from T_2 to T_3 whilst the pressure remains constant at p2.

Heat received = mcp $(T_3 – T_2)$.

Operation 3-4. The air is expanded isentropically from p_2 to p_1, the temperature falling from T_3 to T_4. No heat flow occurs.

Operation 4-1. Heat is rejected from the system as the volume decreases from V_4 to V_1 and the temperature from T_4 to T_1 whilst the pressure remains constant at p1.

Heat rejected = mcp $(T_4 – T_1)$.

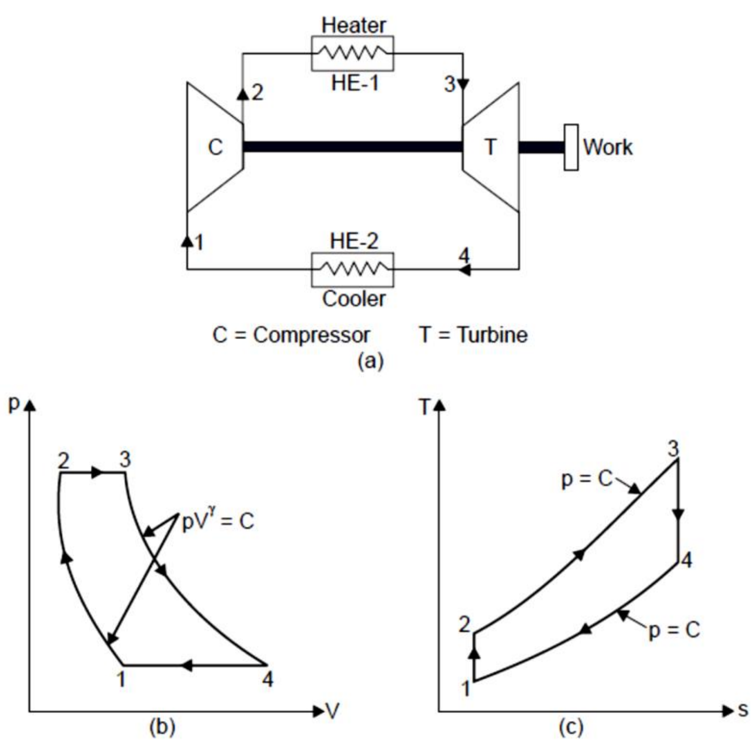


Fig.1.8. Brayton Cycle

$$\eta_{\text{air-standard}} = \frac{\text{Work done}}{\text{Heat received}}$$

$$= \frac{\text{Heat received/cycle} - \text{Heat rejected/cycle}}{\text{Heat received/cycle}}$$

$$= \frac{mc_p\,(T_3 - T_2) - mc_p\,(T_4 - T_1)}{mc_p\,(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Now, from isentropic expansion,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = T_1\,(r_p)^{\frac{\gamma-1}{\gamma}}$$, where r_p = pressure ratio.

Similarly $$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{or} \quad T_3 = T_4\,(r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore \quad \eta_{\text{air-standard}} = 1 - \frac{T_4 - T_1}{T_4(r_p)^{\frac{\gamma-1}{\gamma}} - T_1(r_p)^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} \qquad \text{...(13.16)}$$

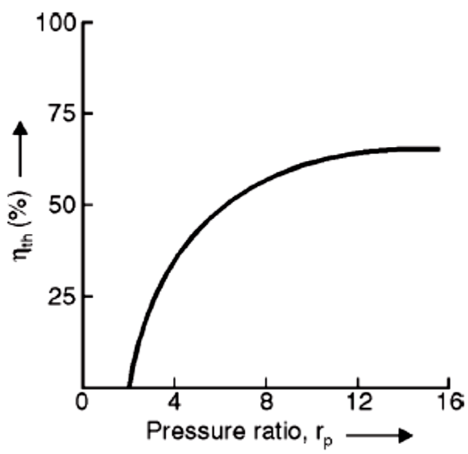


Fig. 1.9. Effect of pressure ratio on the efficiency of Brayton cycle

The eqn. (13.16) shows that the efficiency of the ideal joule cycle increases with the pressure ratio. The absolute limit of upper pressure is determined by the limiting temperature of the material of the turbine at the point at which this temperature is reached by the compression process alone, no further heating of the gas in the combustion chamber would be permissible and the work of expansion would ideally just balance the work of compression so that no excess work would be available for external use.

Pressure Ratio for Maximum Work

Now we shall prove that the pressure ratio for maximum work is a function of the limiting temperature ratio.

Work output during the cycle

$$= \text{Heat received/cycle} - \text{heat rejected/cycle}$$
$$= mc_p(T_3 - T_2) - mc_p(T_4 - T_1)$$
$$= mc_p(T_3 - T_4) - mc_p(T_2 - T_1)$$
$$= mc_p\, T_3\left(1 - \frac{T_4}{T_3}\right) - T_1\left(\frac{T_2}{T_1} - 1\right)$$

In case of a given turbine the minimum temperature T_1 and the maximum temperature T_3 are prescribed, T_1 being the temperature of the atmosphere and T_3 the maximum temperature which the metals of turbine would withstand. Consider the specific heat at constant pressure $c_p$to be constant. Then,

Since, $$\frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1}$$

Using the constant $'z' = \dfrac{\gamma - 1}{\gamma}$,

we have, work output/cycle

$$W = K\left[T_3\left(1 - \frac{1}{r_p^{\,z}}\right) - T_1(r_p^{\,z} - 1)\right]$$

Differentiating with respect to r_p

$$\frac{dW}{dr_p} = K\left[T_3 \times \frac{z}{r_p(z+1)} - T_1\, z r_p^{\,(z-1)}\right] = 0 \text{ for a maximum}$$

$$\therefore \qquad \frac{zT_3}{r_p^{\,(z+1)}} = T_1 z (r_p)^{(z-1)}$$

$$\therefore \qquad r_p^{\,2z} = \frac{T_3}{T_1}$$

$$\therefore \qquad r_p = (T_3/T_1)^{1/2z} \quad \text{i.e.,} \quad r_p = (T_3/T_1)^{\frac{\gamma}{2(\gamma-1)}} \qquad ...(13.17)$$

Thus, the *pressure ratio for maximum work is a function of the limiting temperature ratio.*

13.10.3. Work Ratio

Work ratio is defined as the *ratio of net work output to the work done by the turbine.*

$$\therefore \qquad \text{Work ratio} = \frac{W_T - W_C}{W_T}$$

$$\begin{bmatrix} \text{where,} & W_T = \text{Work obtained from this turbine,} \\ \text{and} & W_C = \text{Work supplied to the compressor.} \end{bmatrix}$$

$$= \frac{mc_p(T_3 - T_4) - mc_p(T_2 - T_1)}{mc_p\,(T_3 - T_4)} = 1 - \frac{T_2 - T_1}{T_3 - T_4}$$

$$= 1 - \frac{T_1}{T_3}\left[\frac{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}{1 - \dfrac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}}\right] = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}} \qquad \text{...(13.18)}$$

Open Cycle Gas Turbine—Actual Brayton Cycle

Refer Fig. 1.10. The fundamental gas turbine unit is one operating on the open cycle in which a rotary compressor and a turbine are mounted on a common shaft. Air is drawn into the compressor and after compression passes to a combustion chamber. Energy is supplied in the combustion chamber by spraying fuel into the air stream, and the resulting hot gases expand through the turbine to the atmosphere. In order to achieve net work output from the unit, the turbine must develop more gross work output than is required to drive the compressor and to overcome mechanical losses in the drive. The products of combustion coming out from the turbine are exhausted to the atmosphere as they cannot be used any more. The working fluids (air and fuel) must be replaced continuously as they are exhausted into the atmosphere.

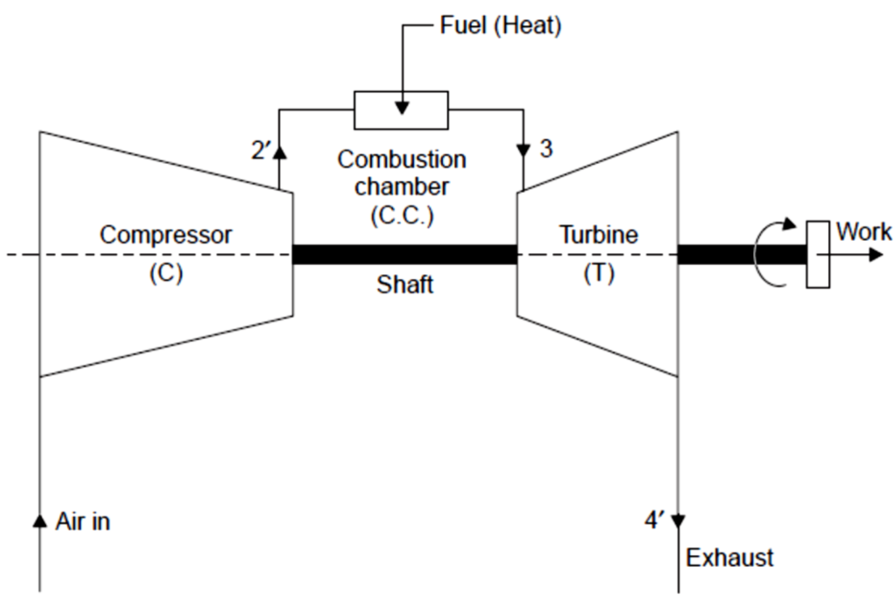


Fig. 1.10. Open cycle gas turbine

If pressure loss in the combustion chamber is neglected, this cycle may be drawn on a T-sdiagram as shown in Fig. 1.11.

1-2’ represents : irreversible adiabatic compression.

2’-3 represents : constant pressure heat supply in the combustion chamber.

3-4’ represents : irreversible adiabatic expansion.

1-2 represents : ideal isentropic compression.

3-4 represents : ideal isentropic expansion.

Assuming change in kinetic energy between the various points in the cycle to be negligibly small compared with enthalpy changes and then applying the flow equation to each part of cycle, for unit mass, we have

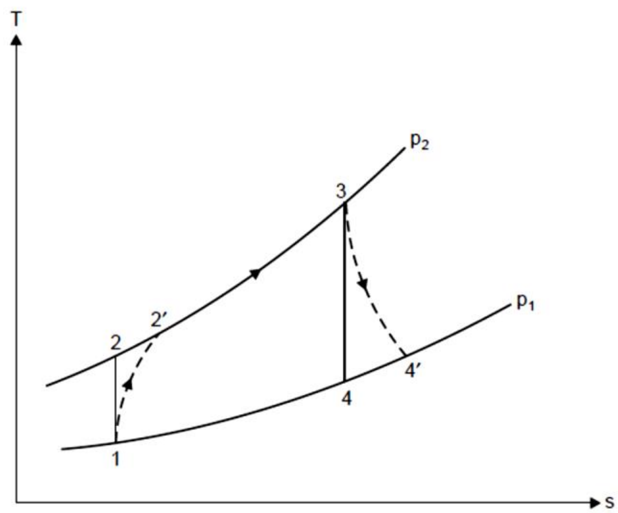
Work input (compressor) $= c_p\,(T_2' - T_1)$

Heat supplied (combustion chamber) $= c_p\,(T_3 - T_2')$

Work output (turbine) $= c_p\,(T_3 - T_4')$

$\therefore$ Net work output = Work output − Work input

$$= c_p\,(T_3 - T_4') - c_p(T_2' - T_1)$$



and
$$\eta_{thermal} = \frac{\text{Net work output}}{\text{Heat supplied}}$$
$$= \frac{c_p(T_3 - T_4') - c_p(T_2' - T_1)}{c_p(T_3 - T_2')}$$

Compressor isentropic efficiency, η_{comp}
$$= \frac{\text{Work input required in isentropic compression}}{\text{Actual work required}}$$
$$= \frac{c_p(T_2 - T_1)}{c_p(T_2' - T_1)} = \frac{T_2 - T_1}{T_2' - T_1} \quad \text{...(13.19)}$$

Turbine isentropic efficiency, $\eta_{turbine}$
$$= \frac{\text{Actual work output}}{\text{Isentropic work output}}$$
$$= \frac{c_p(T_3 - T_4')}{c_p(T_3 - T_4)} = \frac{T_3 - T_4'}{T_3 - T_4} \quad \text{...(13.20)}$$

Note. With the variation in temperature, the value of the specific heat of a real gas varies, and also in the open cycle, the specific heat of the gases in the combustion chamber and in turbine is different from that in the compressor because fuel has been added and a chemical change has taken place. Curves showing the variation of cp with temperature and air/fuel ratio can be used, and a suitable mean value of cp and hence g can be found out. It is usual in gas turbine practice to assume fixed mean value of cp and g for the expansion process, and fixed mean values of cp and g for the compression process. In an open cycle gas turbine unit the mass flow of gases in turbine is greater than that in compressor due to mass of fuel burned, but it is possible to neglect mass of fuel, since the air/ fuel ratios used are large. Also, in many cases, air is bled from the compressor for cooling purposes, or in the case of air-craft at high altitudes, bled air is used for de-icing and cabin air-conditioning. This amount of air bled is approximately the same as the mass of fuel injected therein.

Methods for Improvement of Thermal Efficiency of Open Cycle Gas Turbine Plant

The following methods are employed to increase the specific output and thermal efficiency of the plant :

1. Intercooling
2. Reheating
3. Regeneration.

1. Intercooling. A compressor in a gas turbine cycle utilises the major percentage of powerdeveloped by the gas turbine. The work required by the compressor can be reduced by compressingthe air in two stages and incorporating an intercooler between the two as shown in Fig. 1.12. Thecorresponding T-s diagram for the unit is shown in Fig. 13.38. The actual processes take place asfollows :

1-2' ... L.P. (Low pressure) compression

2'-3 ... Intercooling

3-4' ... H.P. (High pressure) compression

4'-5 ... C.C. (Combustion chamber)-heating

5-6'... T (Turbine)-expansion

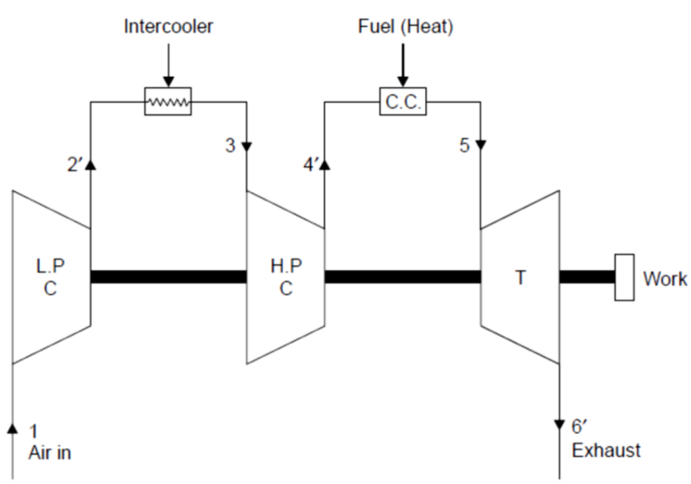


Fig.1.12. Turbine plant with intercooler

The ideal cycle for this arrangement is 1-2-3-4-5-6 ; the compression process without intercooling is shown as 1-L′ in the actual case, and 1-L in the ideal isentropic case. Now,

Work input (*with intercooling*)

$$= c_p(T_2' - T_1) + c_p(T_4' - T_3) \quad ...(13.21)$$

Work input (*without intercooling*)

$$= c_p(T_L' - T_1) = c_p(T_2' - T_1) + c_p(T_L' - T_2') \quad ...(13.22)$$

By comparing equation (13.22) with equation (13.21) it can be observed that the work input with intercooling is less than the work input with no intercooling, when $c_p(T_4{}' - T_3)$ is less than $c_p(T_L{}' - T_2{}')$. This is so if it is assumed that isentropic efficiencies of the two compressors, operating separately, are each equal to the isentropic efficiency of the single compressor which would be required if no intercooling were used. Then $(T_4{}' - T_3) < (T_L{}' - T_2{}')$ since the pressure lines diverge on the T-s diagram from left to the right.

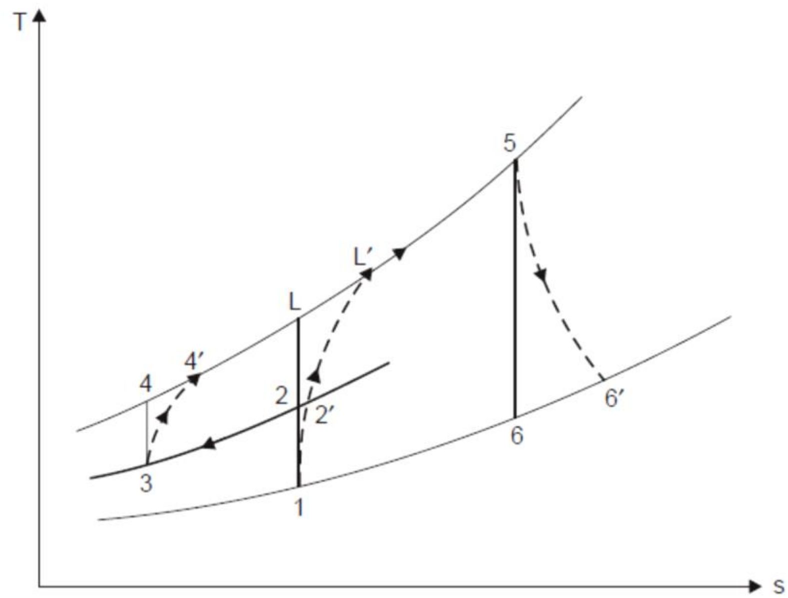


Fig.1.13. *T-s* diagram for the unit

$$\text{Again, work ratio} = \frac{\text{Net work output}}{\text{Gross work output}} = \frac{\text{Work of expansion} - \text{Work of compression}}{\text{Work of expansion}}$$

From this we may conclude that when the compressor work input is reduced then the work ratio is increased.

However the heat supplied in the combustion chamber when intercooling is used in the cycle, is given by,

Heat supplied with intercooling = $C_p(T_5 - T_4{}')$

Also the heat supplied when intercooling is not used, with the same maximum cycletemperature T_5, is given by

Heat supplied without intercooling = $C_p(T_5 - T_L')$

Thus, the heat supplied when intercooling is used is greater than with no intercooling. Although the net work output is increased by intercooling it is found in general that the increase in heat to be supplied causes the thermal efficiency to decrease. When intercooling is used asupply of cooling water must be readily available. The additional bulk of the unit may offset theadvantage to be gained by increasing the work ratio.

2. Reheating: The output of a gas turbine can be amply improved by expanding the gases in two stages with a reheater between the two as shown in Fig. 1.14. The H.P. turbine drives the compressor and the L.P. turbine provides the useful power output. The corresponding T-s diagramis shown in Fig. 1.15. The line 4'-L' represents the expansion in the L.P. turbine if reheating is not employed.

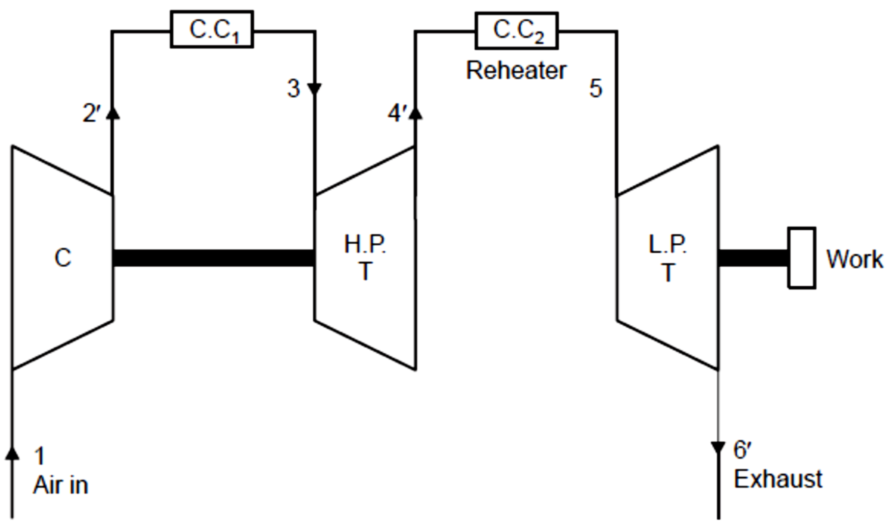


Fig. 1.14. Gas turbine with reheater

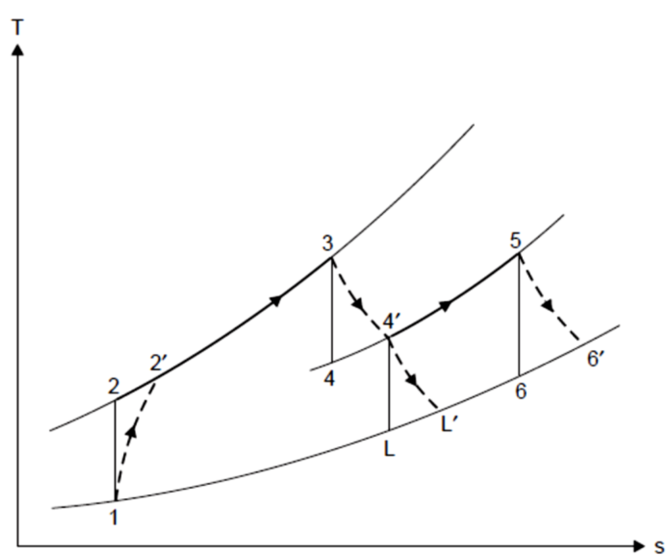


Fig.1.15. *T-s* diagram for the unit

Neglecting mechanical losses the work output of the H.P. turbine must be exactly equal to the work input required for the compressor i.e., $c_{pa}(T_2' - T_1) = c_{pg}(T_3 - T_4')$

The work output (net output) of L.P. turbine is given by,

Net work output (with reheating) $= c_{pg}(T_5 - T_6')$

and Net work output (without reheating) $= c_{pg}(T_4' - T_L')$

Since the pressure lines diverge to the right on *T-s* diagram it can be seen that the temperature difference $(T_5 - T_6')$ is always *greater* than $(T_4' - T_L')$, so that *reheating increases the net work output.*

Although net work is increased by reheating the heat to be supplied is also increased, and the net effect *can be to reduce the thermal efficiency*

Heat supplied $= c_{pg}(T_3 - T_2') + c_{pg}(T_5 - T_4')$.

Note. c_{pa} **and** c_{pg} **stand for specific heats of air and gas respectively at constant pressure.**

3. Regeneration: The exhaust gases from a gas turbine carry a large quantity of heat with them since their temperature is far above the ambient temperature. They can be used to heat the air coming from the compressor thereby reducing the mass of fuel supplied in the combustion chamber. Fig. 1.16 shows a gas turbine plant with a regenerator. The corresponding T-s diagram is shown in Fig. 1.17. 2'-3 represents the heat flow into the compressed air during its passage through the heat exchanger and 3-4 represents the heat taken in from the combustion of fuel.

Point 6 represents the temperature of exhaust gases at discharge from the heat exchanger. The maximum temperature to which the air could be heated in the heat exchanger is ideally that of exhaust gases, but less than this is obtained in practice because a temperature gradient must exist for an unassisted transfer of energy. The effectiveness of the heat exchanger is given by :

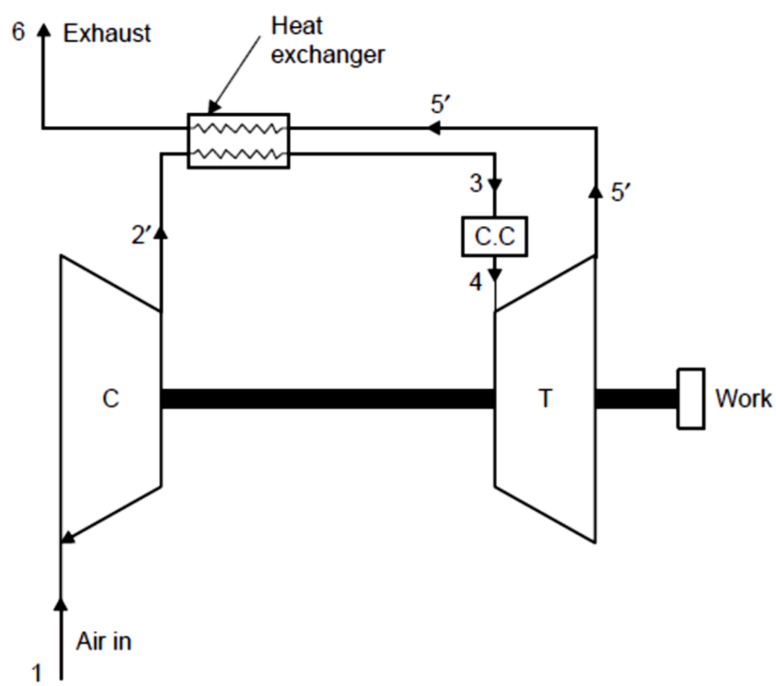


Fig. 1.16. Gas turbine with regenerator

Effectiveness, $\varepsilon = \dfrac{\text{Increase in enthalpy per kg of air}}{\text{Available increase in enthalpy per kg of air}}$

$$= \frac{(T_3 - T_2')}{(T_5' - T_2')} \quad \text{...(13.23)}$$

(assuming c_{pa} and c_{pg} to be equal)

A heat exchanger is usually used in large gas turbine units for marine propulsion or industrial power.

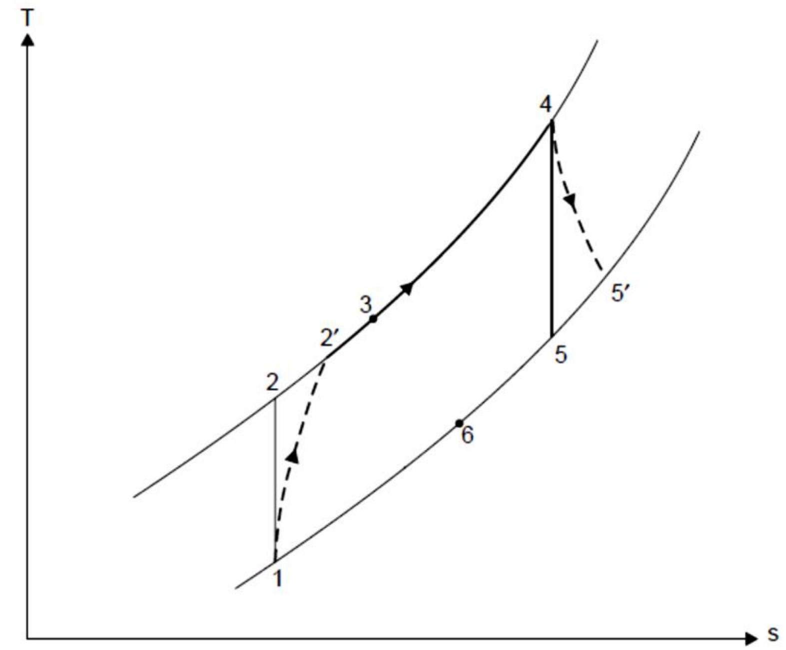


Fig.1.17. *T-s* diagram for the unit

1. A six cylinder petrol engine has a compression ratio of 5:1. The clearance volume of each cylinder is 110CC. It operator on the four stroke constant volume cycle and the indicated efficiency ratio referred to air standard efficiency is 0.56. At the speed of 2400 rpm. It consumer 10kg of fuel per hour. The calorific value of fuel is 44000KJ/kg. Determine the average indicated mean effective pressure.

Given data:

r = 5

V_c =110CC

$\eta_{relative}$ = 0.56

N = 2400rpm

m_f = 10kg

= 10/3600 kg/s

C_v = 44000kJ/kg Z = 6

Solution:

Compression ratio:

$r = V_s + V_c/V_c$ → $5 = V_s + 110/110$ → $V_s = 440CC = 44x10^{-6}m^3$

Air standard efficiency:

$\eta = 1- 1 / (r^{\gamma-1}) = 47.47\%$ ($\gamma = 1.4$)

Relative efficiency:

$\eta_{relative} = \eta_{actual}/ \eta_{air- standard}$ → $0.56 = \eta_{actual}/47.47$

η_{actual} = 26.58%

Actual efficiency = work output/ head input

$0.2658 = W/ m_f C_v$ → $W = 0.2658 x 10/3600x44000$

W = 32.49kw.

The net work output:

$W = P_m x V_s x N/60 x Z$ → $32.49x10^3 = Pm x 440 x10^{-6} x 1200/60 x 6$

P_m = 6.15 bar

2. One kg of air taken through, a) Otto cycle, b) Diesel cycle initially the air is at 1 bar and 290k. The compression ratio for both cycles is 12 and heat addition is 1.9 MJ in each cycle. Calculate the air standard efficiency and mean effective pressure for both the cycles.

Given data:

P_1 = 1 bar = 100KN/ m^2

T_1 = 290K

r = 12

Q_s = 1.9MJ = 1900KJ

Solution:

a)Otto cycle:

For process 1-2: isentropic compression:

$P_2/P_1 = (V_1/V_2)$ → $P_2 = P_1 x r^\gamma$

$P_2 = 3242.3kN/m^2$

$T_2/T_1 = (V_1/V_2)\gamma - 1$→ $T_2 = T_1 x (V_1/V_2) \gamma - 1 = 290 x (12)^{1.4-1}$

$T_2 \quad = 783.55K$

Heat supplied:

$Q \quad = m \times C_v (T_3 - T_2)$

$1900 = 1 \times 0.718 \times (T_3 \; - 783.55)$

$T_3 \quad = 3429.79K$

For process 2-3 : constant volume process

$P_3/P_2 \;= T_3/ T_2 \rightarrow P_3 = P_2 \times T_3/ T_2 \;= 3242.3 \times 3429.79/783.55$

$P \;= 14196.7KN/m^2$

Air standard efficiency:

$\eta \;= 1- 1 / (r^{\gamma-1}) \;= 0.6298$

$\eta \quad = 62.98\%$

Pressure ratio, $\quad K = P_3/P_2 \;= 14196.7/32423 \;= 4.378$

Mean effective pressure,

$Pm = p_1\, r\, (k-1/ \gamma - 1) (r^{\gamma-1}-1/r-1) = \; 100 \times 12 (4.378-1/1.4) [(12^{1.4-1}-1/12-1)]$

$Pm = 1567.93KN/m^2$

b)Diesel cycle:

Consider 1-2 isentropic compression process:

$T_2 = (V_1/V_2)^{\gamma-1} \times T_1 = (r)^{\gamma-1} \times T_1 = (12)^{1.4-1} \times 290$

$T_2 = 783.56K$

Consider 2-3 constant pressure heat addition:

$Q_s \;= C_p (T_3 – T_2)$

$1.9 \times 10^{3} \;= 1.005 \times (T_3 – 783.56)$

$T_3 \;= 2674 K.$

Cut off ratio:

$\rho = V_3/V_2 =T_3/T_2 = 2674/783.56 = 3.413$

Air standard efficiency:

$\eta = 1-1/\boldsymbol{\gamma} (r)^{\gamma-1} \{ \rho^{\gamma-1}/\rho-1\} = 1-1/ 1.4(12)^{1.4-1} \{3.413^{1.4-1}/3.413-1\}$

$= 49.86\%$

Mean effective pressure:

$P_m = P_1 r^{\gamma} [\gamma(\rho-1) - r^{\gamma-1} (\rho^{\gamma}-1)/ (\gamma-1) (r-1)]$

$100 \times (12)^{1.4} [1.4 (3.413-1) - (12)^{1.4-1} (3.413^{1.4-1})]/(1.4-1) (12-1)$

$P_m = 1241 KN/m^2$

3. An air standard dual cycle has a compression ratio of 16 and compression begins at 1 bar and 50ºC. The maximum pressure is 70 bar. The heat transformed to air at constant pressure is equal to heat transferred at constant volume. Find the temperature at all cardial points, cycle efficiency and mean effective pressure take Cp= 1.005KJ/kgK, Cv = 0.718KJ/kgK.

Given data:

$r = 16$

$P_1 = 1$ bar

$T_1 = 50^oC = 323K$

$P_3 = 70$ bar

$Q_{s1} = Q_{s2}$

$C_p = 1.005 kJ/kgk$

$C_v = 0.718 kJ/kgk$

Solution:

Specific volume,

$V_1 \quad RT_1/P_1 = 287 \times 323/1 \times 10^5$

$V_1 = 0.92701 m^3/kg$

$V_2 = 0.05794 m^3/kg$

1-2 isentropic compression process:

$P_2 = (r)^{\gamma} \times P_1 = (16)^{1.4} \times 1 = 48.5$ bar

$T_2 = (r)^{\gamma-1} \times T_1 = (16)^{1.4-1} \times 323$

$T_2 = 979K$

2-3 constant volume heat addition process:

$T_3 = (P_3/P_2) \times T_2 = 70/48.5 \times 979$

$T_3 = 1413K$

$Q_{s1} = Cv (T_3-T_2);\ 0.718(1413 - 979)$

$Q_{s1} = 311.612 KJ/kg$

3-4 constant pressure heat addition:

$Q_{s1} = Q_{s2} = C_p (T_4 – T_3)$

$311.62 = 1.005 (T_4 – 1413)$

$T_4 = 1723K$

$V_4 = T_4/T_3 \ x \ V_3 = 1723/1413 \ X \ 0.05794$

$V_4 = 0.070652 m^3/kg$

Expansion ratio:

$r_e = V_4/V_1 = 0.70652/0.92701 = 0.06215$

4-5 isentropic expansion process:

$P_5 = (r) \ x \ P_4 = (0.076215)^{1.4} \ x \ 70$

$P_5 = 1.9063$ bar

$T_5 = (r)^{ɣ-1} x \ T_4$

$= (0.076215)^{1.4-1} \ x \ 1723$

$= 567$ K

Cut off ratio,

$\rho= V4/ V3$

$= 0.00652/0.05744$

$\rho = 1.2194$

Pressure ratio (K) = $(P_3/P_2) = (70/ 48.5)$

$K = 1.4433$

The cycle efficiency:

$\eta= 1- 1/(r)^{ɣ-1} [(kp^{ɣ}-1)/ (k-1 + Kɣ(p-1)]$

$= 66.34\%$

Net heat supplied to the cycle:

$Q_s = Q_{s1} + Q_{s2}$

$= 311.612 + 311.612$

$= 623.224$ KJ/kg

The mean effective pressure:

$P_m = W/ V_1 – V_2 = 413.45/ (0.92701 -0.05794)$

$P_m = 4.75$ bar

4. In a air standard dual cycle, the compression ratio is 12 and the maximum pressure and temperature of the cycle are 1 bar and 300K. haet added during constant pressure process is upto 3% of the stroke. taking diameter as 25cm and stroke as 30cm, determine.

1) **The pressure and temperature at the end of compression**
2) **The thermal efficiency**
3) **The mean effective pressure**

Take , Cp =1.005KJ/kgK Cv =0.118KJ/kgk ,γ = 1.4

Given data:

P_1 = 1 bar

r = 12

T_1 =300K

K = 3% of Vs = 0.03Vs

P_3 = 70 bar

D = 25 cm

L = 30cm

Solution :

Specific volumes,:

$V_1\ RT_1/ P_1 = 287 \times 300/ 1 \times 10^5$

$= 0.861\ m^3 /kg$

$V_3 = V_2 = V_1/r = 0.861/12$

$= 0.07175 m^3/kg$

$V_4 - V_3 = 0.03 (V_1 - V_2)$

$V_4 = 0.0954275\ m^3/kg$

Cut off ratio:

$\rho = V_4 /V_3 = 0.054275/0.07175$

$\rho = 1.33$

1-2 isentropic compression process:

$P_2 = (r)^{\gamma} \times P1 = (12)^{1.4} \times 1$

$= 32.423$ bar

$V_2 = (r)^{\gamma-1} \times T_1 = (12)^{1.4 -1} \times 300$

$T_2 = 810.57K.$

2-3 constant volume heat addition process

$P_3/T_3 = P_2/T_2$

$T_3 = (P_3/P_2) \times T_2 = (70 / 32.423) \times 810.57$

$T_3 = 1750K$

3-4 constant pressure heat addition process:

$T_4 = (V_4/V_3) \times T_3 = (0.0954275 / 0.07175) \times 1750$

$T_4 = 2327.5$ K

Pressure ratio, $K = (P_3/P_2) = 70/32.423 = 2.159$

Net heat supplied to the cycle:

$Q_S = C_v (T_3 – T_2) + C_p (T_4 - T_3)$

$= 0.718 (1750 -810.57) + 1.005(2327.5-1759)$

= 1254.9 KJ/kg

Efficiency of the cycle:

$\eta = 1- 1/ (r)^{\gamma-1} [(K \times P^{\gamma}-1)/(k-1) + K\boldsymbol{\gamma}(p-1)]$

= 61.92%

Net workdone of the cycle:

$W = \eta \times Q_s$

= 0.6192 x 1254.9

= 777.1 KJ/kg

Mean effective pressure,

$P_m = W/ V_1 – V_2$

= 777.1/ 0.361 – 0.07115

= 984.6 Kpa

P_m = 9.846 bar

5. The compression ratio of a dual cycle is 10. The pressure and temperature at the beginning of the cycle are 1 bar and 27ºC. the maximum pressure of the cycle is limited to 70 bar and heat supplied is limited to 1675KJ/kg fair find thermal efficiency.

Given data:

r= 10

P_1 = 1 bar

T_1 = 27ºC = 300K P_3 = 70 bar

Qs = 1675 KJ/kg

Solution:

Specific volumes:

$V_1 = RT_1/P_1 = 287 \times 300/1 \times 10^5$

$V_2 = V_1/r$

$= 0.861/10$

1-2 isentropic compression process:

$P_1 = (r)^{\gamma} \times P_1 = (10)^{1.4} \times 1 = 25.12$ bar

$T_2 = (r)^{\gamma-1} \times T_1 = (10)^{1.4-1} \times 300 = 753.57K$

2-3 constant volume heat addition process:

$T_3 = (P_3/P_2) \times T_2 = (70 / 25.12) \times 753.37 = 2100K$

Total heat supplied to the cycle:

$Q_s = C_v (T_3 – T_2) + C_p (T_4 – T_3)$

$1675 = 0.718 (2100 -753.57) + 1.005 (T4 – 2100)$

$T_4 = 2804.6$ K

Cut off ratio:

$\rho = V_4/V_3 = T_4/T_3 = 2804.6/2300$

$\rho= 1.3356$

Pressure ratio:

$K = P3/P2 = 70/25.12 = 2.787$

Efficiency of the cycle:

$$\eta = 1 - 1/(r)^{\gamma-1} [(K \times P^{\gamma}-1)/(k-1) + K\gamma(p-1)]$$

$= 59.13\%$

6. In an air standard diesel cycle, the pressure and temperature of air at the beginning of cycle are 1 bar x 40ºC. The temperatures before and after the heat supplied are 400ºC and 1500ºC. Find the air standard efficiency and mean effective pressure of the cycle. What is the power output if it makes 100 cycles / min?

Given data:

$P_1 = 1$ bar $= 100 KN/m^2$

$T_1 = 40^oC = 313K$

$T_2 = 400^oC = 673K$

$T_3 = 1500^oC = 1773K$

Solution :

1-2 isentropic compression:

$T_2/T_1 = (r)^{\gamma-1}$

Compression ratio :

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

$= (673/313)^{1/1.4-1}$

$= 6.779$

2-3 constant pressure heating:

$V_2/T_2 = V_3/T_3$

Cut off ratio, $P = V_3/V_2 = T_3/T_2 = 1773/673 = 2.634$

Efficiency :

$\eta = 1 - 1/\gamma (r)^{\gamma-1} (p^{\gamma}-1/p-1)$

$= 0.4142\%$

Mean effective pressure:

$P_m = P_1 r^\gamma [\gamma(\rho-1) - r^{1-\gamma}(\rho^\gamma-1)]/(\gamma-1)(r-1)$

$=100 \times (6.779)^{1.4} [1.4(2.634-1) - (6.779)^{1-1.4}(2.634^{1.4-1})]/(1.4-1) \times (6.779-1)$

$P_m = 597.77 KN/m^2$

Heat supplied :

$Qs = m \times C_p (T_3 - T_2)$

$= 1 \times 1.005 (1773 - 673)$

$Q_s = 1105.5$ KJ /kg

Work done :

$\eta \times Q_s = 0.4142 \times 1105.5$

$= 457.89$KJ/kg

Power:

= W x cycle /min =457.89 x 100

$=45 \times 10^{-3}$ KJ/kg-min

= 763.16Kj/kg-sec

= 763.16W/kg

7. In a brayton cycle, the air enters the compressor at 1 bar and 25ºC. the pressure of air leaving the compressor is 3 bar and temperature at turbine inlet is 650ºC. determine per kg of aire, i) cycle efficiency ii) heat supplied to air iii) work input iv) heat rejected in the cooler and v) temperature of air leaving the turbine.

Given data:

$P_1 = 1$ bar

$T_1 = 25^oC$

$T_3 = 650^oC$

$P_2 = 3$ bar

Solution :

Consider the process 1-2 adiabatic compression:

$T_2/T_1 = (P_2/P_1)^{\gamma-1/\gamma}$

$T_2 = (P_2/P_1)^{\gamma-1/\gamma} \times T1$

$T_2 = (3/1)^{1.4-1/1.4} \times 298$

3-4 adiabatic expansion:

$T_4/T_3 = (P_4/P_3)^{\gamma-1/\gamma}$

$T_4 = (P_4/P_3)^{\gamma-1/\gamma} \times 923 = 674.3k$

Air standard efficiency :

$\eta = 1- 1/(R_p)^{\gamma-1/\gamma} = 1- 1/(3)^{1.4-1/1.4} = 0.2694$

$= 26.94\%$

Heat supplied $Q_s = C_p (T_3 - T_2) = 1.005 (923 - 408) = 517.575$ KJ/kg

Heat rejected $Q_R = C_p (T_4 - T_1) = 1.008 (673.4 - 298) = 377.277$KJ/kg

Compressor work $W_C = C_p (T_2 - T_1) = 1.005 \times (408 - 298) = 110.55$Kj/kg

Similarly for expander,:

$W_e = C_p \times (T_3 - T_4) = 1.005 (923 - 6734)$

$W_e = 250.848 - 110.55 = 140.288$KJ/kg

Temperature of air leaving the turbine = 673.4K

8. In an air standard brayton cycle, the air enter the compressor at 1 bar and 15⁰C. The pressure leaving the compressor is 5 bar the maximum temperature in the cycle 900⁰C. Find the following .

a) Compressor and expander work per kg of air. b) the cycle efficiency .
If an ideal regenerator is incorporated into the cycle, determine the percentage change in efficiency.

Given data:

$P_1 = P4 = 1 \text{ bar} = 100KN/m^2$

$T_1 = 15^oC$ =288k

$P_2 = P3 = 5$ bar = 500Kn /m^2

$T_3 = 900^oC$ =1173k

Solution:

1-2 isentropic compression:

$T_2/T_1 = (p_2/P_1)^{\gamma-1/\gamma}$; $T_2 = (P_2/P_1)^{\gamma-1/\gamma}$ x T_1 = 456k

Consider the process 3-4 isentropic expansion:

$T_4/T_3 = (P_4/P_3)^{\gamma-1/\gamma}$: $T_4 = (P_4/P_3)^{\gamma-1/\gamma}$ x T_3 = 740.6k

Work done by the compressor when it operates isentropically is given by

Compressor work $W_c = C_p (T_2 – T_1)$ = 1.005(456 – 288) = 168.756KJ

For expander $W_e = C_p (T_3 –T_4)$ = 1.005(1173 – 740.6) = 434.34KJ

Air standard efficiency :

$\eta = 1- 1/(R_p)^{\gamma-1/\gamma}$ = 1- 1/(5) $^{1.4-1/1.4}$ = 36.86%

When ideal regenerator is incorporated:

$T_3 = T_5$ x $T_2 = T_6$

Heat supplied $Q_s = C_p (T_4 –T_3)$

Heat rejected $Q_R = C_p (T_6 –T_1)$

T_1 = 288k

$T_2 =T_6$ = 456k

$T_3 = T_5$ =740.6K

T_4 = 1173k

Q_s = 1.005 (1173 -790.6) = 434.56 KJ/kg

Q_R = 1.005 (456 -288) =186.84 KJ/kg

Efficiency :

η= 1- Q_R/Q_s = 168.84/434.56 =0.6114 = 61.14%

% change in efficiency : = 61.14 – 36.86/61.14 = 39.71%

9. A closed cycle ideal gas plant operates temperature limited of 800ºC and 30ºC and produces a power of 100Kw.The plant is designed such that there is no need for a regenerator. A fuel of calorific value 45000KJ/kg is used. Calculate the mass flow rate of air through the plant and the rate of fuel combustion take C_p = 1 KJ/kgk and γ = 1.4

Given data:

T1 = 30ºC = 303k ,T3 = 800ºC KJ/kg

P = 100KW ,Cp = 1 KJ /kgK , γ = 1.4

Solution:

For maximum net work done:

$T_4 = T_2 = \sqrt{T1\ x T3} = \sqrt{1073 \text{ x } 303}$ = 570.2k

Net work done

$W_{net} = C_p [(T_3 - T_4) - (T_2 - T_1)$ = 235.6 KJ/kg

Total power development

$P = m_a \text{ x } W_{net}$ = 100/ 235.6 = 0.4244kg/sec

Heat supply to the system:

$m_f \text{ x } C_v = m_a \text{ x } C_p \text{ x } (T_3 - T_2)$

$m_f = m_a \text{ x } C_p \text{ x } (T_3 - T_2) /C_v$ 0.4244 x 1 (1073 – 570.2)/45.000=4.742 x 10^{-3} kg/s

Unit -2– INTERNAL COMBUSTION ENGINE PERFORMANCE AND SYSTEMS– SMEA1404

Syllabus

Working of S.I. and C.I engines, two stroke and four stroke engines – Valve timing, port timing and PV diagrams of S.I and C.I engines - Estimation of brake power indicated power, thermal efficiency, Heat balance sheet. Electronic and Common Rail Direct injection systems. Magneto and Battery coil ignition systems, Lubrication and Cooling systems. Supercharging and turbocharging systems.

INTRODUCTION

As the name implies or suggests, the internal combustion engines (briefly written as IC engines) are those engines in which the combustion of fuel takes place inside the engine cylinder. These are petrol, diesel, and gas engines. We have seen in steam engines or steam turbines that the fuel, fed into the cylinder, is in the form of steam which is already heated (or superheated), and is ready for working in the combustion cycle of the engine. But, in case of internal combustion engines, the combustion of fuel takes place inside the engine cylinder by a spark and produces very high temperature as compared to steam engines. The high temperature produced may ruin the metal of cylinder, valves, etc. It is, therefore, necessary to abstract some of heat from the engine cylinder. The abstraction of heat or the cooling of cylinder may be effected by the surrounding air as in case of a motor cycle or aeroplane engine; or by circulating water through jackets surrounding the cylinder barrel and cylinder head. The water cooling is mostly adopted for large pistons.

CLASSIFICATION OF IC ENGINES

The internal combustion engines may be classified in many ways, but the following are important from the subject point of view

1. According to the type of fuel used
   (a) Petrol engines. (b) Diesel engines or oil engines, and (c) Gas engines.
2. According to the method of igniting the fuel
   (a) Spark ignition engines (briefly written as S.1. engines), (b) Compression ignition engines (briefly written as C.I. engines), and (c) Hot spot ignition engines
3. According to the number of strokes per cycle
   (a) Four stroke cycle engines, and (b) Two stroke cycle engines.
4. According to the cycle of operation
   (a) Otto. cycle (also known as constant volume cycle) engines, (b) Diesel cycle (also known as constant pressure cycle) engines, and (c) Dual combustion cycle (also known as semi-diesel cycle) engines.
5. According to the speed of the engine
   (a) Slow speed engines, (b) Medium speed engines, (c) High speed engines.
6. According to the cooling system
   (a) Air-cooled engines. (b) Water-cooled engines. (c) Evaporative cooling engines.
7. According to the method of fuel injection
   (a) Carburetor engines, (b) Air injection engines, (c) Airless or solid injection engines.
8. According to the number of cylinders
   (a) Single cylinder engines (b) Multi-cylinder engines.
9. According to the arrangement of cylinders
   (a) Vertical engines, (b) Horizontal engines, (c) Radial engines, (d) In-line multi-cylinder engines, (e)V-type multi-cylinder engines, (j) Opposite-cylinder engines, (g) Opposite-piston engine

MAIN COMPONENTS OF IC ENGINES

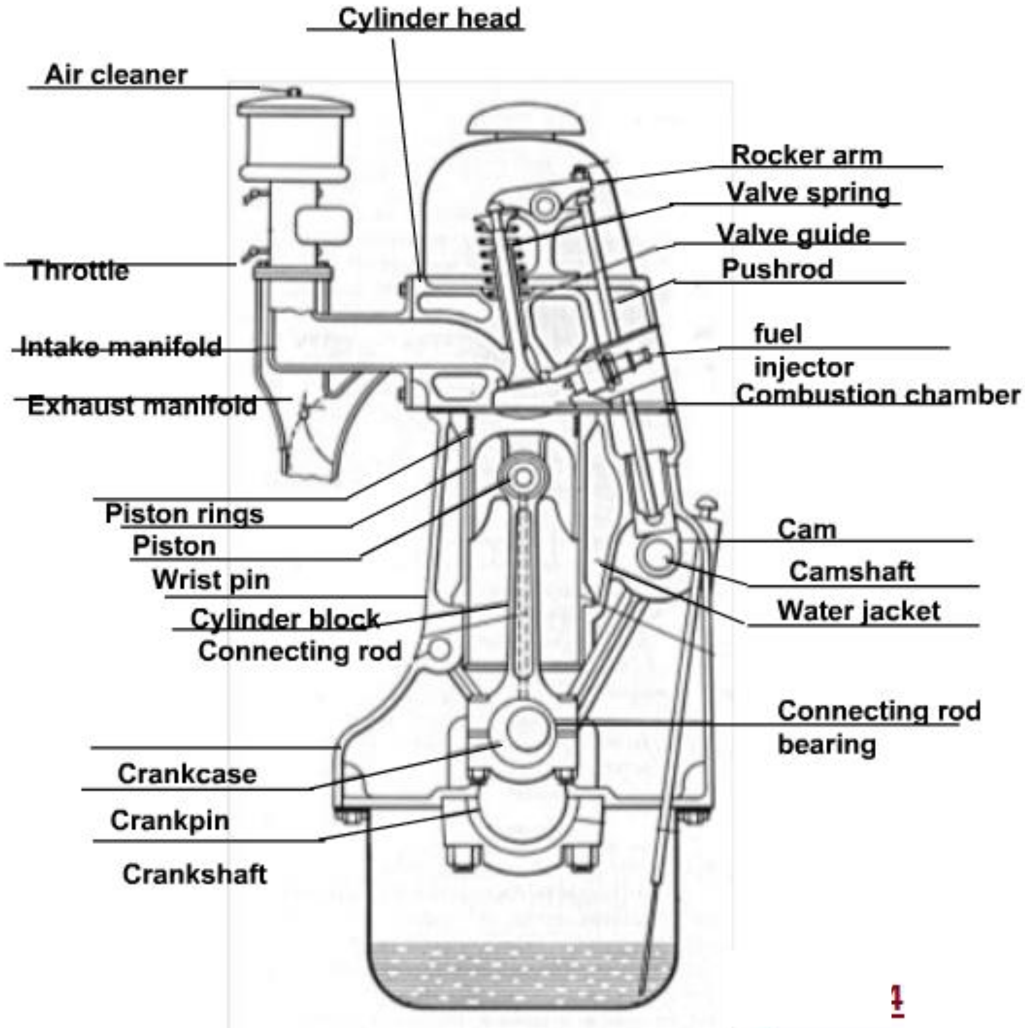


Fig. 2.1. Components of IC Engines

As a matter of fact, an IC engine consists of hundreds of different parts, which are important for its proper working. The description of all these parts is beyond the scope of this book. However, the main components, which are important from academic point of view, are shown and are discussed below:

1. Cylinder. It is one of the most important part of the engine, in which the piston moves to and fro in order to develop power. Generally, the engine cylinder has to withstand a high pressure (more than 50 bar) and temperature (more than 2000°C). Thus the materials for an engine cylinder should be such that it can retain sufficient strength at such a high pressure and temperature. For ordinary engines, the cylinder is made of ordinary cast iron. But for heavy duty engines, it is made of steel alloys or aluminium alloys. In case of multiple cylinder engines, the cylinders are cast in one block known as cylinder block.Sometimes, a liner or sleeve is inserted into the cylinder, which can be replaced when worn out. As the material required for liner is comparatively small, it can be made of alloy cast iron having long life and sufficient resistance to rapid wear and tear to the fast moving reciprocating parts.

2. Cylinder head: It is fitted on one end of the cylinder, and acts as a cover to close the cylinder bore. Generally, the cylinder head contains inlet and exit valves for admitting fresh charge and exhausting the burnt gases. In petrol engines, the cylinder head also contains a spark plug for igniting the fuel-air mixture, towards the end of compression stroke. But in diesel engines, the cylinder head contains nozzle *(i.e.* fuel valve) for injecting the fuel into the cylinder. The cylinder head is, usually, cast as one piece and bolted to one end of the cylinder. Generally, the cylinder block and cylinder head are made from the same material. A copper or asbestos gasket is provided between the engine cylinder and cylinder head to make an air-tight joint.

3. Piston: It is considered as the heart of an l.c. engine, whose main function is to transmit the force exeI1ed by the burning of charge to the connecting rod. The pistons are generally made of aluminium alloys which are light in weight. They have good heat conducting property and also greater strength at higher temperatures.

4. Piston rings: These are circular rings and made of special steel alloys which retain elastic properties even at high temperatures. The piston rings are housed in the circumferential grooves provided on the outer surface of the piston. Generally, there are two sets of rings mounted for the piston. The function of the upper rings is to provide air tight seal to prevent leakage of the burnt gases into the lower portion. Similarly, the function of the lower rings is to provide effective seal to prevent leakage of the oil into the engine cylinder.

5. Connecting rod: It is a link between the piston and crankshaft, whose main function is
to transmit force from the piston to the crankshaft. Moreover, it converts reciprocating motion of the piston into circular motion of the crankshaft, in the working stroke. The upper *(i.e.* smaller) end of the connecting rod is fitted to the piston and the lower *(i.e.* bigger) end to the crank. The special steel alloys or aluminium alloys are used for the manufacture of connecting rods. A special care is required for the design and manufacture of connecting rod, as it is subjected to alternatively compressive and tensile stresses as well as bending stresses.

6. Crankshaft: It is considered as the backbone of an l.c. engine whose function is to
convert the reciprocating motion of the piston into the rotary motion with the help of connecting rod. This shaft contains one or more eccentric portions called cranks. That part of the crank, to which bigger end of the connecting rod is fitted, is called crank pin.It has been experienced that too many main bearings create difficulty of correct alignment. Special steel alloys are used for the manufacture of crankshaft. A special care is required for the design and manufacture of crankshaft.

7. Crank case: It is a cast iron case, which holds the cylinder and crankshaft of an I.c.
engine. It also serves as a sump for the lubricating oil. The lower portion of the crank case is known as bed plate, which is fixed with the help of bolts.

8. Flywheel: It is a big wheel, mounted on the crankshaft, whose function is to maintain its speed constant. It is done by storing excess energy during the power stroke, which is returned during other strokes.

SEQUENCE OF OPERATIONS IN A CYCLE

Strictly speaking, when an engine is working continuously, we may consider a cycle starting from any stroke. We know that when the engine returns back to the stroke where it started we say that one cycle has been completed. The readers will find different sequence of operations in different books. But in this chapter, we shall consider the following sequence of operation in a cycle, which is widely used.

I. Suction stroke: In this stroke, the fuel vapor in correct proportion, is supplied to the
engine cylinder.

2. Compression stroke: In this stroke, the fuel vapor is compressed in the engine cylinder.

3. Expansion or working stroke: In this stroke, the fuel vapor is fired just before the
compression is complete. It results in the sudden rise of pressure, due to expansion of the combustion products in the engine cylinder. This sudden rise of the pressure pushes the piston with

a great force, and rotates the crankshaft. The crankshaft, in turn, drives the machine connected to it.

4. Exhaust stroke: In this stroke, the burnt gases (or combustion products) are exhausted from the engine cylinder, so as to make space available for the fresh fuel vapor.

TWO STROKE AND FOUR STROKE CYCLE ENGINE

In a two-stroke engine, the working cycle is completed in two strokes of the piston or one revolution of the crankshaft. This is achieved by carrying out the suction and compression processes in one stroke (or more precisely in inward stroke), expansion and exhaust processes in the second stroke (or more precisely in outward stroke). In a four-stroke engine, the working cycle is completed in four-strokes of the piston or two-revolutions of the crankshaft. This is achieved by carrying out suction, compression, expansion and exhaust processes in each stroke. It will be interesting to know that from the thermodynamic point of view, there is no difference between two-stroke and four-stroke cycle engines. The difference is purely mechanical.

Advantages and Disadvantage of Two-stroke over Four-stroke Cycle Engines

Advantages

1. A two stroke cycle engine gives twice the number of power strokes than the four stroke cycle engine at the same engine speed. Theoretically, a two-stroke cycle engine should develop twice the power as that of a four-stroke cycle engine. But in actual practice, a two-stroke cycle engine develops 1.7 to 1.8 times greater value for slow speed engines the power developed by four-stroke cycle engine of the same dimensions and speed. This is due to lower compression ratio and effective stroke being less than thetheoretical stroke.
2. For the same power developed, a two-stroke cycle engine is lighter, less bulky and occupies less floor area. Thus it makes, a two-stroke cycle engine suitable for marine engines and other light vehicles.
3. As the number of working strokes in a two-stroke cycle engine are twice than the four-stroke cycle engine, so the turning moment of a two-stroke cycle engine is more uniform. Thus it makes a two-stroke cycle engine to have a lighter flywheel and foundations. This also leads to a higher mechanical efficiency of a two-stroke cycle engine.
4. The initial cost of a two-stroke cycle engine is considerably less than a four-stroke cycle engine.
5. The mechanism of a two-stroke cycle engine is much simpler than a four-stroke cycle engine.
6. The two-stroke cycle engines are much easier to start.

Disadvantages

1. Thermal efficiency of a two-stroke cycle engine is less than that a four-stroke cycle engine, because a two-stroke cycle engine has less compression ratio than that of a four-stroke cycle engine.
2. Overall efficiency of a two stroke cycle engine is also less than that of a four-stroke cycle engine because in a two-stroke cycle, inlet and exhaust ports remain open simultaneously for some time. In spite of careful design, a small quantity of charge is lost from the engine cylinder.
3. In case of a two-stroke cycle engine, the number of power strokes is twice as those of a four-stroke cycle engine. Thus the capacity of the cooling system must be higher. Beyond a certain limit, the cooling capacity offers a considerable difficulty. Moreover, there is a greater wear and tear in a two-stroke cycle engine.

4. The consumption of lubricating oil is large in a two-stroke cycle engine because of high operating temperature.
5. The exhaust gases in a two-stroke cycle engine create noise, because of short time available for their exhaust.

VALVE TIMING DIAGRAM

A valve timing diagram is a graphical representation of the exact moments, in the sequence of operations, at which the two valves *(i.e.* inlet and exhaust valves) open and close as well as firing of the fuel. It is, generally, expressed in terms of angular positions of the crankshaft. Here we shall discuss theoretical valve timing diagrams for four stroke and two stroke cycle engines.

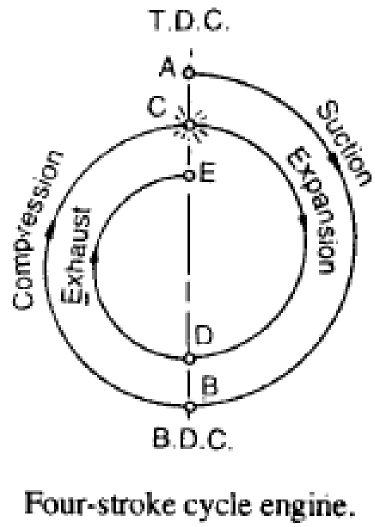


Fig. 2.2. Valve Timing Diagram of Four-Stroke Engine

1. Theoretical valve timing diagram for four-stroke cycle engine

The theoretical valve timing diagram for a four-stroke cycle engine is shown In this diagram, the inlet valve opens at *A* and the suction takes place from *A* to *B.* The crankshaft revolves through 180^0 and the piston moves from *T.D.C.* to *B.D.C. At B,* theinlet valve closes and the compression takes place from *B* to C. The crankshaft revolves through 180^0 and the piston moves from *B.D.* C. to *T.D.* C. At C, the fuel is fired and theexpansion takes place from C to *D.* The crankshaft revolves through 180^0 and the piston again moves from *T.D.C.* to *B.D.C.* At *D,* the exhaust valve opens and the exhaust takes place from *D* to *E.* The crankshaft again revolves through 180^0 and the piston moves back to *T.D.*C.

2. Theoretical valve timing diagram for two-stroke cycle engine.

The theoretical valve timing diagram for a two-stroke cycle engine is shown. In this diagram, the fuel is fired at *A* and the expansion of gases takes place from *A* to *B.* The crankshaft revolves through approximately 120^0 and the piston moves from *T.D.C.* towards *B.D.C.* At *B,* the valves open and suction as well as exhaust take place from *B* to C. The crankshaft revolves through approximately 120^0 and the piston moves first to *B.D.C* and then little upwards. At C. both the valves close and compression takes place from C to *A.* The crankshaft revolves through approximately 120^0 and the piston moves to T.D.C

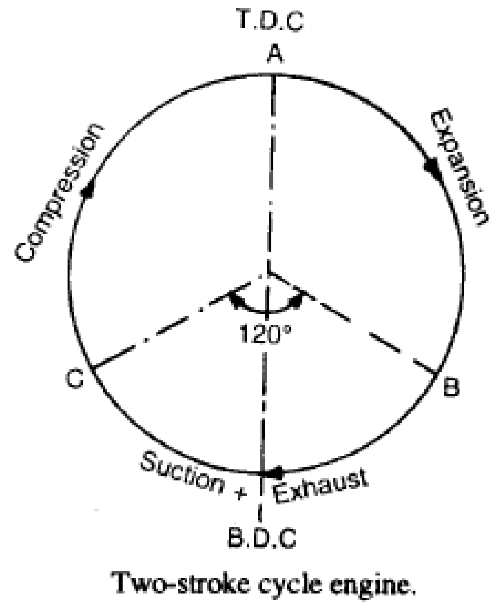


Fig. 2.2. Port Timing Diagram of Two-Stroke Engine

FOUR STROKE CYCLE PETROL ENGINE

It is also known as Otto cycle. It requires four strokes of the piston to complete one cycle of operation in the engine cylinder. The four strokes of a petrol engine sucking fuel-air mixture (petrol mixed with proportionate quantity of air in the carburetor known as charge) are described below:

1. Suction or charging stroke: In this stroke, the inlet valve opens and charge is sucked into the cylinder as the piston moves downward from top dead centre *(T.D.C.).* It continues till the piston reaches its bottom dead centre *(B.D.* C.) as shown in (a).

2. Compression stroke: In this stroke, both the inlet and exhaust valves are closed and the charge is compressed as the piston moves upwards from *B.D.* C. to *TD.* C. As a result of compression, the pressure and temperature of the charge increases considerably (the actual values depend upon the compression ratio). This completes one revolution of the crank shaft. The compression stroke is shown in (b).

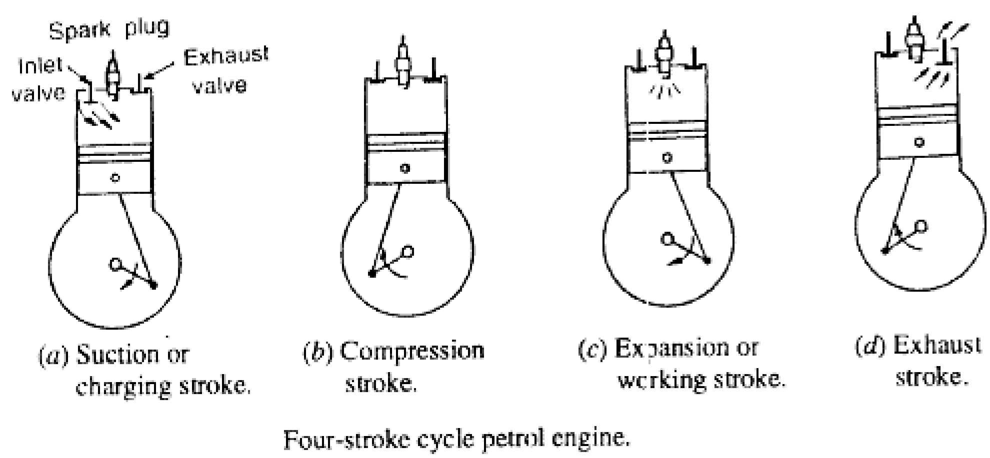


Fig. 2.3 working of 4-stroke Petrol Engine

3. Expansion or working stroke Shortly before the piston reaches *T.D.*C. (during compression stroke), the charge is ignited with the help of a spark plug. It suddenly increases the pressure and temperature of the products of combustion but the volume, practically, remains constant. Due to the rise in pressure, the piston is pushed down with a great force. The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy produced is transformed into mechanical work. It may be noted that during this working stroke, as shown in *(c)*, both the valves are closed and piston moves from *T.D.C.* to *B.D.C*

4. Exhaust stroke: In this stroke, the exhaust valve is open as piston moves from *B.D.C.* to *T.D.*C. This movement of the piston pushes out the products of combustion, from the engine cylinder and are exhausted through the exhaust valve into the atmosphere, as shown in *(d).* This completes the cycle, and the engine cylinder is ready to suck the charge again.

Actual Indicator Diagram for a Four-Stroke Cycle Petrol Engine

The actual indicator diagram for a four stroke cycle petrol engine is shown. The suction stroke is shown by the line 1-2, which lies below the atmospheric pressure line. It is this pressure difference, which makes the fuel-air mixture to flow into the engine cylinder. The inlet valve offers some resistance to the incoming charge. That is why, the charge can not enter suddenly into the engine cylinder. As a result of this, pressure insidethe cylinder remains somewhat below the atmospheric pressure during the suction stroke. Thecompression stroke is shown by the line 2-3, which shows that the inlet valve closes (lVC)a little beyond 2 (i.e. BDC*)*. At the end of this stroke, there is an increase in the pressure inside the engine cylinder. Shortly before the end of compression stroke *(i.e. TDC),* the charge is ignited (lGN)with the help of spark plug as shown in the figure. The sparking suddenly increases pressure and temperature of the products of combustion. But

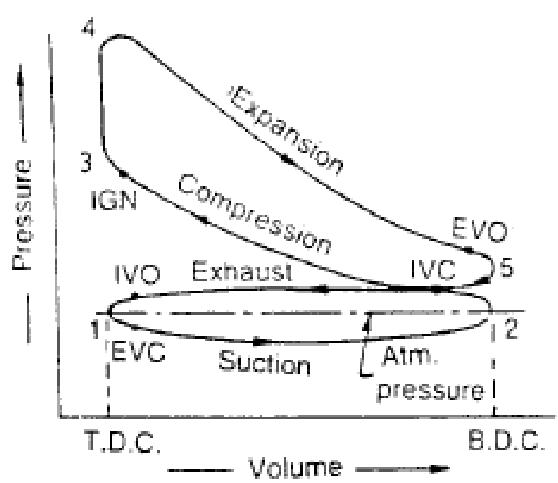


Fig. 2.4.Actual p-V diagram of 4-stroke Petrol Engine

The volume, practically, remains constant as shown by the line 3-4. The expansion stroke is shown by the line 4-5, in which the exit valve opens *(EVO)* a little before 5 *(i.e. BDC).* Now the burnt gases are exhausted into the atmosphere through the exit valve. The exhaust stroke is shown by the line 5-1, which lies above the atmospheric pressure line. It is this pressure difference, which makes the burnt gases to flow out of the engine cylinder. The exit valve offers some resistance to the outgoing burnt gases. That is why the burnt gases cannot escape suddenly from the engine cylinder. As a result of this, pressure inside the cylinder remains somewhat above the atmospheric pressure line during the exhaust stroke

Valve timing diagram for a four-stroke cycle petrol engine

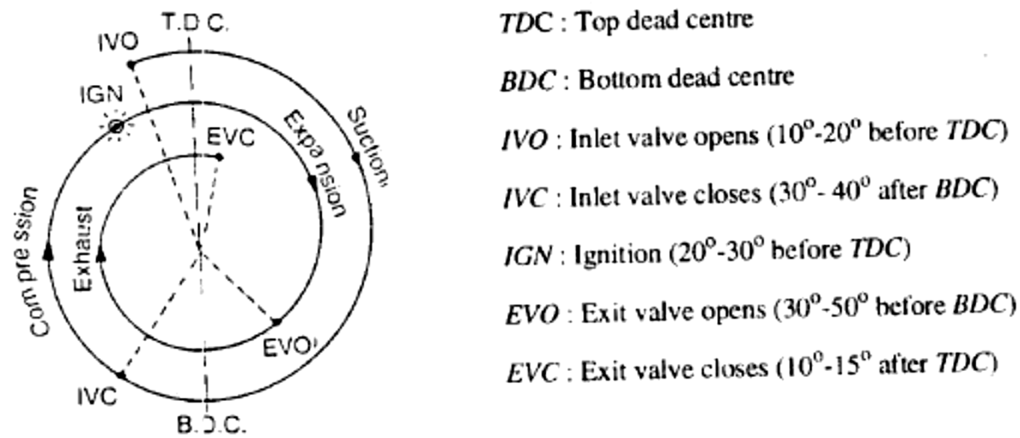


Fig. 2.5.Actual Valve timing diagram of 4-stroke Petrol Engine

In the valve timing diagram, as shown we see that the inlet valve opens before the piston reaches *TDC* or in other words, while the piston is still moving up before the beginning of the suction stroke. Now the piston reaches the *TDC* and the suction stroke starts. The piston reaches the *BDC* and then starts moving up. The inlet valve closes, when the crank has moved a little beyond the *BDC* This is done as the incoming charge continues to flow into the cylinder although the piston is moving upwards from *BDC* Now the charge is compressed (with both valves closed) and then and temperature) push the piston downwards with full force and the expansion or working stroke takes place. Now the exhaust valve opens before the piston again reaches *BDC* and the burnt gases start leaving the engine cylinder. Now the piston reaches *BDC* and then starts moving up, thus performing the exhaust stroke. The inlet valve opens before the piston reaches *TDC* to start suction stroke. This is done as the fresh incoming charge helps in pushing out the burnt gases. Now the piston again reaches *TDC,* and the suction stroke starts. The exit valve closes after the crank has moved a little beyond the *TDC.* This is done as the burnt gases continue to leave the engine cylinder although the piston is moving downwards. It may be noted that for a small fraction of a crank revolution, both the inlet and outlet valves are open. This is known as valve overlap.

FOUR-STROKE CYCLE DIESEL ENGINE

It is also known as *compression ignition engine* because the ignition takes p\ace due to the heat produced in the engine cylinder at the end of compression stroke. The four strokes of a diesel engine sucking pure air are described below:

1. Suction or charging stroke: In this stroke, the inlet valve opens and pure air is sucked into the cylinder as the piston moves downwards from the top dead centre *(TDC).* It continues till the piston reaches its bottom dead centre *(BDC)* as shown *(a).*

2. Compression stroke: In this stroke, both the valves are closed and the air is compressed as the piston moves upwards from *BDC* to *TDC.* As a result of compression, pressure and temperature of the air increases considerably (the actual value depends upon the compression ratio). This completes one revolution of the crank shaft. The compression stroke is shown in *(b).*

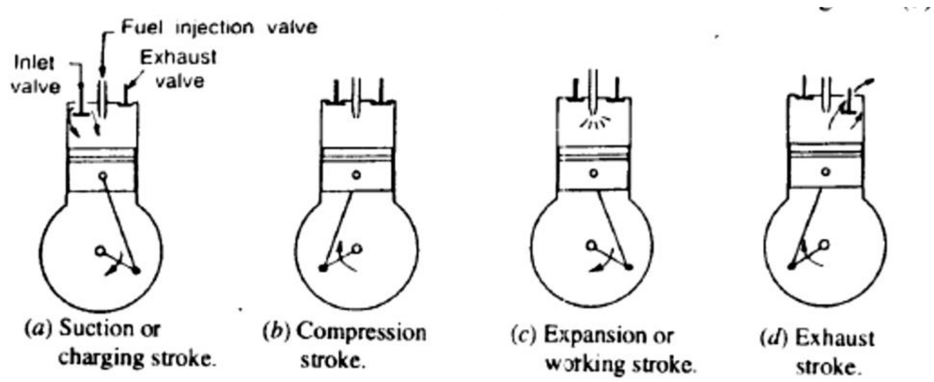


Fig. 2.6 working of 4-stroke Diesel Engine

3. Expansion or working stroke: Shortly before the piston reaches the *TDC* (during the compression stroke), fuel oil is injected in the form of very fine spray into the engine cylinder, through the nozzle, known as fuel injection valve. At this moment temperature of the compressed air is sufficiently high to ignite the fuel. It suddenly increases the pressure and temperature of the products of combustion. The fuel oil is continuously injected for a fraction of the revolution. The fuel oil is assumed to be burnt at constant pressure. Due to increased pressure, the piston is pushed down with a great force. The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy is transformed into mechanical work. It may be noted that during this working stroke, both the valves are closed and the piston moves from *TDC* to *BDC.*
4. Exhaust stroke: In this stroke, the exhaust valve is open as the piston moves from *BDC* to *TDC.* This movement of the piston pushes out the products of combustion from the engine cylinder through the exhaust valve into the atmosphere. This completes the cycle and the engine cylinder is ready to suck the fresh air again.

Actual Indicator Diagram for a Four-Stroke Cycle Diesel Engine

The actual indicator diagram for a four-stroke cycle diesel engine is shown. The suction stroke is shown by the line 1-2 which lies below the atmospheric pressure line. It is this pressure difference, which makes the fresh air to flow into the engine cylinder. The inlet valve offers some resistance to the incoming air. That is why, the air can not enter suddenly into the engine cylinder. As a result **of this pressure inside the cylinder remains somewhat below** the atmospheric pressure during the suction stroke. The compression stroke is shown by the line 2-3, which shows that the inlet valves closes *(/VC)* a little beyond 2 *(i.e. BDC).*At the end of this stroke, there is an increase of pressure inside the engine cylinder. Shortly before the end of compression stroke *(i.e. TDC),* fuel valve opens *(FVO)* and the fuel is injected into the engine cylinder. The fuel is ignited.

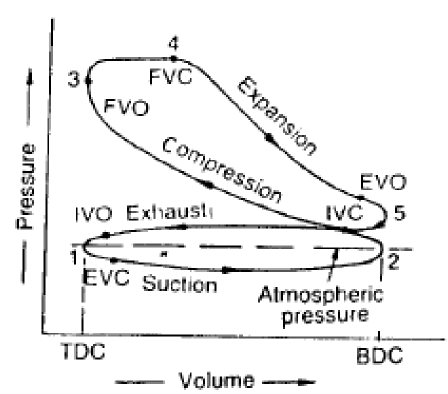


Fig. 2.7.Actual p-V diagram of 4-stroke Diesel Engine

The ignition suddenly increases volume and temperature of the products of combustion. But the pressure, practically, remains constant as shown by the line 3-4. The expansion stroke is shown by the line 4-5, in which the exit valve opens a little before 5 *(i.e. BDC).* Now the burnt gases are exhausted into the atmosphere through the exhaust valve. The exhaust stroke is shown by the line 5-1, which lies above the atmospheric pressure line. It is this pressure difference, which makes the burnt gases to flow out of the engine cylinder. The exhaust valve offers some resistance to the outgoing burnt gases. That is why, the burnt gases cannot (escape suddenly from the engine cylinder. As a result of this, pressure inside the cylinder remains somewhat above the atmospheric pressure during the exhaust stroke.

Valve Timing Diagram for a Four-Stroke Cycle Diesel Engine

In the valve timing diagram as shown we see that the inlet valve opens before thepiston reaches *TDC;* or in other words while the piston is still moving up before the beginning of the suction stroke. Now the piston reaches the *TDC* and the suction stroke starts. The piston reaches the *BDC* and then starts moving up. The inlet valve closes, when the crank has moved a little beyond the BDC.

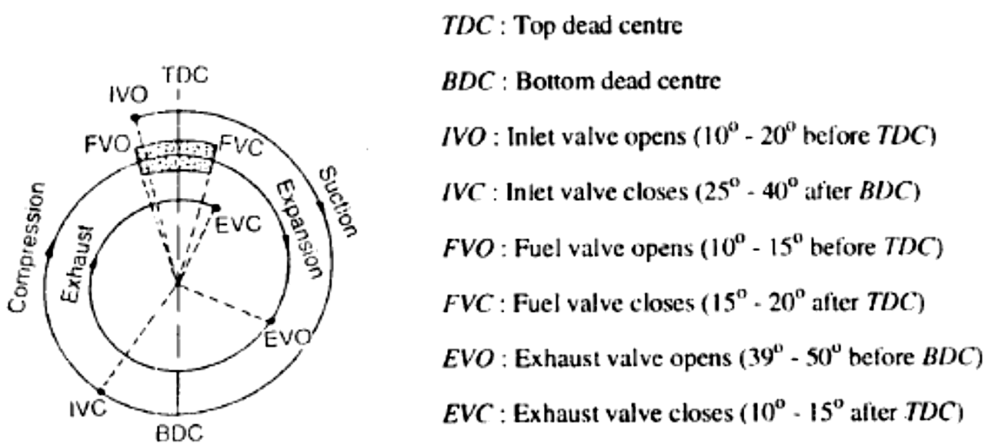


Fig. 2.8.Actual Valve timing diagram of 4-stroke Diesel Engine

This is done as the incoming air continues to flow into the cylinder although the piston is moving upwards from *BDC.* Now the air is compressed with both valves closed. Fuel valve opens a little before the piston reaches the *TDC.* Now the fuel is injected in the form of very fine spray, into the engine cylinder, which gets ignited due to high temperature of the compressed air. The fuel valve closes after the piston has come down a little from the *TDC.* This is done as the required quantity of fuel is injected into the engine cylinder. The burnt gases (under high pressure and temperature) push the piston downwards, and the expansion or working stroke takes place. Now the exhaust valve opens before the piston again reaches *BDC* and the burnt gases start leaving the engine cylinder. Now the piston reaches *BDC* and then starts moving up thus performing the exhaust stroke. The inlet valve opens before the piston reaches *TDC* to start suction stroke. This is done as the fresh air helps in pushing out the burnt gases. Now the piston again reaches *TDC,* and the suction starts. The exhaust valve closes when the crank has moved a little beyond the *TDC.* This is done as the burnt gases continue to leave the engine cylinder although the piston is moving downwards.

COMPARISON OF PETROL AND DIESEL ENGINES

Following points are important for the comparison of petrol engines and diesel engines:

Table 2.1. Comparison between Petrol and Diesel Engines

Petrol Engines	Diesel Engines
• A petrol engine draws a mixture of petrol and air during suction stroke. • The carburetor is employed to mix air and petrol in the required proportion and to supply it to the engine during suction stroke • Pressure at the end of compression is about 10 bar • The charge *(i.e.* petrol and air mixture) is ignited with the help of spark plug • The combustion of fuel takes place approximately at constant volume. In other words, it works on Otto cycle • A petrol engine has compression ratio approximately from 6 to 10. • The starting' is easy due to low compression ratio. • As the compression ratio is low, the petrol engines are lighter and cheaper. • The running cost of a petrol engine is high because of the higher cost of petrol. • The maintenance cost is less. • The thermal efficiency is up to about 26%. • Overheating trouble is more due to low thermal efficiency. • These are high speed engines. • The petrol engines arc generally employed in light duty vehicles such as scooters, motorcycles, cars. These are also used in aero planes	• A diesel engine draws only air during suction stroke • The injector or atomizer is employed to inject the fuel at the end of compression stroke. • Pressure at the end of compression is about 35 bar. • The fuel is injected in the form of fine spray. The temperature of the compressed air (about 600"C at a pressure of about *35*bar) is sufficiently high to ignite the fuel. • The combustion of fuel takes place approximately at constant pressure. In other words. It works on Diesel cycle. • A diesel engine has compression ratio approximately from 15 to 25. • The starting is little difficult due. to high compression ratio. • As the compression ratio is high. the diesel engine;; are heavier and costlier. • The running cost of diesel engine is low because of the lower cost of diesel. • The maintenance cost is more. • The thermal efficiency is up to about 40% • Overheating trouble is less due to high thermal efficiency • These are relatively low speed engines. • The diesel engines are generally employed in heavy duty vehicles like buses. trucks, and earth moving machines etc.

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TWO-STROKE CYCLE PETROL ENGINE

A two-stroke cycle petrol engine was devised by Duglad Clerk in I RHO. In this cycle, the suction, compression, expansion and exhaust takes place during two strokes of the piston. It means that there is one working stroke after every revolution of the crank shaft. A two stroke engine has ports instead of valves. All the four stages of a two stroke petrol engine are described below:

1. Suction stage: In this stage, the piston, while going down towards *BDC,* uncovers both the transfer port and the exhaust port The fresh fuel-air mixture flows into the engine cylinder from the crank case, as shown *(a).*

2. Compression stage: In this stage, the piston, while moving up, first covers the transfer port and then exhaust port. After that the fuel is compressed as the piston moves upwards as shown *(b).* In this stage, the inlet port opens and fresh fuel-air mixture enters into the crank case.

3. Expansion stage: Shortly before this piston reaches the *TDC* (during compression stroke), the charge is ignited with the help of a spark plug. It suddenly increases the pressure and temperature of the products of combustion. But the volume, practically, remains constant. Due to rise in the pressure, the piston is pushed downwards with a great force as shown in *(c).* The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy produced is transformed into mechanical work.

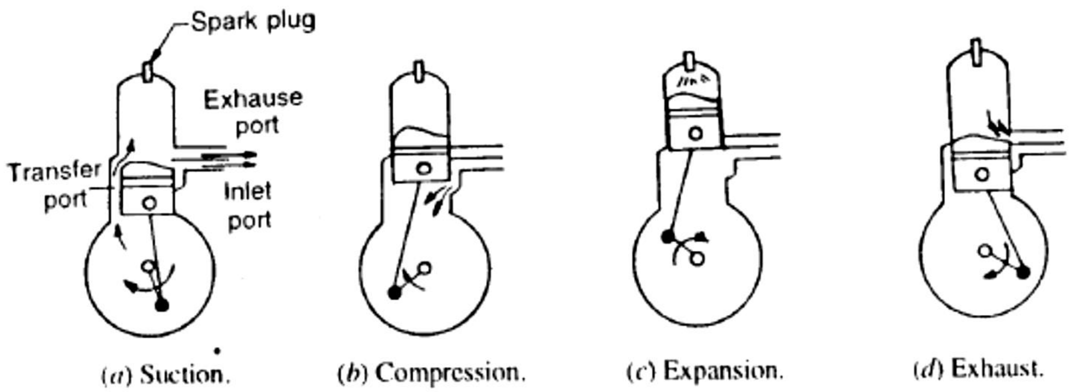


Fig. 2.9. Working of 2-Stroke Petrol Engine

4. Exhaust stage: In this stage, the exhaust port is opened as the piston moves downwards. The products of combustion, from the engine cylinder are exhausted through the exhaust port into the atmosphere, as shown *(d).* This completes the cycle and the engine cylinder is ready to suck the charge again

Actual Indicator Diagram for a Two Stroke Cycle Petrol Engine

The actual indicator diagram for a two-stroke cycle petrol engine is shown in suction is shown by the line 1-2-3, *i.e.* from the instant transfer port opens *(TPO)* and transfer port closes *(TPC).* We know that during the suction stage, the exhaust port is also open. In the first half of suction stage, the volume of fuel-air mixture and burnt gases increases. This happens as the piston moves from I to 2 *(i.e. BDC).* In the second half ofthe suction stage, the volume of charge and burnt gases decreases. This happens as the piston moves upwards from 2 to 3. A little beyond 3, the exhaust port closes *(EPC)* at 4. Now the charge inside the engine cylinder is compressed which is shown by the line 4-5.

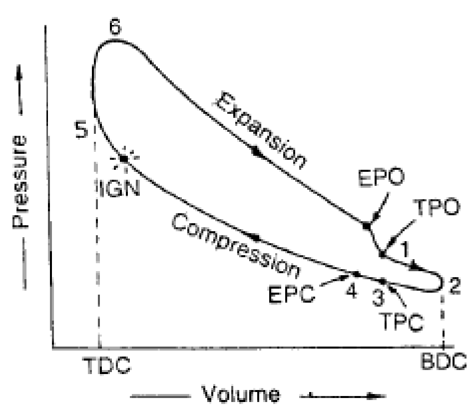


Fig.2.10. Actual p-V Diagram of 2-Stroke petrol engine

At the end of the compression, there is an increase in the pressure inside the engine cylinder. Shortly before the end of compression *(i.e. TDC)* the charge is ignited *(IGN)* with the help of spark plug. The sparking suddenly increases pressure and temperature of the products of combustion. But the volume, practically, remains constant as shown by the line 5-6. The expansion is shown by the line 6-7. Now the exhaust port opens *(EPO)* at 7, and the burnt gases are exhausted into the atmosphere through the exhaust port. It reduces the pressure. As the piston is moving towards *BDC*, therefore volume of burnt gases increases from 7 to 1. At 1, the transfer port opens *(TPO)* and the suction starts.

Valve Timing Diagram for a Two-Stroke Cycle Petrol Engine

In the valve timing diagram, as shown we see that the expansion of the charge (after ignition) starts as the piston moves from *TDC* towards *BDC*. First of all, the exhaust port opens

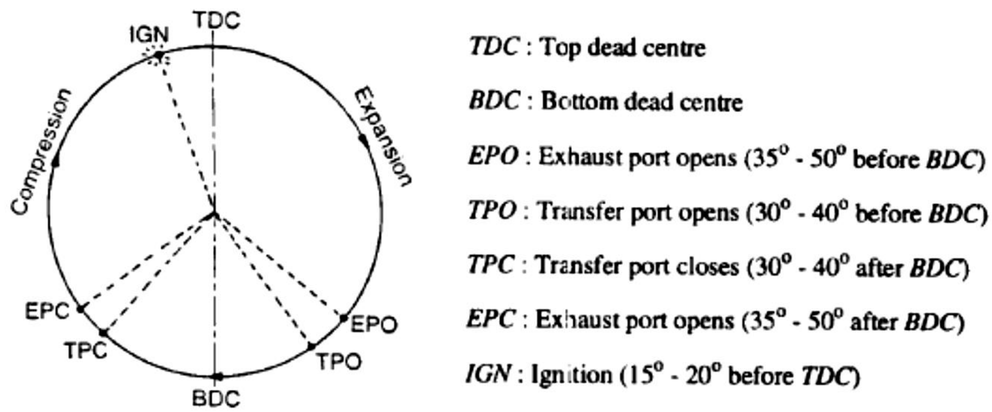


Fig.2.11. Actual Port timing Diagram of 2-Stroke petrol engine

before the piston reaches *BDC* and the burnt gases start leaving the cylinder. After a small fraction of the crank revolution, the transfer port also opens and the fresh fuel-air mixture enters into the engine cylinder. This is done as the fresh incoming charge helps in pushing out the burnt gases. Now the piston reaches *BDC* and then starts moving upwards. As the crank moves a little beyond *BDC*, first the transfer port closes and then the exhaust port also closes. This is done to suck fresh charge through the transfer port and to exhaust the burnt gases through the exhaust port simultaneously. Now the charge is compressed with both ports closed, and then ignited with the help of a spark plug before the end of compression stroke. This is done as the charge requires some time to ignite. By the time the piston reaches *TDC*, the burnt gases (under high pressure and temperature) push the piston downwards with full force and expansion of the burnt gases takes place. It may be noted that the exhaust and transfer ports open and close at equal angles on either side of the *BDC* position.

TWO-STROKE CYCLE DIESEL ENGINE

A two-stroke cycle diesel engine also has one working stroke after every revolution of the crank shaft. All the four stages of a two stroke cycle diesel engine are described below:

1. Suction stage: In this stage, the piston while going down towards *BDC* uncovers thetransfer port and the exhaust port. The fresh air flows into the engine cylinder from the crank case, as shown in *(a)*.

2. Compression stage: In this stage, the piston while moving up, first covers the transferport and then exhaust port. After that the air is compressed as the piston moves upwards as shown in *(b).* In this stage, the inlet port opens and the fresh air enters into the crank case.

3. Expansion stage: Shortly before the piston reaches the *TDC* (during compression stroke), the fuel oil is injected in the form of very fine spray into the engine cylinder through the nozzle known as fuel injection valve, as shown in *(c).* At this moment, temperature of the compressed air is sufficiently high to ignite the fuel. It suddenly increases the pressure and temperature of the products of combustion. The fuel oil is continuously injected for a fraction of the crank revolution. The fuel oil is assumed to be burnt at constant pressure. Due to increased pressure, the piston is pushed with a great force. The hot burnt gases expand due to high speed of the piston. During the expansion, some of the heat energy produced is transformed into mechanical work .

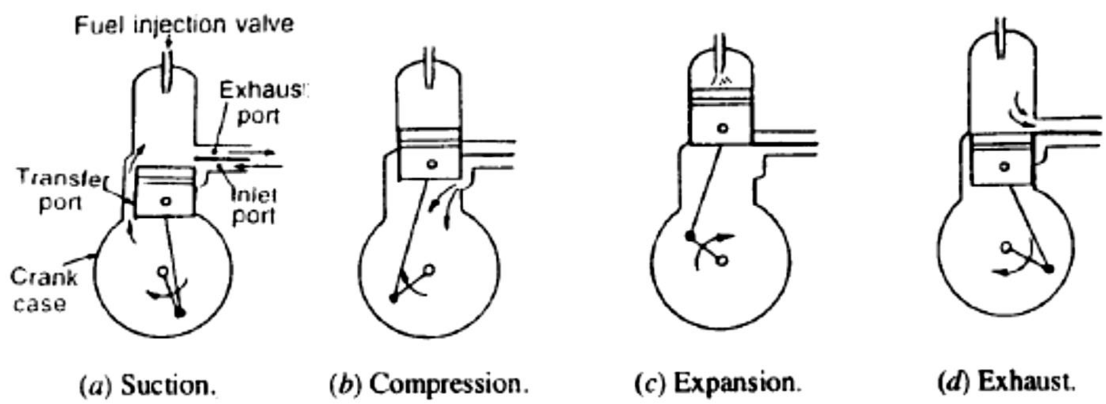


Fig.2.12. Working of 2-Stroke Diesel engine

4. Exhaust stage: In this stage, the exhaust port is opened and the piston moves downwards. The products of combustion from the engine cylinder are exhausted through the exhaust port into the atmosphere as shown in *(d).* This completes the cycle, and the engine cylinder is ready to suck the air again.

Actual Indicator Diagram for Two-Stroke Cycle Diesel Engine

The actual indicator diagram for a two-stroke cycle diesel engine is shown. The line shows the suction: 1-2-3 *i.e.* from the instant transfer port opens *(TPO)* and transfer portcloses *(TPC).* We know that during the suction stage, the exhaust port is also open. In the first half of suction stage, the volume: of air and burnt gases increases. This happens as the piston moves from *1-2 (i.e. BDC).* In the second half of the suction stage, the volume of air and burnt gases decreases. This happens as the piston moves upwards from 2-3. A little beyond 3, the exhaust port closes *(EPC)* at 4. Now the air inside the engine cylinder is compressed which is shown by the line 4-5. At the end of compression, there is an increase in the pressure inside the engine cylinder. Shortly before the end of compression *(i. e. TDC),* fuel valve opens *(FVO)* and the fuel is injected into the engine cylinder. The fuel is ignited by high temperature of the compressed air.

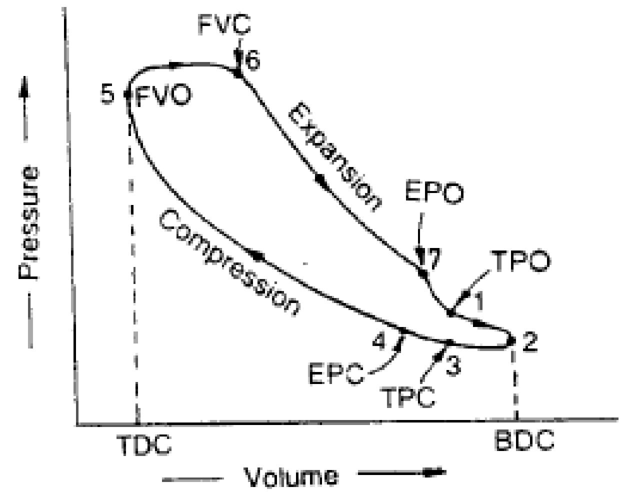
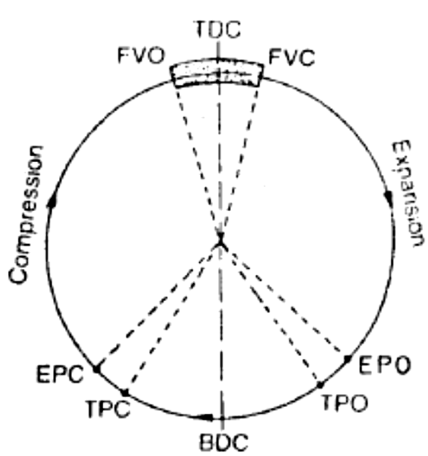


Fig.2.13. Actual p-V diagram of 2-Stroke Diesel engine

The ignition suddenly increases volume and temperature of the products of combustion. But the pressure, practically, remains constant as shown by the line 5-6. The line 6-7 shows the expansion, Now the exhaust port opens *(EPO)* at 7 and the burnt gases are exhausted into the atmosphere through the exhaust port. It reduces the pressure. As the piston is moving towards *BDC,* therefore volume of burnt gases increases from 7 to 1. At 1, the transfer port opens *(TPO)* and the suction starts.

Valve Timing Diagram for a Two-Stroke Cycle Diesel Engine

In the valve-timing diagram, as shown, we see that the expansion of the charge(after ignition) starts as the piston moves from *TDC* towards *BDC.* First of all, the exhaust port opens before the piston reaches *BDC* and the burnt gases start leaving the cylinder. After a small fraction of the crank revolution, the transfer port also opens and the fresh air enters into the engine cylinder. This is done as the fresh incoming air helps in pushing out the burnt gases. Now the piston reaches *BDC* and then starts moving upwards. As the crank moves a little beyond *BDC,* first the transfer port closes and then the exhaust port also closes. This is done to suck fresh air through the transfer port and to exhaust the burnt gases through the exhaust port simultaneously. Now the charge is compressed



TDC : Top dead centre

BDC Bottom dead centre

FVO : Fuel valve opens (10° - 15° before *TDC*)

FVC : Fuel valve closes (15° - 20° after *TDC*)

EPO : Exhaust port opens (35° - 50° before *BDC*)

TPO : Transfer port opens (30° - 40° before *BDC*)

TPC : Transfer port closes (30° - 40° after *BDC*)

EPC : Exhaust port closes (35° - 50° after *BDC*)

Fig.2.14. Actual Port timing diagramof 2-Stroke Diesel engine

with both the ports closed. Fuel valve opens a little before the piston reaches the *TDC*. Now the fuel is injected in the form of very fine spray into the engine cylinder, which gets ignited due to high temperature of the compressed air. The fuel valve closes after the piston has come down a little from the *TDC*. 'This is done as the required quantity of fuel is injected into the engine cylinder. Now the burnt gases (under high pressure and temperature) push the piston downwards with full force and expansion of the gases takes place. It may be noted that in a two-stroke cycle diesel engine, like two-stroke petrol engine, the exhaust and transfer ports open and close at equal angles on either side of the *BDC* position.

SCAVENGING OF I.C. ENGINES

We have already discussed the sequence of operations in a cycle of an I.C. engine. The last stroke of an IC engine is the exhaust, which means the removal of burnt gases from the engine cylinder. It has been experienced that the burnt gases in the engine cylinder are not completely exhausted before the suction stroke. But a part of the gases still remain inside the cylinder and mix with the fresh charge. As a result of this mixing, the fresh charge gets diluted and its strength is reduced. The scientists and engineers, all over the world, have concentrated on the design of their IC engines so that the burnt gases are completely exhausted from the engine cylinder before the suction starts. The process of removing burnt gases, from the combustion chamber of the engine cylinder, is known a scavenging. Now we shall discuss the scavenging in four-stroke and two-stroke
cycle engines.

1. Four-stroke cycle engines: In a four stroke cycle engine, the scavenging is very effective as the piston during the exhaust stroke pushes out the burnt gases from the engine cylinder. It may be noted that a small quantity of burnt gases remain in the engine cylinder in the clearance space.
2. Two-stroke cycle engine: In a two-stroke cycle engine, the scavenging is less effective as the exhaust port is open for a small fraction of the crank revolution. Moreover, as the transfer and exhaust port arc open simultaneously during a part of the crank revolution, therefore fresh charge also escapes out along with the burnt gases. This difficulty is overcome by designing the piston crown of a particular shape .

TYPES OF SCAVENGING

Though there are many types of scavenging, yet the following are important from the subject point of view:
1. Cross flow scavenging: In this method, the transfer port (or inlet port for the engine
cylinder) and exhaust port are situated on the opposite sides of the engine cylinder (as is done in case of two-stroke cycle engines). The piston crown is designed into a particular shape, so that the fresh charge moves upwards and pushes out the burnt gases in the form of cross flow as. Shown in (a).
2. Backflow or loop scavenging: In this method, the inlet and outlet ports are situated on
the same side of the engine cylinder. The fresh charge, while entering into the engine cylinder, forms a loop and pushes out the burnt gases as shown in *(b)*.

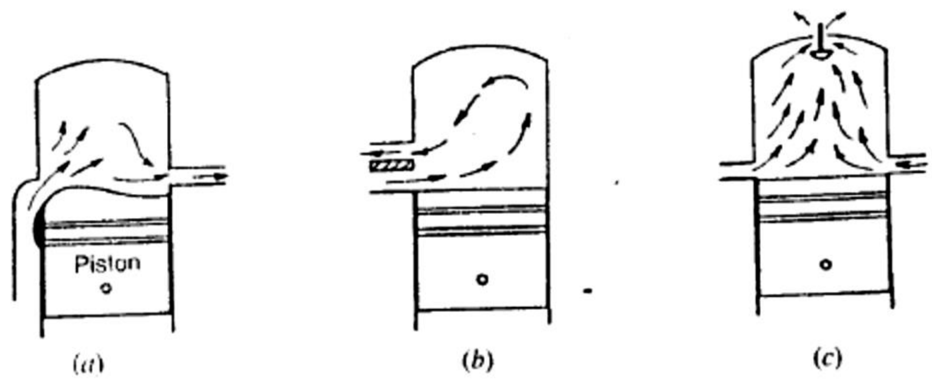


Fig.2.15. Types of Scavenging

3. Uniflow scavenging: In this method, the fresh charge, while entering from one side (qrsometimes two sides) of the engine cylinder pushes out the gases through the exit valve situated on the top of the cylinder. In uniflow scavenging, both the fresh charge and burnt gases move in the same upward direction as shown in *(c).*

DETONATION IN IC ENGINES

The loud pulsating noise heard within the engine cylinder is known as *detonation* (also called *knocking* or *pinking).* It is caused due to the propagation of a high speed pressure wave created by the auto-ignition of end portion of unburnt fuel. The blow of this pressure wave may be of sufficient strength to break the piston. Thus, the detonation is harmful to the engine and must be avoided. The following are certain factors which causes detonation:

- The shape of the combustion chamber,
- The relative position of the sparking plugs in case of petrol engines,
- The chemical nature of the fuel,
- The initial temperature and pressure of the fuel,
- The rate of combustion of that portion of the fuel which is the first to ignite. This portion of the fuel in heating up, compresses the remaining unbumt fuel, thus producing the conditions for auto-ignition to occur.

The detonation in petrol engines can be suppressed or reduced by the addition of a small amount of lead ethide or ethyl fluid to the fuel. This is called *doping.*

The following are the chief effects due to detonation:

- A loud pulsating noise which may be accompanied by a vibration of the engine.
- An increase in the heat lost to the surface of combustion chamber.
- An increase in carbon deposits.

RATING OF SI ENGINE FUELS OCTANE NUMBER

The hydrocarbon fuels used in spark ignition (S.I.) engine have a tendency to cause engine knock when the engine operating conditions become severe. The knocking tendency of a fuel in S.I. engines is generally expressed by its *octane number.* The percentage, by volume, of iso-octane in a mixture of iso-octane normal heptane ,which exactly matches the knocking intensity of a given fuel, in a standard engine, under given standard operating conditions, is termed as the *octane* *number rating* of that fuel. Thus, if a mixture of 50 percent iso-octane and 50 percent normal heptane matches the fuel under test, then this fuel is assigned an octane number rating of 50. If a fuel matches in knocking intensity a mixture of 75 percent iso-octane and 25 percent normal heptane, then this fuel would be assigned an octane number rating of 75. This octane number rating is an expression which indicates the ability of a fuel to resist knock in a S.1. engine. Since iso-octane is a very good anti-knock fuel, therefore it is assigned a rating of 100 octane number. On the other hand, normal heptane has very poor anti-knock qualities, therefore it is given a rating of 0 (zero) octane number. These two fuels, *i.e.* iso-octane and normal heptane are known as primary reference fuels. It may be noted that higher the octane number rating of a fuel, the greater will be its resistance to knock and the higher will be the· compression ratio. Since the power output and specific fuel consumption are functions of compression ratio, therefore we may say that these are also functions of octane number rating. This fact indicates the extreme importance of the octane number rating in fuels for S.I. engines.

RATING OF CI ENGINE FUELS CETANE NUMBER

The knocking tendency is also found in compression ignition (C.I.) engines with an effect similar to that of S.1. engines, but it is due to a different phenomenon. The knock in C.I. engines is due to sudden ignition and abnormally rapid combustion of accumulated fuel in the combustion chamber Such a situation occurs Because of an Ignition lag in the combustion of the fuel between the time of injection and the actual burning. The property of ignition lag is generally measured in terms of *cetane number*. It is defined as the percentage, by volume, of cetane in a mixture of cetane and alpha-methyl-naphthalene that produces the same ignition lag as the fuel being tested in the same engine and under the same operating conditions. For example, a fuel of cetane number 50 has the same ignition quality as a mixture of 50 percent cetane and 50 percent alpha-methyl-naphthalene. The cetane which is a straight chain paraffin with good ignition quality is assigned a cetane number of 100 and alpha-methyl-naphthalene which is a hydrocarbon with poor ignition quality, is assigned a 0 (zero) cetane number.

IGNITION SYSTEMS OF PETROL ENGINES

We have already discussed that the ignition in a petrol engine, takes place by means of a spark plug at the end of the compression stroke. The voltage required to produce a spark across the gap between the sparking points of a plug, is about 8000 volts. Thus, the ignition system in a petrol engine has to transform the normal battery voltage (6 to 12 volts) to 8000 volts. In addition to this, the ignition system has to provide spark in each cylinder at the appropriate time. Following two ignition systems of petrol engines are important from the subject point of view:
1. Coil ignition system 2. Magneto ignition system.

COIL IGNITION SYSTEM

It is also known as *battery ignition system,* and has an induction coil, which consists of two coils known as primary and secondary coils wound on a soft iron core, as shown . The primary coil consists of a few hundred turns (about 300 turns) of wire. Over this coil, but insulated from it, are wound several thousand turns (about 20,000 turns) of secondary coil. The one end of the primary coil is connected to a ignition switch, ammeter and battery generally of 6 volts. The other end of the primary coil is connected to a condenser and a contact breaker.

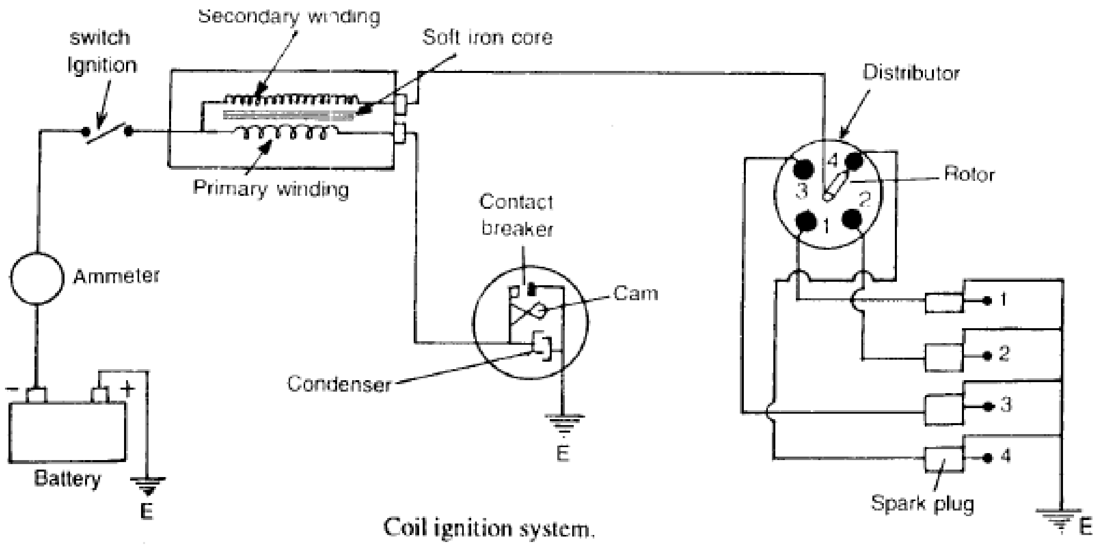


Fig.2.16. Coil Ignition System

A condenser is connected across the contact-breaker for the following two reasons:

- It prevents sparking across the gap between the points,
- It causes a more rapid break of the primary current, giving a higher voltage in the secondary circuit.

The secondary coil is connected to a distributor (in a multi-cylinder engine) with the central terminal of the sparking plugs. The outer terminals of the sparking plugs are earthed together, and connected to the body of the engine. When the current flows through the primary coil, it sets up a magnetic field which surrounds both the primary and secondary coils. As the switch is on, the contact-breaker connects the two ends. The magnetic field in coils has tendency to grow from zero to maximum value. Due to this change in the magnetic field, a voltage is generated in both the coils, but opposite to the applied voltage (of battery). Thus the primary coil does not give the final value. The voltage in the secondary coil is, therefore, not sufficient to overcome the resistance of the air gap of the sparking plug, hence no spark occurs. When the current in the primary coil is switched off by the moving cam, the magneticfield generated around the coil collapses immediately. The sudden variation of flux, which takes place, gives rise to the voltage generated in each coil. The value of the voltage depends upon the number of turns in each coil. As a matter of fact, the voltage required to produce a spark across the gap, between the sparking points, is between 10 000 to 20 000 volts. Since the secondary coil has several thousand turns, so it develops a sufficient high voltage to overcome the resistance of the gap of the sparking plug. This high voltage then passes to a distributor. It connects the sparking plugs in rotation depending upon the firing order of the engine. Hence, the ignition of fuel takes place in all the engine cylinders. The coil ignition system is employed in medium and heavy spark ignition engines such as in cars.

MAGNETO IGNITION SYSTEM

The magneto ignition system as shown has the same principle of working as that of coil ignition system; except that no battery is required as the magneto acts its own generator. It consists of either rotating magnets in fixed coils, or rotating coils in fixed magnets. The current produced by the magneto is made to flow to the induction coil which works in the same way as that of coil ignition system. The high voltage current is then made to flow to the distributor, which connects the sparking plugs in rotation depending upon the firing order of the engine. This type of ignition system is generally employed in small spark ignition engines such as scooters, motor cycles and small motor boat engines.

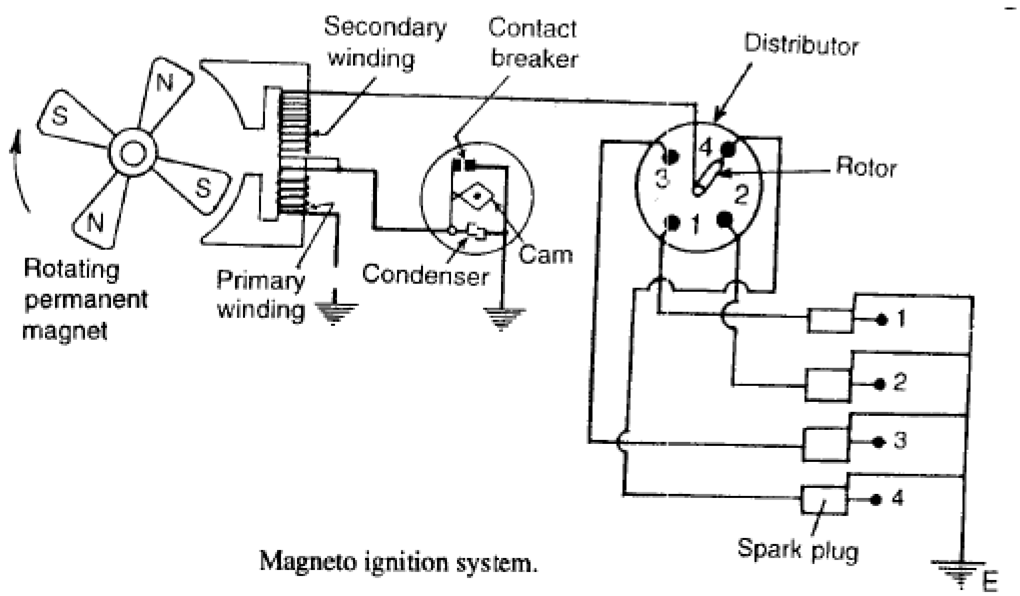


Fig.2.17. Magneto Ignition System

FUEL INJECTION SYSTEM FOR DIESEL ENGINES

The fol1owing two methods of fuel injection system are generally employed with diesel engines *(i.e.compression* ignition engines) :

1. Air injection method, and
2. Airless or solid injection method.

These methods are discussed, in detail, as follows:

1. Air injection method: In this method of fuel injection, a blast of compressed air is used to inject the fuel into the engine cylinder. This method requires the aid of an air compressor which is driven by the engine crankshaft. The air is compressed at a pressure higher than that of engine cylinder at the end of its compression stroke. This method is not used now-a-days because of complicated and expensive system.

2. Airless or solid injection method: The most modern compression ignition engines use, now-a-days, the solid injection system. In this method, a separate fuel pump driven by the main crankshaft is used for forcing the fuel. The fuel is compressed in this pump to a pressure higher than that of engine cylinder at the end of compression. This fuel under pressure is directly sprayed into the combustion chamber of the engine cylinder at the end of compression stroke, with the help of an injector. The solid injection method may be further divided into the following two commonly used systems.

- Common rail system
- Individual pump system

Common rail system: In the common rail system as shown a multi cylinder high pressure pump is used to supply the fuel at a high pressure to a common rail or header.

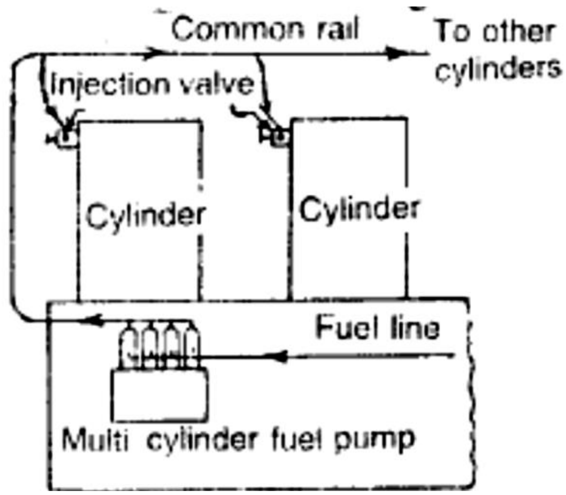


Fig.2.18. Common Rail Injection System

The high pressure in the common rail forces the fuel to each of the nozzle located in the cylinders. The pressure in this common rail is kept constant with the help of a high pressure relief valve. A metered quantity of fuel is supplied to each cylinder through the nozzle by operating the respective fuel injection valve with the help of cam mechanism driven by the crankshaft of the engine.

Individual pump system: In the individual pump system, as shown each cylinder of the engine is provided with an individual injection valve, a high pressure pump and a metering device run by the crankshaft of the engine. The high pressure pump plunger is actuated by a cam and produces the fuel pressure necessary to open the injection valve at the correct time. The amount of fuel injected depends upon the effective stroke of the plunger.

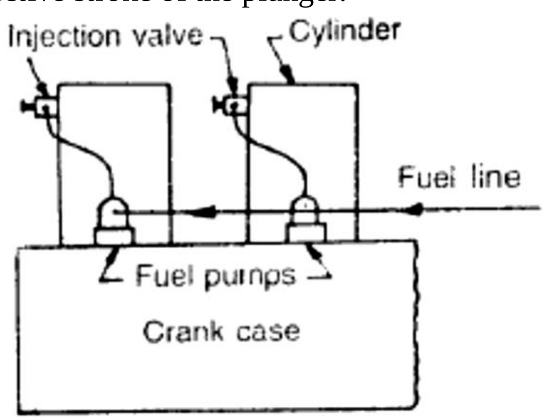


Fig.2.19. Individual Pump System

COOLING OF I.C. ENGINES

We have already discussed that due to combustion of fuel inside the engine cylinder of I.C. engines, intense heat is generated. It has been experimentally found that about 30% of the heat generated is converted into mechanical work. Out of the remaining heat (about 70%) about 40% is carried away by the exhaust gases into the atmosphere. The remaining part of the heat (about

30%), if left un-attended, will be absorbed by engine cylinder, cylinder head piston, and engine valves etc. It has also been found that the overheating of these parts causes the following effects:

- The overheating causes thermal stresses in the engine parts, which may lead to their distortion.
- The overheating reduces strength aft he piston. The overheating may cause even seizure of the piston.
- The overheating causes decomposition of the lubricating oil, which may cause carbon deposit on the engine and piston head.
- The over heating, causes burning of valves and valve seats.
- The overheating reduces volumetric efficiency of the engine.
- The overheating increases tendency of the detonation.

In other to avoid the adverse effects of overheating, it is very essential to provide some cooling system for an I.C. engine. In general, the cooling system provided should have the following two characteristics for its efficient working:

- It should be capable of removing about 30% of that total heat generated in the combustion chamber. It has been experienced that removal of more than *30%* of heat generated reduces thermal efficiency of the engine. Similarly, removal of less than 30% of the heat generated will have some adverse effects as mentioned above.
- It should be capable of removing heat at a fast rate, when the engine is hot. But at the time of starting the engine, the cooling should be comparatively slow, so that the various components of the engine attain their working temperature in a short time.

COOLING SYSTEMS FOR I.C. ENGINES

We have already discussed, in the last article, the adverse effects of overheating of an I.C. engine and characteristics of the cooling system adopted. The following two systems are used for cooling the I.C. engines these days:

1. Air cooling system: The air-cooling system, as shown, is used in the engines of motor cycles, scooters, aeroplanes and other stationary installations. In countries with cold climate, this system is also used in car engines. In this system, the heat is dissipated directly to the atmospheric air by conduction through the cylinder walls. In order to increase the rate of cooling, the outer surface area of the cylinder and cylinder head is increased by providing radiating fins and flanges. In bigger units, fans are provided to circulate the air around the cylinder walls and cylinder head.

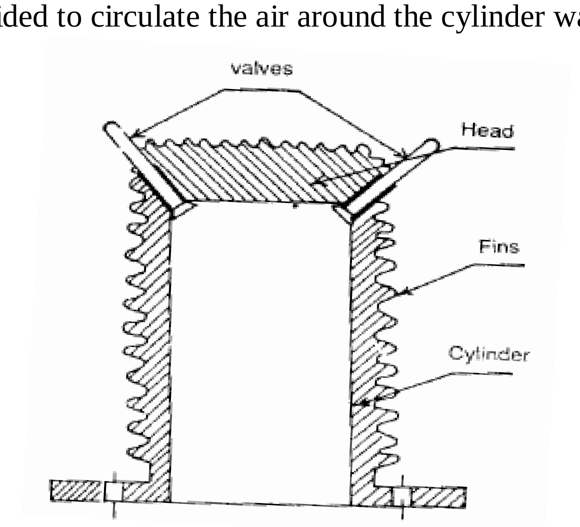


Fig. 2.20. Air Cooling System

2. Water cooling system (Thermo-syphon system of cooling). The water cooling system as shown , is used in the engines of cars, buses, trucks etc. In this system, the water is circulated through water jackets around each of the combustion chambers, cylinders, valve seats and valve stems. The water is kept continuously in motion by a centrifugal water pump which is driven by a V-belt from the pulley on the engine crank shaft. After passing through the engine jackets in the cylinder block and heads, the water is passed through the radiator. In the radiator, the water is cooled by air drawn through the radiator by a fan. Usually, fan and water pump are mounted and driven on a common shaft. After passing through the radiator, the water is drained and delivered to the water pump through a cylinder inlet passage. The water is again circulated through the engine jackets.

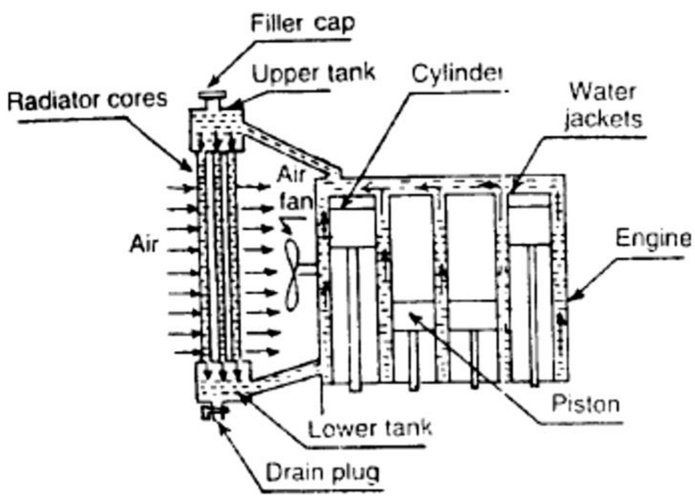


Fig.2.21. Water Cooling System

COMPARISON OF AIR COOLING AND WATER COOLING SYSTEMS

The following points are important for the comparison of air-cooling and water-cooling systems.

Table 2.2. Comparison of Air Cooling and Water Cooling Systems

Air Cooling System	**Water Cooling System**
• The design of this system is simple and less costly	• The design of this system is complicated and more costly
• The mass of cooling system is very less	• The mass of cooling system is much more
• The fuel consumption is more	• The fuel consumption is less
• Its installation and maintenance is very easy and less costly	• Its installation and maintenance is difficult and more costly
• There is no danger of leakage or freezing of the coolant	• There is a danger of leakage or freezing of the coolant
• It works smoothly and continuously. Moreover it does not depend on any coolant	• If the system fails, it may cause serious damage to the engine within a short time.

SUPERCHARGING OF IC ENGINES

It is the process of increasing the mass, or in other words density, of the air-fuel mixture (in spark ignition engine) or air (in compression ignition *i.e.* diesel engines) induced into the engine cylinder. This is, usually, done with the help of compressor or blower known as supercharger. It has been experimentally found that the supercharging increases the power developed by the engine. It is widely used in aircraft engines, as the mass of air, sucked in the engine cylinder, decreases at very high altitudes. This happens, because atmospheric pressure decreases with the increase in altitude. Now-a-days, supercharging is also used in two-stroke and four-stroke petrol and diesel engines. It will be interesting to know that a supercharged engine is lighter, requires smaller foundations and consumes less lubricating oil as compared to an ordinary engine. Following are the

Objects of supercharging the engines.

1. To reduce mass of the engine per brake power (as required in aircraft engines).
2. To maintain power of aircraft engines at high altitudes where less oxygen is available for combustion.
3. To reduce space occupied by the engine (as required in marine engines).
4. To reduce the consumption of lubricating oil (as required in all type of engines).
5. To increase the power output of an engine when greater power is required (as required in racing cars and other engines).

Methods of Supercharging

Strictly speaking, a supercharger is an air pump, which receives air from the' atmosphere surrounding the engine, compresses it to a higher pressure and then feeds it into the inlet valve of the engine.

Following two method; of supercharging are important from the subject point of view:

1. Reciprocating type:It has a piston which moves to and fro inside a cylinder. It is an old method and is not encouraged these days, as it occupies a large space and has lubrication problem.

2. Rotary type:It resembles a centrifugal pump i 1its outward appearance, but differs in action. There are many types of rotary pumps, but gear type, lobe type and vane type are commonly used.

LUBRICATION OF I.C. ENGINES

As a matter of fact, the moving parts of an I.C engine are likely to wear off due to continuous rubbing action of one part with another. In order to avoid an early wearing of the engine parts, a proper lubrication arrangement is provided in I.C. engines.

In general, following are the main advantages of lubrication of I.C. engines:

1. It reduces wear and tear of the moving parts.
2. It damps down the vibrations of the engine.
3. It dissipates the heat generated from the moving parts due to friction.
4. It cleans the moving parts.
5. It makes the piston gas-tight.

Lubrication System for IC Engines

The following two lubrication systems of I.C. engines are important from the subject point of view:

1. Splash lubrication: This method is generally employed for lubricating small I.C. engines. In this method, an oil sump is fixed to the bottom of the crank case and the pump is immersed, in the lubricating oil, as shown A small hole s drilled in the crank shaft and the oil is forced through this hole to the bearing. The oil is also forced along the connecting rod either through a hole drilled in the rod or along a small copper pipe to the gudgeon pin and piston.

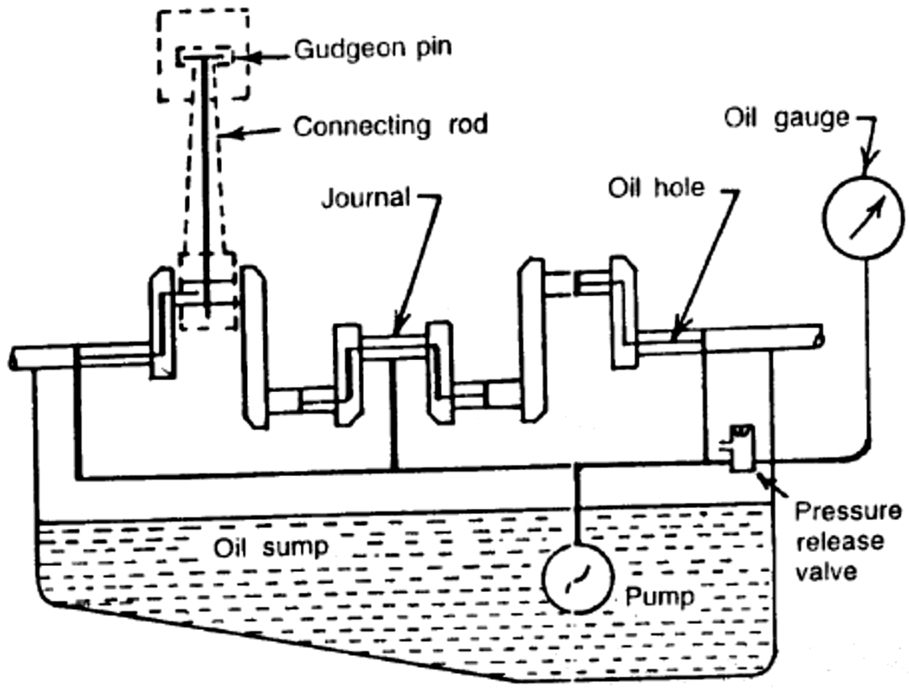


Fig. 2.22. Splash lubrication

Surplus oil lubricates the cams, tappets and valve stems. The whole oil is drained back into the sump.

2. Forced lubrication: In this method, the lubricating oil is carried in a separate tank andis pumped at a high pressure to the main bearings. It passes at a lower pressure to the camshaft and timing gears, as the oil drains with the sump it is pumped back by a pump known as scavenge pump through an oil cooler to the oil tank.

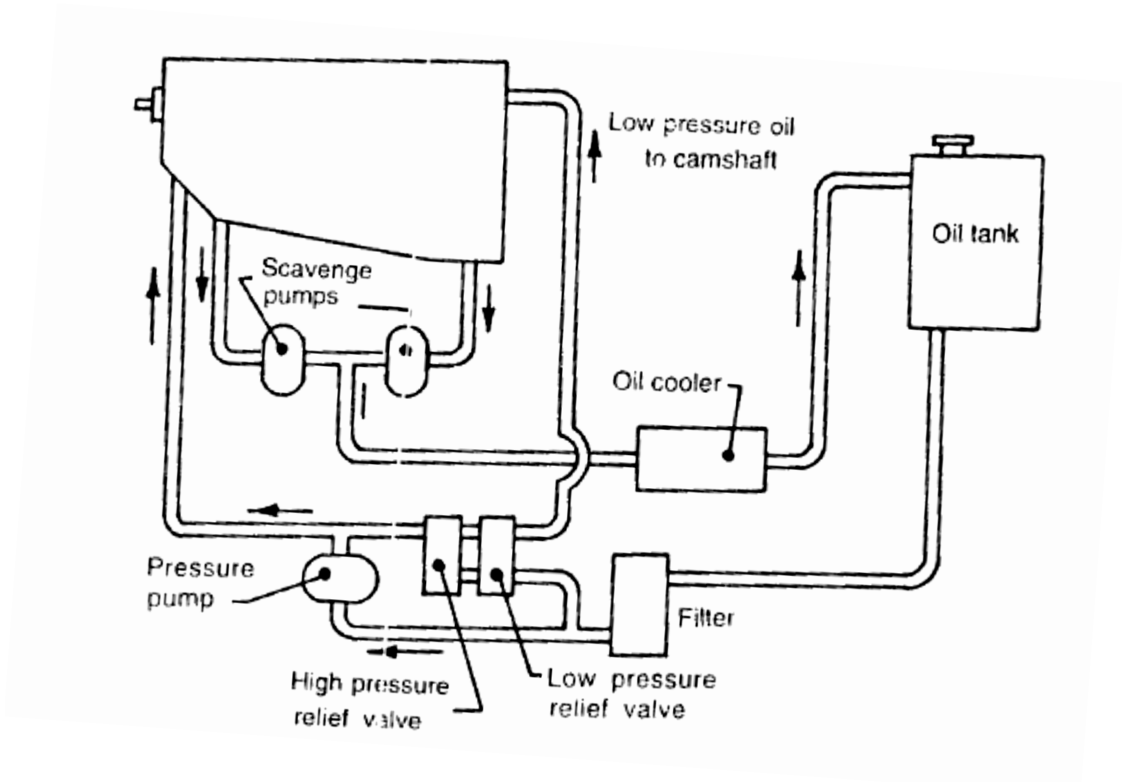


Fig. 2.23. Forced lubrication

GOVERNING OF I.C. ENGINES

As a matter of fact, all the I.C. engines like other engines, are always designed to run at a particular speed. But in actual practice, load on the engine keeps on fluctuating from time to time. A little consideration will show, that change of load, on an I.C. engine, is sure to change its speed. It has been observed that if load on an I.C. engine is decreased without changing the quantity of fuel, the engine will run at a higher speed. Similarly, if load on the engine is increased without changing the quantity of fuel, the engine will run at a lower speed. Now, in order to have a high efficiency of an I.C. engine, at different load conditions, its speed must be kept constant as far as possible. The process of providing any arrangement, which will keep the speed constant (according to the changing load conditions) is known as *governing of I.C. engines.*

Methods of Governing I.C.Engines

Through there are many methods for the governing of I.C. engines, yet the following are important from the subject point of view:

I. Hit and miss governing: This method of governing is widely used for IC engines of smaller capacity or gas engines This method is most suitable for engines, which are frequently subjected to reduced loads and ,LS a result of this, the engines tend to run at higher speeds. In this system of governing, whenever the engine starts running at higher speed (due to decreased load), some explosion are omitted or missed. This is done with help of centrifugal governor in which the inlet valve of fuel is closed and the explosions are omitted till the engine speed reaches its normal value. The only disadvantage of this method is that there is uneven turning moment due to missing of explosions. As a result of this, it requires a heavy flywheel.

2. Qualitative governing: In this system of governing, a control valve is fitted in the fueldelivery pipe, which controls the quantity of fuel to be mixed in the charge. The movement of control valve is regulated by the centrifugal governor through rack and pinion arrangement. It may be noted that in this system, the amount of air used in each cycle remains the same. But with the change in the quantity of fuel (with quantity of air remaining constant), the quality of charge *(i. e.* air-fuel ratio of mixture) changes. Whenever the engine starts running at higher speed (due to decreased load), the quantity of fuel is reduced till the engine speed reaches its normal value. Similarly, whenever theengine starts running at lower speed (due to increased load), the quantity of fuel is increased. In automobile engines, the rack and pinion arrangement is connected with the accelerator.

3. Quantitative governing*:* In this system of governing, the quality of charge *(i.e.* air-fuel ratio of the mixture) is kept constant. But quantity of mixture supplied to the engine cylinder is varied by means of a throttle valve which is regulated by the centrifugal governor through rack and pinion arrangement. Whenever the engine starts running at higher speed (cue to decreased load), the quantity of charge is reduced till the engine speed reaches its normal value. Similarly, whenever the engine starts running at lower speed (due to increased load), the quantity of charge is increased. This method is used for governing large engines.

4. Combination system of governing: In this system of governing, the above mentioned two methods of governing *(i.e.* qualitative and quantitative) are combined together, so that quality as well as quantity of the charge is varied according to the changing conditions. This system is complicated, and has not proved to be successful.

CARBURETTOR

The carburettor is a device for atomising and vaporing the fuel and mixing it with the air in the varying proportions to suit the changing operating conditions of the engine. The process of breaking up and mixing the fuel with the air is called *carburation*. There are many types of the carburettors in use, but the simplest form of the carburettor is shown. It consists of a fuel jet located in the centre of the choke tube. A float chamber is provided for maintaining the level of the fuel in the jet and is controlled by a float and lever, which operates its needle valve.

The fuel is pumped into the float chamber and when the correct level of the fuel is reached, the float closes the needle valve, and shuts off the petrol supply. The suction produced by the engine draws air through t:le choke tube. The reduced diameter of the choke tube increases the velocity of air and reduces the pressure. The high velocity and low pressure in the tube facilitates the breaking up of fuel and its admixture with the air. A throttle valve controls the flow of the mixture delivered to the engine cylinder.

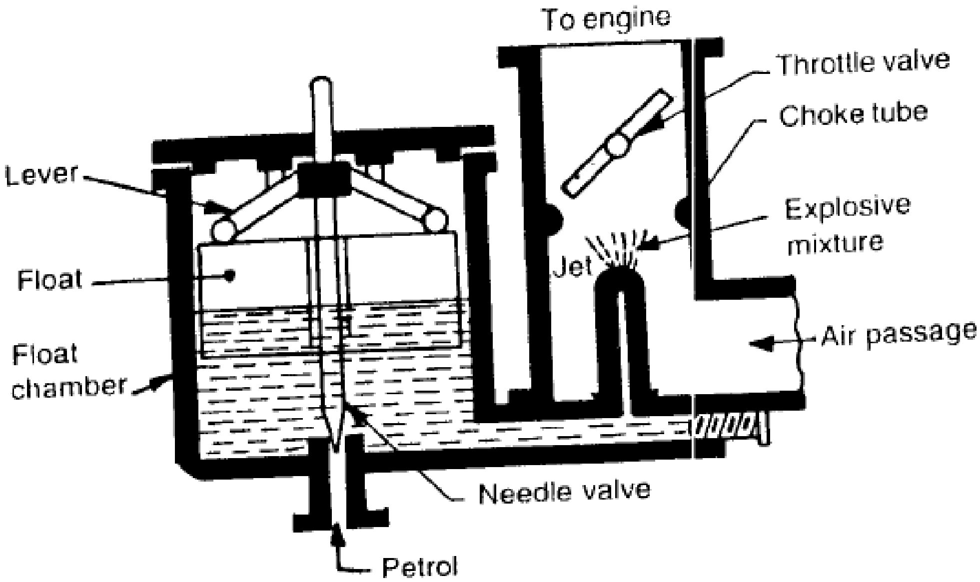


Fig. 2.24. Carburetor

SPARK PLUG

It is always screwed into the cylinder head for igniting the charge of petrol engines: It is, usually, designed to withstand a pressure up to 35 bar and operate under a current of 10000 to 30000 volts. Terminal A spark plug c01sists of central porcelain insulator, containing an axial electrode length wise and ground electrode welded to it. The central electrode have an external contact at the top, which is connected to the terminal and communicates with the distributor. A metal tongue is welded to the ground electrode. which bends over to lie across the end of the central electrode. There is a small gap known as spark gap between the end of the central electrode and the metal tongue, as shown. The high tension electric spark jumps over the gap to ignite the charge in the
engine cylinder. The electrode material should be such which can withstand corrosiveness, high temperature having good thermal conductivity. The electrodes are generally made from the alloys of platinum, nickle, chromium, barium etc.

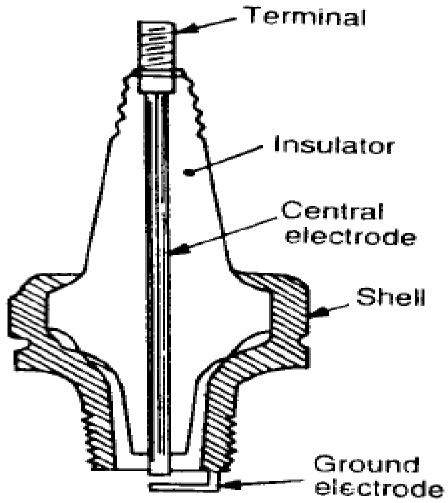


Fig. 2.25. Spark Plug

FUEL PUMP

The main object of a fuel pump in a diesel engine is to deliver a fuel to the injector which sprays the finely divided particles of the fuel suitable for rapid combustion.

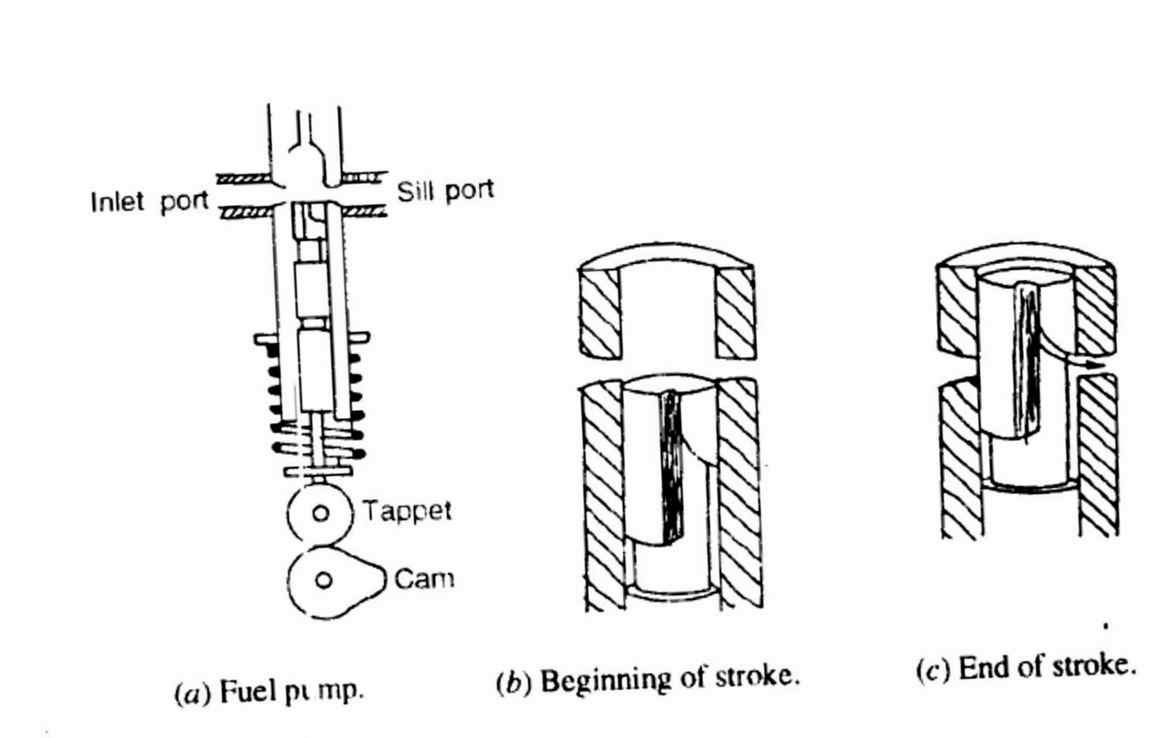


Fig. 2.26. Fuel Pump

The simplified sketch of a fuel pump is shown. It consist of a plunger which moves up and down in the barrel by the cam and spring arrangement provided for pushing and lowering the plunger respectively. The fuel oil is highly filtered by means of felt-pack filter before entering the barrel of the pump. The upper end part of the plunger is cut away in a helix shaped piece forming a groove between the plunger and barrel, which is the most important one. Therefore, the amount of fuel delivered and injected into the engine cylinder depends upon the rotary position of the plunger in the barrel. Figure *(b)* and *(c)* shows how the top part of the plunger is designed so that the correct amount of fuel is delivered to the injector. When the plunger is at the bottom of its stroke as shown ill Figure *(b)*, the fuel enters the barrel through the inlet port. As the plunger rises, it forces this fuel up into the injector, until the upper part cut away comes opposite the sill port. Then the fuel escapes down the groove and out through the sill port so that injection ceases, as shown in Figure *(c)*. The plunger can be made to rotate in the barrel and therefore more fuel is injected. When the plunger is rotated so that the groove is opposite to the sill port, no fuel at all is injected and thus the engine stops

INJECTOR OR ATOMISER

The injector or atomiser is also an important part of the diesel engine which breaks up the fuel and sprays into the cylinder into a very fine divided particles. Figure shows the type of an injector in which fuel is delivered from the pump along the horizontal pipe connected at *A*. The vertical spindle of the injector is spring loaded at the top which holds the spindle down with a pressure of 140 bar so that the fuel pressure must reach this value before the nozzle will lift to allow fuel to be injected into the engine cylinder. The fuel which leaks past the vertical spindle is taken off by means of an outlet pipe fitted at *B* above the fuel inlet pipe

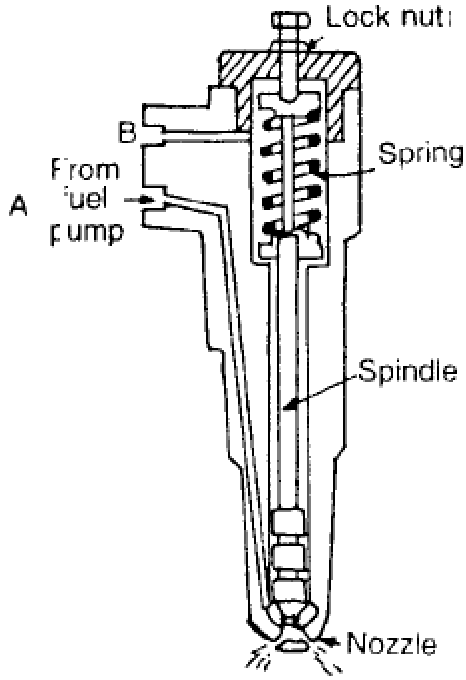


Fig. 2.27. Fuel Injector

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 following : (a) Power and Mechanical Efficiency. (b) Mean Effective Pressure and Torque. I 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

$$BP=2\Pi NT/60$$

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.

$$I.P = PmLANK/60$$

Where,
Pm = Mean effective pressure, N/m^2, L = Length of the stroke, m, A = Area of the piston, m^2, 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=BP/IP

The difference between ip and bp is called friction power (FP). FP = IP − Bp Mechanical efficiency= 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,

$$Pm = 60X I.P/LANk$$

where, Pm = Mean effective pressure, N/m^2, IP = Indicated power, Watt, L = Length of the stroke, m, A = Area of the piston, m^2, 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 BP it is called the brake mean effective pressure(Pm), and if based on IP it is called indicated mean effective pressure (imep). Similarly, the friction mean effective pressure (fmep) can be defined as,

$$fmep = imep – bmep$$

The torque is related to mean effective pressure by the relation

$$B.P = 2\Pi nT/60$$

$$I.P = PmLANk/60$$

$$2\Pi nT/60 = [bmep.A.L.(Nk/60)]$$

or, $$T = (bmep.A.L.k)/2\pi$$

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,
Specific output=B.P/A.L

Constant = bmep × rpm •

The specific output consists of two elements – the bmep (force) available to work and the speed with which it is working. • Therefore, for the same piston displacement and bmep 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 bmep. Increasing speed involves increase in the mechanical stress of various engine parts whereas increasing bmep requires better heat release and more load on engine cylinder.

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, =(Actual Fuel- Air ratio)/(Stoichiometric fuel-Air ratio)

Indicated Specific Fuel Consumption*:* This is defined as the mass of fuel consumption per hour in order to produce an indicated power of one kilo watt.

$$\text{Thus, indicated specific fuel consumption} = \text{isfc} = \frac{3600\ m_f}{IP}\ \text{kg/kWh}$$

Brake Specific fuel consumption*:*- This defined as the mass of fuel consumed per hour, in order to develop a brake power of one kilowatt.

$$\text{Thus, brake specific fuel consumption} = \text{bsfc} = \frac{3600\ m_f}{BP}\ \text{kg/kWh}$$

Thermal Efficiency : There are two definitions of thermal efficiency as applied to IC engines. One is based on indicated power and the other on brake power.The one based on indicated power is called as *'indicated thermal efficiency"*, and the one based on brake power is known as *"brake thermal efficiency".*

Indicated thermal efficiency is defined as the ratio of indicated power to the energy available due to combustion of the fuel.

Or $$ith = \frac{ip}{m \times CV}$$

Similarly brake thermal efficiency is defined as the ratio of brake power to energy available due to combustion of the fuel.

Or $$bth = \frac{bp}{m \times CV}$$

Mechanical Efficiency: Mechanical efficiency takes into account the mechanical losses in an engine. The mechanical losses include (i) frictional losses, (ii) power absorbed by engine auxillaries like fuel pump, lubricating oil pump, water circulating pump, magneto and distributor, electric generator for battery charging, radiator fan etc., and (iii) work requited to charge the cylinder with fresh charge and work for discharging the exhaust gases during the exhaust stroke. It is defined as the ratio of brake power to indicated power. Thus

$$mech = \frac{bp}{ip}$$

Volumetric efficiency: Volumetric efficiency is the ratio of the actual mass of air drawn into the cylinder during a given period of time to the theoretical mass which should have been drawn in during the same interval of time based on the total piston displacement, and the pressure and temperature of the surrounding atmosphere.

Thus $$v = \frac{V_a}{V_{th}}$$

where n is the number of intake strokes per minute and V_S is the stroke volume of the piston.

Heat balance Sheet

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

A heat balance sheet is an account of heat supplied and heat utilized in various ways in the system. Necessary information concerning the performance of the engine is obtained from the heat balance.

The heat balance is generally done on second basis or minute basis or hour basis.

The heat supplied to the engine is only in the form of fuel-heat and that is given by

$Q_S = m_f$ X CV

Where m_f is the mass of fuel supplied per minute or per sec. and CV is the lower calorific value of the fuel.

The various ways in which heat is used up in the system is given by

(a) Heat equivalent of BP = kW = kJ/sec. = kJ/min.

(b) Heat carried away by cooling water

$= C_{pw}$ X $m_W (T_{wo} - T_{wi})$ kJ/min

Where m_w is the mass of cooling water in kg/min or kg/sec circulated through the cooling jacket and $(T_{wo} - T_{wi})$ is the rise in temperature of the water passing through the cooling jacket of the engine and C_{pw} is the specific heat of water in kJ/kg-K.

(c) Heat carried away by exhaust gases

$= m_g C_{pg} (T_{ge} - T_a)$ (kJ/min.) or (kJ/sec)

Where m_g is the mass of exhaust gases in kg/min. or kg/sec and it is calculated by using one of the methods already explained.

T_g = Temperature of burnt gases coming out of the engine.

T_a = Ambient Temperature.

C_{pg} = Sp. Heat of exhaust gases in (kJ/kg-K)

(d) A part of heat is lost by convection and radiation as well as due to the leakage of gases. Part of the power developed inside the engine is also used to run the accessories as lubricating pump, cam shaft and water circulating pump. These cannot be measured precisely and so this is known as unaccounted 'losses'. This unaccounted heat energy is calculated bythe different between heat supplied Q_S and the sum of *(a) + (b)+(c).*

The results of the above calculations are tabulated in a table and this table is known as "Heat Balance Sheet". It is generally practice to represent the heat distribution as percentage of heat supplied. This is also tabulated in the same heat balance sheet.

Heat input per minute	*(kJ)*	%	*Heat expenditure per minute*	*(kJ)*	%
Heat supplied by the combustion fuel	Q_S	100%	(a) Heat in BP.	--	--
			(b) Heat carried by jacket cooling water	--	--
			(c) Heat Carried by exhaust gases	--	--
			(d) Heat unaccounted for $= Q_S - (a + b + c)$	--	--
Total	Q_S	100%			100%

A sample tabulation which is known as a heat balance sheet for particular load condition is shown below:

NOTE: The heat in frictional FP (IP – BP) should not be included separately in heat balance sheet because the heat of FP (frictional heat) will be dissipated in the cooling water, exhaust gases and radiation and convection. Since each of these heat quantities are separately measured and heat in FP is a hidden part of these quantities; the separate inclusion would mean that it has been included twice.

The arrangement either for measuring the air or measuring the mass of exhaust gas is sufficient to find the heat carried away by exhaust gases. In some cases, both arrangements are used for cross-checking. Heat carried away by exhaust gases is calculated with the help of volumetric analysis of the exhaust gases provided the fraction of carbon in the fuel used is known.

Illustrative examples:

Example 1:- The following observations have been made from the test of a fourcylinder, two – stroke petrol engine. Diameter of the cylinder = 10 cm; stroke = 15cm; speed = 1600 rpm; Area of indicator diagram = 5.5 cm^2; Length of the indicator diagram = 55 mm; spring constant = 3.5 bar/cm; Determine the indicated power ofthe engine.

Given:- d = 0.1 m; L = 0.15 m ; No. of cylinders = K = 4; N = 1600 rpm; n = N (two – stroke); a = 5.5 cm^2; length of the diagram = l_d = 5.5. cm; spring constant = k_s = 3.5 bar/cm ;
To find: indicated power, ip.

Solution: Indicated mean effective pressure = $p_{im} = \dfrac{a\,k_s}{l_d}$

$$\text{or} \quad p_{im} = \frac{5.5 \times 3.5}{5.5} = 3.5 \text{ bar} = 3.5 \times 10^5 \text{ N/m}^2$$

$$\text{Indicated power} = ip = \frac{p_{im}\,LAnK}{60{,}000} = \frac{3.5 \times 10^5 \times 0.15 \times (3.14\ /4) \times 0.1^2 \times 1600 \times 4}{60{,}000}$$

$$= 43.98 \text{ kW}$$

Example 2:- A gasoline engine (petrol engine) working on Otto cycle consumes 8 litres of petrol per hour and develops 25 kW. The specific gravity of petrol is 0.75 andits calorific value is 44,000 kJ/kg. Determine the indicated thermal efficiency of the engine

Given:- Volume of fuel consumed/hour = v/t = 8 x 10^3 / 3600 cc/s ;

ip = 25 kW; CV = 44,000 kJ/kg;

Specific gravity of petrol = s = 0.75
To find: ith

Solution: Mass of fuel consumed $= m = \dfrac{v\,s}{1000\,t} = \dfrac{0.75 \times 8 \times 10^{3}}{1000 \times 3600} = 1.67 \times 10^{-3}$ kg/s.

Indicated thermal efficiency $= \eta_{ith} = \dfrac{ip}{m_f\,CV} = \dfrac{25}{1.67 \times 10^{-3} \times 44000}$

$= 0.3402 = 34.02\ \%$.

Example 3:- The bore and stroke of a water cooled, vertical, single-cylinder, four stroke diesel engine are 80 mm and 110 mm respectively.The torque is 23.5 N- m.Calculate the brake mean effective pressure.What would be the mean effective pressure and torque if the engine rating is 4 kW at1500 rpm?

Given:- Diameter = d = 80 x10^{-3} = 0.008 m ; stroke = L = 0.110 m; T = 23.5 N-m;

To find (i) bmep ; (ii) bmep if bp = 4 kW and N= 1500 rpm.

Solution: (i) Relation between brake power (bp) and brake mean effective pressure (bmep) is given by

$$bp = \frac{2\pi\ NT}{60{,}000} = \frac{(bmep)LAn}{60{,}000}$$

Hence bmep = $(2\pi\ NT) / (LAn) = (2\pi\ NT) / \{(L\ d^2\ \pi/4)\ N/2\}$

$$= \frac{16T}{d^2L} = \frac{16 \times 23.5}{0.08^2 \times 0.11} = 5.34 \times 10^5\ N / m^2 = 5.34\ bar$$

(ii) when bp = 4 kw and N = 1500 rpm, we have

$$bmep = \frac{60{,}000\ bp}{LAn} = \frac{60{,}000 \times 4}{0.110 \times (3.14\ /4) \times 0.08^2 \times (1500 / 2)}$$

$= 5.79 \times 10^5\ N/m^2 = 5.79$ bar.

Also bp = $2\pi NT / 60{,}000$ or $T = \dfrac{60{,}000\ bp}{2\pi\ N} = \dfrac{60{,}000 \times 4}{2 \times \pi \times 1500} = 25.46$ N – m.

Example 4:-Find the air fuel ratio of a four stroke, single cylinder, air cooled enginewith fuel consumption time for 10 cc is 20.4 s and air consumption time for 0.1 $m^3$is 16.3 s. The load is 7 N at the speed of 3000 rpm. Find also the brake specific fuelconsumption in kg/kWh and brake thermal efficiency. Assume the density of air as 1.175 kg/m^3 and specific gravity of the fuel to be 0.7. The lower heating value of the fuel is 43MJ/kg and the dynamometer constant is 5000.

Given:- v= 10 cc ; t = 20.4 s ; $V_a = 0.1\ m^3$; $t_a = 16.3$ s ; W = 7 N ; N = 3000 rpm;

$_a = 1.175$ kg/m^3; s = 0.7 ; CV = 43×10^3 kJ/kg; Dynamometer constant = C = 5000.

To find:- (i) m_a / m_f ; (ii) bsfc ; (iii)$_{bth}$

Solution: (i) Mass of air consumed = $m_a = \dfrac{0.1 \times 1.175}{16.3} = 7.21 \times 10^{-3}$ kg/s.

Mass of fuel consumed = $m_f = \dfrac{v\ s}{1000\ t} = \dfrac{10 \times 0.7}{1000 \times 20.4} = 0.343 \times 10^{-3}$ kg/s

Air fuel ratio = $\dfrac{m_a}{m_f} = \dfrac{7.21 \times 10^{-3}}{0.343 \times 10^{-3}} = 21$

(ii) Brake power = bp = WN / C = $\dfrac{7 \times 3000}{5000} = 4.2$ kW

bsfc = $\dfrac{m_f \times 3600}{bp} = \dfrac{0.343 \times 10^{-3} \times 3600}{4.2} = 0.294$ kg/kWh

(iii) $_{bith}$ = $\dfrac{bp}{m_f\ CV} = \dfrac{4.2}{0.343 \times 10^{-3} \times 43 \times 10^3} = 0.2848 = 28.48$

Example 5:- A six cylinder, gasoline engine operates on the four stroke cycle. The bore of each cylinder is 80 mm and the stroke is 100 mm. The clearance volume in each cylinder is 70 cc. At a speed of 4000 rpm and the fuel consumption is 20 kg/h. The torque developed is 150 N-m. Calculate (i) the brake power, (ii) the brake mean effective pressure, (iii) brake thermal efficiency if the calorific value of the fuel is 43000 kJ/kg and (iv) the relative efficiency if the ideal cycle for the engine is Otto cycle.

Given:- K = 6 ; n = N /2 ; d = 8 cm ; L = 10 cm ; V_C = 70 cc ; N = 4000 rpm ; m_f = 20 kg/h ; T = 150 N-m ; CV = 43000 kJ/kg ;

To find:- (i) bp ; (ii) bmep ; (iii) η_{bth} ; (iv) $\eta_{Relative}$.

Solution:

(i) bp = $\frac{2\pi NT}{60,000} = \frac{2 \times \pi \times 4000 \times 150}{60,000}$

= 62.8 kW

(ii) bmep = $\frac{60,000\ bp}{L A n K} = \frac{60,000 \times 62.8}{0.1 \times (\pi/4) \times 0.08^2 \times (4000/2) \times 6}$

= 6.25 x 10^5 N/m^2 = 6.25 bar

(iii) $\eta_{bth} = \frac{bp}{m_f\ CV} = \frac{62.8}{(20/3600) \times 43,000}$ = 0.263 = 26.3%

(iv) Stroke volume = $V_S = (\pi/4)\, d^2 L = (\pi/4) \times 8^2 \times 10$ = 502.65 cc

Compression Ratio of the engine = $R_C = \frac{V_S + V_C}{V_C} = \frac{502.65 + 70}{70}$ = 8.18

Air standard efficiency of Otto cycle = $\eta_{Otto} = 1 - (1/r^{\gamma-1})$

$= 1 - \frac{1}{8.18^{1.4-1}}$ = 0.568 = 56.8 %

Hence Relative efficiency = $\eta_{Relative} = \eta_{bth} / \eta_{Otto}$ = 0.263 / 0.568 = 0.463 = 46.3 %.

Example .6:- An eight cylinder, four stroke engine of 9 cm bore, 8 cm stroke and with a compression ratio of 7 is tested at 4500 rpm on a dynamometer which has 54 cm arm. During a 10 minute test, the dynamometer scale beam reading was 42 kg and the engine consumed 4.4 kg of gasoline having a calorific value of 44,000 kJ/kg. Air at 27 C and 1 bar was supplied to the carburetor at a rate of 6 kg/min. Find (i) the brake power, (ii) the brake mean effective pressure, (iii) the brake specific fuel consumption, (iv) the brake specific air consumption, (v) volumetric efficiency , (vi) the brake thermal efficiency and (vii) the air fuel ratio.

Given:- K = 8 ; Four stroke hence n = N/2 ; d = 0.09 m; L = 0.08 m; R_c = 7; N = 4500

rpm; Brake arm = R = 0.54 m ; t = 10 min ; Brake load = W = (42 x 9.81) N.
m_f = 4.4 kg ; CV = 44,000 kJ/kg ; T_a = 27 + 273 = 300 K ; p_a = 1 bar; m_a = 6 kg/min;. .
To find:- (i) bp ; (ii) bmep ; (iii) bsfc ; (iv) bsac ; (v) $_v$; (vi) $_{bth}$; (vii) m_a / m_f

$$\text{(i) bp} = \frac{2\pi NT}{60{,}000} = \frac{2\pi NWR}{60{,}000} = \frac{2 \times \pi \times 4500 \times (42 \times 9.81) \times 0.54}{60{,}000}$$

$$= 104.8 \text{ kW}$$

$$\text{(ii) bmep} \quad \frac{60{,}000\ bp}{L\,A\,n\,K} = \frac{60{,}000 \times 104.8}{0.08 \times (\pi / 4) \times 0.09^2 \times (4500/2) \times 8}$$

6.87×10^5 N/m^2

(iii) mass of fuel consumed per unit time = m_f = m_f / t = 4.4 x 60 / 10 kg/h

= 26.4 kg/h.

$$\text{Brake specific fuel consumption} = bsfc = \frac{m_f}{bp} = \frac{26.4}{104.8} = 0.252 \text{ kg / kWh}$$

$$\text{(iv) brake specific air consumption} = bsac = \frac{m_a}{bp} = \frac{6 \times 60}{104.8}$$

= 3.435 kg / kWh

$$\text{(v)} \quad \eta_{bth} = \frac{bp}{m_f \; CV} = \frac{104.8}{(26.4 / 3600) \times 44{,}000.} = 0.325 = 32.5$$

(vi) Stroke volume per unit time = V_s = (π d^2/4) L n K

$$= \frac{\pi}{4} (0.09^2) \times 0.08 \times (4500 / 2) \times 8$$

= 9.16 m^3 / min..

$$\text{Volume flow rate of air per minute} = \dot{V}_a = \frac{m_a R_a T_a}{p_a} = \frac{6 \times 286 \times 300}{1 \times 10^5}$$

= 5.17 m^3 / min

Volumetric efficiency = η_v = V_a / V_s = 5.17 / 9.16 = 0.5644 = 56.44%.. .

(vii) Air fuel ratio = m_a / m_f = 6 /(4.4 / 10) =13.64

Example 8:- The following observations were recorded during a trail of a four – stroke, single cylinder oil engine.Duration of trial = 30 min ; oil consumed = 4 litres ; calorific value of oil = 43 MJ/kg ;specific gravity of fuel = 0.8 ; average area of the indicator diagram = 8.5 cm^2; lengthof the indicator diagram = 8.5 cm; Indicator spring constant = 5.5 bar/cm; brake load =150 kg; spring balance reading = 20 kg; effective brake wheel diameter = 1.5 m ; speed= 200 rpm ; cylinder diameter = 30 cm ; stroke = 45 cm ; jacket cooling water = 10kg/min ; temperature rise of cooling water = 36 C. Calculate (i) indicated power,(ii) brake power, (iii) mechanical efficiency, (iv) brake specific fuel consumption,
(v) indicated thermal efficiency, and (vi) heat carried away by cooling water.

Given:- t = 30 min ; v= 4000 cc; CV = 43 x10^3 kJ/kg; s = 0.8 ; area of the diagram = a = 8.5 cm^2; length of the diagram = l_d = 8.5 cm ; indicator spring constant = k_s = 5.5 bar / cm; W = 150 x 9.81 N ; Brake radius = R = 1.5 / 2 = 0.75 m; N = 200 rpm ; d = 0.3 m ; L = 0.45 m ; m_w = 10 kg/min ; T_w = 36 C; Spring Balance Reading = S = 20 x 9.81 N.

To find:- (i) ip ; (ii) bp ; (iii) η_{mech} ; (iv) bsfc ; (v) η_{ith} ; (vi) Q_w

Solution:

(i) $p_{im} = \dfrac{a}{l_d} k_s = \dfrac{8.5}{8.5} \times 5.5 = 5.5 \text{ bar} = 5.5 \times 10^5 \text{ N/m}^2$

$$ip = \frac{P_{im} L A n K}{60{,}000} = \frac{5.5 \times 10^5 \times 0.45 \times (\pi / 4) \times 0.3^2 \times (200 / 2) \times 1}{60{,}000}$$

$= 29.16$ kW

$$\text{(ii) } bp = \frac{2\pi N(W - S) R}{60{,}000} = \frac{2 \times \pi \times 200 \times (150 - 20) \times 9.81 \times 0.75}{60{,}000}$$

$= 20.03$ kW

(iii) $\eta_{mech} = bp / ip = 20.03 / 29.16 = 0.687 = 68.7$ %.

(iv) Mass of fuel consumed per hour $= \dot{m}_f = \dfrac{vs}{1000\, t} \times 60 = \dfrac{4000 \times 0.8}{1000 \times 30}\, 60$

$= 6.4$ kg / h.

$$bsfc = m_f / bp = \frac{6.4}{20.03} = 0.3195 \text{ kg/kWh}$$

$$\text{(v) } \eta_{ith} = \frac{ip}{\dot{m}_f\, CV} = \frac{29.16}{(6.4 / 3600) \times 43 \times 10^3} = 0.3814 = 38.14$$

(vi) $Q_W = m\, C_p\, T_W = (10 / 60) \times 4.2 \times 36 = 25.2$ kW

Example 9:- A four stroke gas engine has a cylinder diameter of 25 cm and stroke 45 cm. The effective diameter of the brake is 1.6 m.The observations made in a test ofthe engine were as follows.

Duration of test = 40 min; Total number of revolutions = 8080 ; Total number of explosions = 3230; Net load on the brake = 80 kg ; mean effective pressure = 5.8 bar;Volume of gas used = 7.5 m^3; Pressure of gas indicated in meter = 136 mm of water (gauge); Atmospheric temperature = 17 C; Calorific value of gas = 19 MJ/ m^3 at NTP; Temperature rise of cooling water = 45 C; Cooling water supplied = 180 kg. Draw up a heat balance sheet and find the indicated thermal efficiency and brake thermal efficiency. Assume atmospheric pressure to be 760 mm of mercury.

Given:- d = 0.25 m ; L = 0.45 m; R = 1.6 / 2 =0.8 m; t = 40 min ; N_{total} = 8080

; Hence N = 8080 / 40 = 202 rpm n_{total} = 3230 ;

Hence n = 3230 / 40 = 80.75 explosions / min; W = 80 x 9.81 N; p_{im} = 5.8 bar ;

V_{total} = 7.5 m^3 ; hence $\dot{V}$ = 7.5 / 40 = 0.1875 m^3 /min; p_{gauge} = 136 mm of water (gauge);

T_{atm} = 17 + 273 = 290 K; $(CV)_{NTP}$ = 19 x 10^3 kJ/ m^3 ; T_W = 45 C;.
m_W = 180 / 40 = 4.5 kg/min; p_{atm} = 760 mm of mercury

To find:- (i) ith ; (ii) bth ; (iii) heat balance sheetSolution:

(i) $$ip = \frac{p_{im} \, L \, A \, n \, K}{60,000} = \frac{5.8 \times 10^5 \times (\pi / 4) \times 0.25^2 \times 0.45 \times 80.75}{60,000}$$

= 17.25 kW.

$$bp = \frac{2 \pi N W R}{60,000} = \frac{2 \times \pi \times 202 \times (80 \times 9.81) \times 0.8}{60,000}$$

= 13.28 kW

Pressure of gas supplied = p = p_{atm} + p_{gauge} = 760 + 136 / 13.6 = 770 mm of mercury.

Volume of gas supplied as measured at NTP = V_{NTP} = $\dot{V}$ (T_{NTP} / T)(p / p_{NTP})

$$= \frac{0.1875 \times 273 \times 770}{290 \times 760} = 0.17875 \, m^3 / min$$

Heat supplied by fuel = $Q_f = \dot{V}_{NTP}$ $(CV)_{NTP}$ = 0.17875 x 19 x 10^3 = 3396.25 kJ/min

Heat equivalent of bp in kJ/min = 13.28 x 60 = 796.4 kJ/min.
Heat lost to cooling water in kJ/min = m_W C_p T_W = 4.5 x 4.2 x 45 = 846.5 kJ/min

Friction power = ip – bp = 17.25 – 13.28 = 3.97 kW

Hence heat loss due to fiction, pumping etc. = 3.97 x 60 = 238.2 kJ/min

Heat lost in exhaust, radiation etc (by difference) = 3396.25 – (896.4 + 796.4 + 238.2)

= 1465.15 kJ/min

Heat Balance Sheet:

Item No.		*Heat Energy Input (kJ/mi)*	*(percen)*	*Heat Energy spent (kJ/min)*	*(percent)*
1	Heat supplied by fuel	3396.25	100.00		
2	Heat equivalent of bp			896.4	26.4
3	Heat lost to cooling Water			796.4	23.4
4	Heat equivalent of fp			238.2	7.0
5	Heat unaccounted (by difference)			1465.15	43.2
	Total	3396.25	100.0	3396.25	100.0

Example 10:- A test on a two-stroke engine gave the following results at full load.
Speed = 350 rpm; Net brake load = 65 kg ; mean effective pressure = 3 bar ; Fuel consumption = 4 kg/h ; Jacket cooling water flow rate = 500 kg/h ; jacket water temperature at inlet = 20 C ; jacket water temperature at outlet = 40 C ; Test room temperature = 20 C ; Temperature of exhaust gases = 400 C; Air used per kg of fuel =32 kg ; cylinder diameter = 22 cm ;stroke = 28 cm; effective brake diameter = 1 m ;Calorific value of fuel = 43 MJ/kg ; Mean specific heat of exhaust gases = 1 kJ/kg K. Find indicated power, brake power and draw up a heat balance for the test in kWand in percentage.

Given:- Two stroke engine. Hence n = N ; N = 350 rpm ; W = (65 x 9.81) N ;..
p_{im} = 3 bar ; m_f = 4 kg/h ; m_w = 500 kg/h ; T_{wi} = 20 C ; T_{wo} = 40 C ; T_{atm} = 20 ^{0}C ;.   .
T_{eg} = 400^0 C ; m_a / m_f = 32 ; d = 0.22 m ; L = 0.28 m ; Brake radius = R = ½ m ;

CV = 43,000 kJ/kg ; $(C_p)_{eg}$ = 1.0 kJ/(kg-K) ;

To find:- (i) ip ; (ii) bp ; and (iii) heat balance;

Solution:

$$\text{(i) ip} = \frac{p_{im}\, LAn}{60{,}000} = \frac{3 \times 10^5 \times 0.28 \times (\pi/4) \times 0.22^2 \times 350}{60{,}000}$$

$$= 18.63 \text{ kW.}$$

$$\text{(ii) bp} = \frac{2\pi NWR}{60{,}000} = \frac{2 \times \pi \times 350 \times (65 \times 9.81) 0.5}{60{,}000}$$

$$= 11.68 \text{ kW.}$$

(iii) Heat supplied in kW = m_f CV = (4 / 3600) x 43,000

$$= 47.8 \text{ kW.}$$

Heat lost to cooling water = $m_w (C_p)_w [T_{wo} - T_{wi}]$

$$= (500 / 3600) \times 4.2 \times [40 - 20]$$

$$= 11.7 \text{ kW.}$$

Heat lost in exhaust gases = $(m_a + m_f) (C_p)_{eg} [T_{eg} - T_{atm}]$

$$= \frac{(32 + 1) \times 4}{3600} \times 1.0 \times [400 - 20]$$

$$= 13.9 \text{ kW}$$

Heat balance sheet:

Heat Input	kW	%	Heat Expenditure	kW	%
Heat supplied by fuel	47.8	100	Heat in bp	11.68	24.4
			Heat lost to cooling Water	11.70	24.5
			Heat lost to exhaust Gases	13.90	29.1
			Unaccounted heat (by difference)	10.52	22.0
Total	**47.80**	**100**	**Total**	**47.80**	**100.0**

UNIT – 3 – AIR COMPRESSORS – SMEA1404

Syllabus

Classification and working of air compressors - Classification of compressors and their comparison - Reciprocating air compressor-principle of operation, work requirement, isothermal efficiency, volumetric efficiency and effect of clearance, Multi stage compression with inter cooling, saving work, Working of Rotary Compressors and comparison with reciprocating air compressors.

Unit 3: Air Compressors

3.1 Introduction:

Compression of air and vapour plays an important role in engineering fields. Compression of air is mostly used since it is easy to transmit air compared with vapour.

3.2 Uses of compressed air:

The applications of compressed air are listed below:

1) It is used in gas turbines and propulsion units.
2) It is used in striking type pneumatic tools for concrete breaking, clay or rock drilling, chipping, caulking, riveting etc.
3) It is used in rotary type pneumatic tools for drilling, grinding, hammering etc.
4) Pneumatic lifts and elevators work by compressed air.
5) It is used for cleaning purposes
6) It is used as an atomiser in paint spray and insecticides spray guns.
7) Pile drivers, extractors, concrete vibrators require compressed air.
8) Air-operated brakes are used in railways and heavy vehicles such as buses and lorries.
9) Sand blasting operation for cleaning of iron castings needs compressed air.
10) It is used for blast furnaces and air-operated chucks.
11) Compressed air is used for starting I.C.engines and also super charging them.

3.3 Working principle of a compressor:

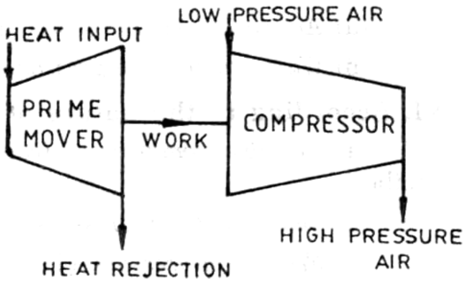


Fig:3.1 Air Compressor

A line diagram of a compressor unit is shown in fig:4.1. The compression process requires work input. Hence a compressor is driven by a prime mover. Generally, an electricmotor is used as prime mover. Air from atmosphere enters into the compressor It is compressed to a high pressure. Then, this high pressure air is delivered to a storage vessel (reservoir).From the reservoir, it can be conveyed to the desired place through pipe lines.

Some of the energy supplied by the prime mover is absorbed in work done against friction. Some portion of energy is lost due to radiation and coolant. The rest of the energy is maintained within the high pressure air delivered.

3.4 Classification of compressors:

Air compressors may be classified as follows:

According to design and principle of operation:

(a) Reciprocating compressors in which a piston reciprocates inside the cylinder.
(b) Rotary compressors in which a rotor is rotated.

According to number of stages:

(a) Single stage compressors in which compression of air takes place in one cylinder only.
(b) Multi stage compressors in which compression of air takes place in more than one cylinder.

According to pressure limit:

(a) Low pressure compressors in which the final delivery pressure is less than 10 bar,
(b) Medium pressure compressor in which the final delivery pressure is 10 bar to 80 bar and
(c) High pressure compressors in which the final delivery pressure is 80 to 100 bar.

According to capacity:

(a) Low capacity compressor (delivers $0.15m^3$ /s of compressed air),
(b) Medium capacity compressor (delivers $5m^3$ /s of compressed air) and
(c) High capacity compressor (delivers more than $5m^3$ /s of compressed air).

According to method of cooling:

*(a)*Air cooled compressor (Air is the cooling medium) *and*
*(b)*Water cooled compressor (Water is the cooling medium).

According to the nature of installation:

(a) Portable compressors (can be moved from one place to another).
(b) Semi-fixed compressors and
(c) Fixed compressors (They are permanently installed in one place).

According to applications:

(a) Rock drill compressors (used for drilling rocks),
(b) Quarrying compressors (used in quarries),
(c) Sandblasting compressors (used for cleaning of cast iron)and
(d) Spray painting compressors (used for spray painting).

According to number of air cylinders

(a) Simplex - contains one air cylinder
(b) Duplex - contains two air cylinders
(c) Triplex - contains three air cylinders

3.4.1 Reciprocating compressors may be classified as follows:

(a) Single acting compressors in which suction, compression and delivery of air (or gas) take place on one side of the piston.

(b) Double acting compressors in which suction, compression and delivery of air (or gas) take place on both sides of the piston.

3.5 Single stage reciprocating air compressor:

In a single stage compressor, the compression of air (or gas) takes place in a single cylinder. A schematic diagram of a single stage, single acting compressor is shown in fig:3.2.

Construction*:* It consists of a piston which reciprocates inside a cylinder. The piston is connected to the crankshaft by means of a connecting rod and a crank. Thus, the rotary movement of the crankshaft is converted into the reciprocating motion of the piston. Inlet and outlet valves (suction and delivery valves) are provided at the top of the cylinder.

Working: When the piston moves down, the pressure inside the cylinder is reduced. When the cylinder pressure is reduced below atmospheric pressure, the inlet valve opens. Atmospheric air is drawn into the cylinder till the piston reaches the bottom dead centre. The delivery valve remains closed during this period. When the piston moves up, the pressure inside the cylinder increases. The inlet valve is closed, since the pressure inside the cylinder is above atmospheric. The pressure of air inside the cylinder is increased steadily. The outlet valve is then opened and the high pressure air is delivered through the outlet valve in to the delivery pipe line.

At the top dead centre of the piston, a small volume of high pressure air is left in the clearance space. When the piston moves down again, this air is expanded and pressure reduces, Again the inlet valve opens and thus the cycle is repeated.

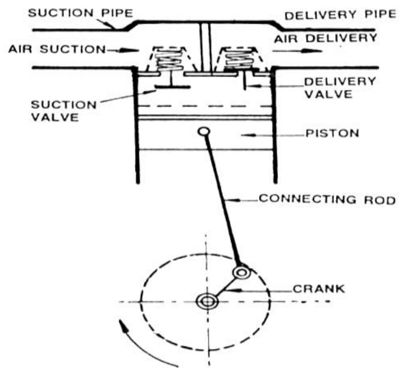


Fig 3:.2 Single stage reciprocating

Disadvantages

1. Handling of high pressure air results in leakage through the piston.
2. Cooling of the gas *is not effective.*
3. Requires a stronger cylinder to withstand high delivery pressure.

Applications:It is used in places where the required pressure ratio is small.

3.6 Compression processes:

The air may be compressed by the following processes.

(a) Isentropic or adiabatic compression,

(b) Polytropic compression and

(c) Isothermal compression

(a)Isentropic(or)adiabatic compression*:*

In internal combustion engines, the air (or air fuel mixture) is compressed isentropically. By isentropic compression, maximum available energy in the gas is obtained.

(b)Polytropic compression:

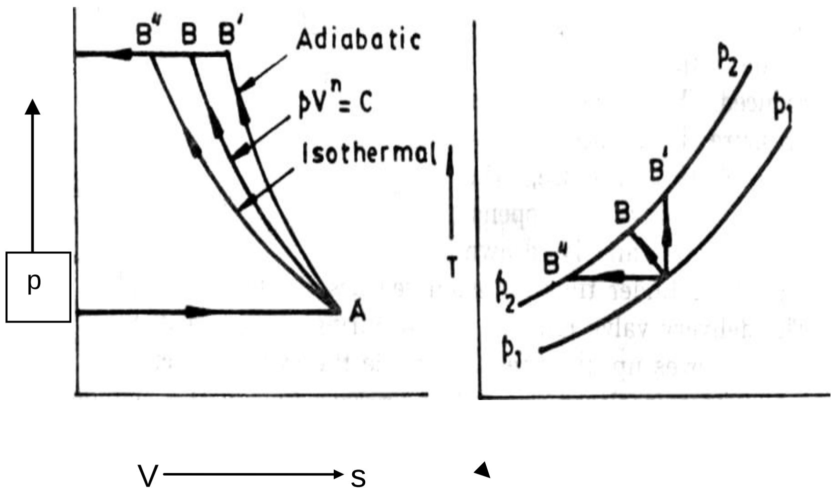


Fig: 3.3 Compression processes
A-B": Isothermal; A-B: Polytropic; A-B': Isentropic

The compression follows the law pV^n = Constant. This type of compression may be used in Bell-Coleman cycle of refrigeration.

(c)Isothermal compression:

When compressed air (or gas) is stored in a tank, it loses its heat to the surroundings. It attains the temperature of surroundings after some time. Hence, the overall effect of this compression process is to increase the pressure of the gas keeping the temperature constant. Thus isothermal compression is suitable if the compressed air (or gas) is to be stored.

3.7 Power required for driving the compressor:

The following assumptions are made in deriving the power required to drive the compressor.

1. There is no pressure drop through suction and delivery valves.
2. Complete compression process takes place in one cylinder.
3. There is no clearance volume in the compressor cylinder.
4. Pressure in the suction line remains constant. Similarly, pressure in the delivery line remains constant.
5. The working fluid behaves as a perfect gas.
6. There is no frictional losses.

The cycle can be analysed for the three different case of compression. Work required can be obtained from the p - V diagram.

Let,

p_1 =Pressure of the air (kN/m^2), before compression
V_1 = Volume of the air (m^3), before compression

T_1 =Temperature of the air *(K)*, before compression

p_2, V_2 and T_2 be the corresponding values after compression.

m - Mass of air induced or delivered by the cycle (kg).
N - Speed in RPM.

3.7.1 Polytropic Compression

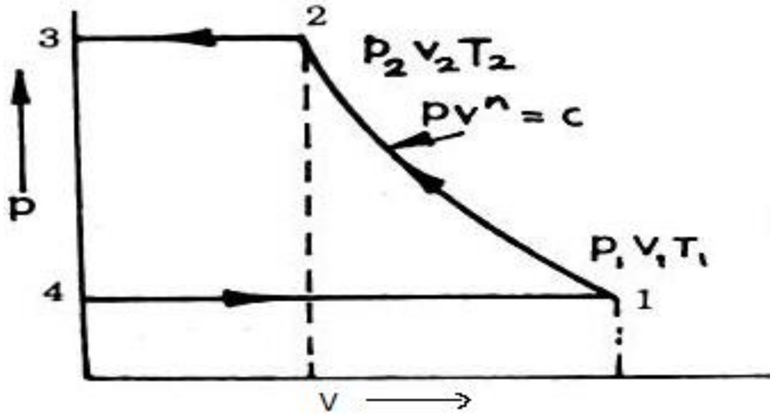


Fig:3.4 Polytropic compression
(Compression follows pV^n = Constant)

Let n= Index of polytropic compression

Net work done on air/cycle is given by

W = Area 1-2-3-4-1

= Work done during compression (1-2) + Work done during air delivery (2-3) - Work done during suction (4-1).

$$W = \frac{p_2v_2 - p_1v_1}{n-1} + p_2v_2 - p_1v_1$$

$$W = \frac{p_2v_2 - p_1 + (n-1)p_2v_2 - (n-1)p_1v_1}{n-1}$$

$$= \frac{np_2v_2 - np_1v_1}{n-1} = \left(\frac{n}{n-1}\right)p_2v_2 - p_1v_1$$

We know that, $p_1V_1 = m\,RT_1$ & $p_2V_2 = m\,RT_2$

Therefore, **W =** $\frac{n}{n-1}$ **m R (T$_2$ - T$_1$)**

W = $\frac{n}{n-1}$ **m R T$_1$**$\left[\frac{T_2}{T_1} - 1\right]$

For polytropic process, $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$

Therefore, W = $\frac{n}{n-1}$ m R T$_1$$\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$kJ/cycle

W = $\frac{n}{n-1}$ p$_1$ V$_1$$\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$ kJ/cycle

Indicated power (or) Power required, P = W x N , kW for single acting reciprocating compressor;

= W x 2N, kW for double acting reciprocating compressor.

3.7.2Isentropic compression

Compression follows, pV^{γ}= Constant

Let γ= Index of isentropic compression

Net work done on air/cycle is given by

W = Area 1-2-3-4-1

= Work done during compression (1-2) + Work done during air delivery (2-3) - Work done during suction (4-1).

$$W = \frac{p_2v_2 - p_1v_1}{\gamma-1} + p_2v_2 - p_1v_1$$

$$W = \frac{p_2v_{2-}\ p_1 + (\gamma-1)p_2v_2 - \ (\gamma-1)p_1v_1}{\gamma-1}$$

$$= \frac{\gamma p_2v_2 - \gamma p_1v_1}{\gamma-1} = \left(\frac{\gamma}{\gamma-1}\right)p_2v_2 - p_1v_1$$

We know that, p$_1$V$_1$ = m RT$_1$ & p$_2$V$_2$ = m RT$_2$

W = $\frac{\gamma}{\gamma-1}$ **m R (T$_2$ - T$_1$)**

W = $\frac{\gamma}{\gamma-1}$ **m R T$_1$**$\left[\frac{T_2}{T_1} - 1\right]$

For isentropic process, $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$

Therefore, W = $\frac{\gamma}{\gamma-1}$ m R T$_1$$\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$kJ/cycle

$$W = \frac{\gamma}{\gamma-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \text{ kJ/cycle}$$

3.7.3 Isothermal Compression

Compression follows, pV= Constant

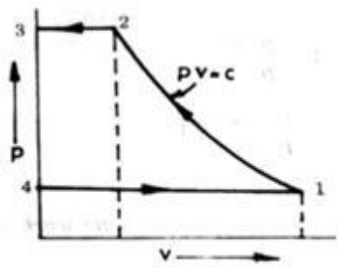


Fig: 3.5 Isothermal Compression

Isothermal Work input, W = Area 1-2-3-4-1 = area under 1-2 + area under 2-3 - area under 4-1

$$W = p_1V_1 \ ln\left(\frac{V_1}{V_2}\right) + p_2V_2 - p_1V_1$$

But $p_1V_1 = p_2V_2$

$$W = p_1V_1 \ ln\left(\frac{V_1}{V_2}\right) \qquad and \ \frac{V_1}{V_2} = \frac{p_2}{p_1}$$

Therefore, $W = p_2V_2 \ ln\left(\frac{p_2}{p_1}\right)$ kJ/cycle

3.8 Isothermal efficiency: Isothermal efficiency is defined as the ratio of isothermal work input to the actual work input. This is used for comparing the compressors.

Isothermal efficiency, $\eta_{iso} = \frac{Isothermalworkinput}{Actualworkoutput}$

3.9 Adiabatic efficiency: Adiabatic efficiency is defined as the ratio of adiabatic work input to the actual work input. This is used for comparing the compressors.

Adiabatic efficiency, $\eta_{adia} = \frac{Adiabaticworkinput}{Actualworkoutput}$

3.10 Mechanical efficiency:

The compressor is driven by a prime mover. The power input to the compressor is the shaft power (brake power) of the prime mover. This is also known as brake power of the compressor.

Mechanical efficiency is defined as the ratio of indicated power of the compressor to the power input to the compressor.

$$\eta_m = \frac{\text{Indicated power of compressor}}{\text{Power input}}$$

Indicated Power, $IP = \frac{\boldsymbol{p_m l \, aNk}}{\mathbf{60}}$,

where, p_m = mean effective pressure, kN/m²

l = length of stroke of piston, m

a = area of cross section of cylinder,m²
N= crank speed in rpm, and
K=number of cylinders

3.11 Clearance and clearance volume:

When the piston reaches top dead centre (TDC) in the cylinder, there is a dead space between piston top and the cylinder head. This space is known as clearance space and the volume occupied by this space is known as clearance volume, V_c.

The clearance volume is expressed as percentage of piston displacement. Its value ranges from 5% - 10% of swept volume or stroke volume (V_s).The p - V diagram for a single stage compressor, considering clearance volume is shown in fig. . At the end of delivery of high pressure air (at point 3), a small amount of high pressure air at p_2 remains in the clearance space. This high pressure air which remains at the clearance space when the piston is at TDC is known as remnant air. It is expanded polytropically till atmospheric pressure (p_4=p_1) is reached. The inlet valve is opened and the fresh air is sucked into the cylinder. The suction of air takes place for the rest of stroke (upto point 1). The volume of air sucked is known as effective suction volume (V_1 - V_4). At point 1, the air is compressed polytropically till the delivery pressure (p_2) is reached. Then the delivery valve is opened and high pressure air is discharged into the receiver. The delivery of air continues till the piston reaches its top dead centre, then the cycle is repeated.

3.11.1 Effect of clearance volume:

The following are the effects of clearance space.
1. Suction volume (volume of air sucked) is reduced.
2. Mass of air is reduced.
3. If clearance volume increases, heavy compression is required.
4. Heavy compression increases mechanical losses

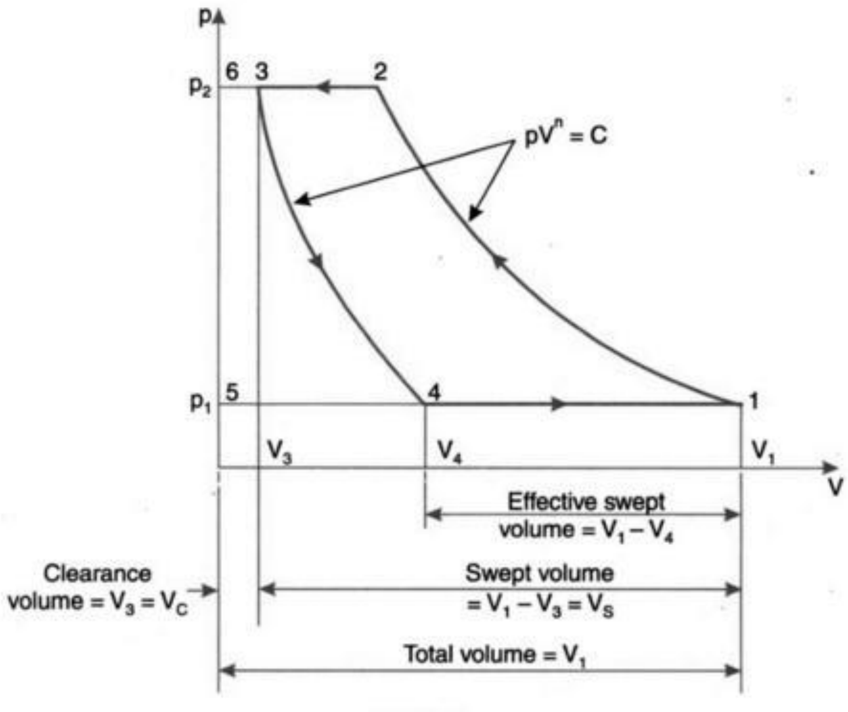


Fig: 3.6 p-V diagram with clearance volume

3.11.3 Work input considering clearance volume:

Assuming the expansion (3-4) and compression (1-2) follow the law p V^n = C,
Work input per cycle is given by,
W = Area (1-2-3-6-5-4-1)- Area (3-6-5-4 -3)
W = Workdone during compression - Work done during expansion

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

But, $p_3 = p_2$ and $p_4 = p_1$
therefore

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

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

V_1-V_4 is called as effective suction volume.

3.12 Volumetric efficiency:

The clearance volume in a compressor reduces the intake capacity of the cylinder. This leads to a term called volumetric efficiency.
The volumetric efficiency is denned as the volume of free air sucked into the compressor per cycle to the stroke volume of the cylinder,the volume measured at the intake pressure and temperature or at standard atmospheric conditions,(p_s =101.325 kN/m^2 and T_s = 288K)

$$\text{Volumetric efficiency, } \eta_{vol} = \frac{\text{Volume of free air taken in per cycle}}{\text{Stroke volume of the cylinder}}$$

$$= \frac{\text{Effective suction volume}}{\text{Swept volume}} = \frac{(V1 - V4)}{(V1 - V3)} = \frac{V_1 - V_4}{V_s}$$

Clearance ratio: Clearance ratio is defined as, the ratio of clearance volume to swept volume. It is denoted by the letter C.

$$\text{Clearance ratio,} C = \frac{Clearance\ volume}{Swept\ volume} = \frac{V_c}{V_s} = \frac{V_c}{V_{1-V_3}}$$

$$\text{Pressure ratio, } R_p = \frac{Delivery\ pressure}{Suction\ pressure} = \frac{p_2}{p_1} = \frac{p_3}{p_4}$$

3.12.1 Expression for Volumetric efficiency

Let the compression and expansion follows the law, pV^n =Constant.

$$\text{Clearance ratio, } C = \frac{Clearancevolume}{Sweptvolume} = \frac{V_c}{V_s} = \frac{V_3}{V_{1-V_3}}$$

V_1-V_3 $=\frac{V_3}{C}$-------------(1)

$V_1 = \frac{V_3}{C} + V_3$

$V_1 = V_3\left(\frac{1}{C} + 1\right)$--------------------- (2)

$$\text{We know that, Pressure ratio, } R_p = \frac{Delivery\ pressure}{Suction\ pressure} = \frac{p_2}{p_1} = \frac{p_3}{p_4}$$

By polytropic expansionprocess 3-4:

$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^n$$

$$\frac{V_4}{V_3} = \left(\frac{p_3}{p_4}\right)^{1/n} = \left(R_p\right)^{\frac{1}{n}}$$

Therefore, $V_4 = V_3\left(R_p\right)^{\frac{1}{n}}$----------------- (3)

Volumetric efficiency, $\eta_{vol} = \frac{\text{Effective suction volume}}{\text{Swept volume}} = \frac{(V_1 - V_4)}{(V_1 - V_3)}$------------ (4)

Using equations 1,2 and 3 in 4,

$$\eta_{vol} = \frac{V_3\left(\frac{1}{C}+1\right) - V_3\left[R_p\right]^{1/n}}{\frac{V_3}{C}} = \frac{V_3\left\{\left(\frac{1}{C}+1\right) - \left[R_p\right]^{1/n}\right\}}{V_3\left(\frac{1}{C}\right)} = \frac{\left\{\left(\frac{1}{C}+1\right) - \left[R_p\right]^{1/n}\right\}}{\left(\frac{1}{C}\right)} = C\left[\left(\frac{1}{C}+1\right) - \left[R_p\right]^{1/n}\right]$$

$$\eta_{vol} = 1 + C - C\left[R_p\right]^{1/n} = 1 + C - C\left[\frac{p_2}{p_1}\right]^{1/n}$$

3.13 Multi-stage air compressor:

In a multi stage air compressor, compression of air takes place in more than one cylinder. Multi stage air compressor is used in places where high pressure air is required. Fig. shows the general arrangement of a two-stage air compressor. It consists of a low pressure (L.P) cylinder, an intercooler and a high pressure (H.P) cylinder. Both the pistons (in L.P and H.P cylinders) are driven by a single prime mover through a common shaft.

Atmospheric air at pressure p_1 taken into the low pressure cylinder is compressed to a high pressure (p_2). This pressure is intermediate between intake pressure (p_1) and delivery pressure p_3). Hence this is known as intermediate pressure.

The air from low pressure cylinder is then passed into an intercooler. In the intercooler, the air is cooled at constant pressure by circulating cold water. The cooled air from the intercooler is then taken into the high pressure cylinder. In the high pressure cylinder, air is further compressed to the final delivery pressure (p_3) and supplied to the air receiver tank.

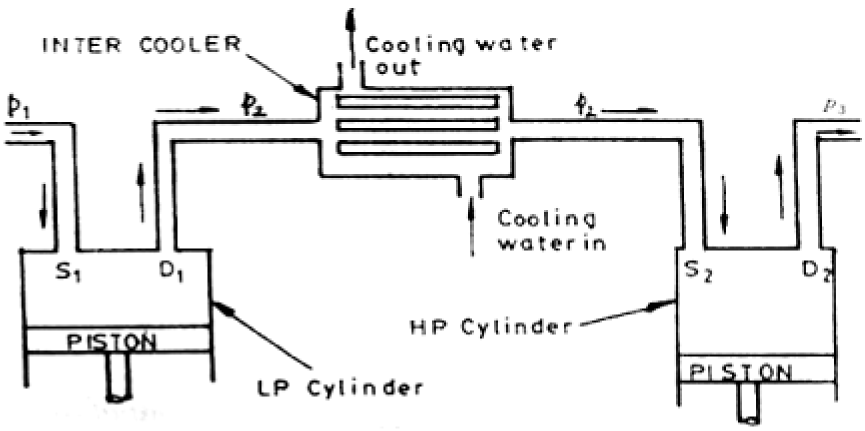


Fig: 3.7 Multistage compressor (Two stage)

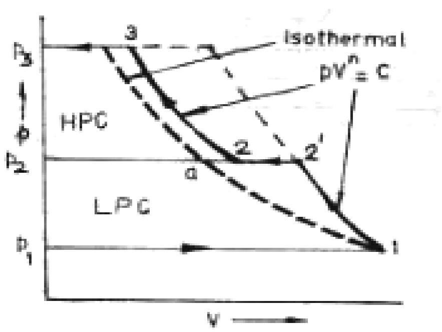


Fig:3.8pV diagram of two stage compressor

Advantages:

1. Saving in work input: The air is cooled in an intercooler before entering the high pressure cylinder. Hence less power is required to drive a multistage compressor as compared to a single stage compressor for delivering same quantity of air at the same delivery pressure.

2. Better balancing: When the air is sucked in one cylinder, there is compression in the other cylinder. This provides more uniform torque. Hence size of the flywheel is reduced.

3. No leakage and better lubrication: The pressure and temperature ranges are kept within desirablelimits. This results in a) Minimum air leakage through the piston of the cylinder and b) effective lubrication due to lower temperature.

4. More volumetric efficiency: For small pressure range, effect of expansion of the remnant air (high pressure air in the clearance space) is less. Thus by increasing number of stages, volumetric efficiency is improved.

5. High delivery pressure: The delivery pressure of air is high with reasonable volumetric efficiency.

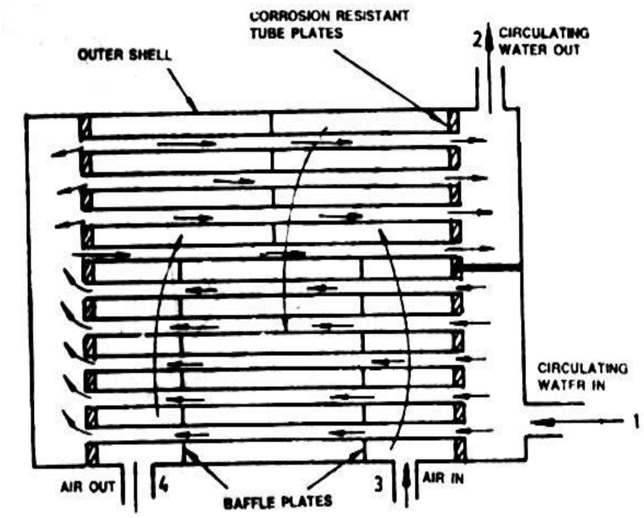
6. Simple construction of LP cylinder: The maximum pressure in the low pressure cylinder is less. Hence, low pressure cylinder can be made lighter in construction.

7. Cheaper materials: Lower operating temperature permits the use of cheaper materials for construction.

Disadvantages:
1. More than one cylinder is required.
2 An intercooler is required. This increases initial cost. Also space required is more.
3. Continuous flow of cooling water is required.
4. Complicated in construction.

3.14 Intercoolers:

An intercooler is a simple heat exchanger. It exchanges the heat of compressed air from the LP compressor to the circulating water before the air enters the HP compressor. It consists of a number of special metal tubes connected to corrosion resistant plates at both ends. The entire nest of tubes is covered by an outer shell



Working: Cold water enters the bottom of the intercooler through water inlet (1) and flows into the bottom tubes. Then they pass through the top tubes and leaves through the water outlet (2) atthe top. Air from LP compressor enters through the air inlet (3) of the intercooler and passes over the tubes. While passing over the tubes, the air is cooled (by the cold water circulated throughthe tubes). This cold air leaves the intercooler through the air outlet (4). Baffle plates are provided in the intercooler to change the direction of air. This provides a better heat transfer from air to the circulating water.

Fig:3.9 Intercooler

3.15 Work input required in multistage compressor:

The following assumptions are made for calculating the work input in multistage compression.
1. Pressure during suction and delivery remains constant in each stage.
2. Intercooling takes place at constant pressure in each stage.

3. The compression process is same for each stage.
4. The mass of air handled by LP cylinder and HP cylinder is same.
5. There is no clearance volume in each cylinder.
6 There is no pressure drop between the two stages, i.e., exhaust pressure of one stage is equal to the suction pressure of the next stage.

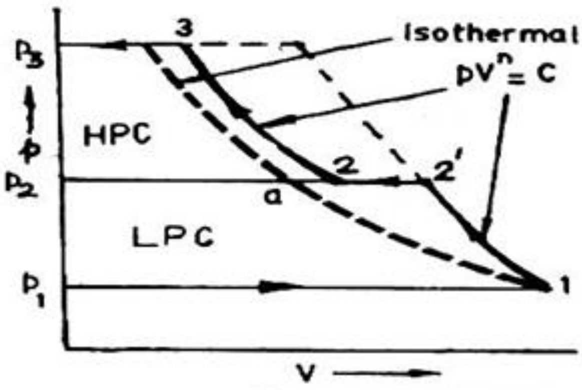


Fig:3.10 Two Stage compression

Work required to drive the multi-stage compressor can be calculated from the area of the p - V diagram .
Let, p_1, V_1 and T_1 be the condition of air entering the LP cylinder.
P_2, V_2 and T_2 be the condition of air entering the HP cylinder.
p_3 be the final delivery pressure of air.

Then,
Total work input = Work input for LP compressor + Work input for HP compressor.

$$W = \frac{n}{n-1} p_1 V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} p_1 V_1\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right] \text{ kJ/cycle}$$

$$W = \frac{n}{n-1} m R T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} m R T_2\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right] \text{kJ/cycle}$$

If intercooling is perfect, $T_2 = T_1$, therefore,

$$W = \frac{n}{n-1} m R T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1} m R T_1\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right] \text{kJ/cycle}$$

$$W = \frac{n}{n-1} m R T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{kJ/cycle}$$

Or

$$W = \frac{n}{n-1} p_1 V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{kJ/cycle}$$

3.16 Condition for maximum efficiency (or) Condition for minimum work input (or) To prove that for minimum work input the intermediate pressure of a two-stage compressor with perfect intercooling is the geometric mean of the intake pressure and delivery pressure (or)

To prove $p_2 = \sqrt{p_1\, p_3}$

Work input for a two-stage air compressor with perfect intercooling is given by,

$$W = \frac{n}{n-1} p_1 V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{ kJ/cycle}$$

If the initial pressure ($\mathbf{p_1}$) and final pressure ($\mathbf{p_3}$) are fixed, the value of intermediate pressure ($\mathbf{p_2}$) can be determined by differentiating the above equation of work input in terms of $\mathbf{p_2}$ and equating it to zero.

Let, $\frac{n}{n-1} p_1 V_1 = k\ (constant)$ and $\frac{n-1}{n} = a$

then,

$$W = k\left[\left(\frac{p_2}{p_1}\right)^a + \left(\frac{p_3}{p_2}\right)^a - 2\right]$$

or

$$W = k(p_2^a p_1^{-a} + p_3^a p_2^{-a} - 2) \quad \text{-------------------- (1)}$$

Differentiating the above equation (1) with respect to p_2 and equating it to zero,

$$\frac{dW}{dp_2} = k\ a\ p_2^{a-1} p_1^{-a} + k\ (-a) p_3^a p_2^{-a-1} = 0$$

$$k\ a\ \frac{p_2^a}{p_2 p_1^a} - k\ a\ p_3^a \frac{1}{p_2^a\ p_2} = 0$$

or

$$\frac{k\ a\ p_2^a}{p_2\ p_1^a} = \frac{k\ a\ p_3^a}{p_2\ p_2^a}$$

$$\left(\frac{p_2}{p_1}\right)^a = \left(\frac{p_3}{p_2}\right)^a$$

or

$$\frac{p_2}{p_1} = \frac{p_3}{p_2}$$

or

$$=> \quad p_2^2 = p_1 p_3$$

Intermediate pressure, $\mathbf{p_2} = \sqrt{\mathbf{p_1 p_3}}$

Thus for maximum efficiency the intermediate pressure is the geometric mean of the initial and final pressures.

3.17 Minimum work input for multistage compression with perfect intercooling:

Work input for a two-stage compressor with perfect intercooling is given by

$$W = \frac{n}{n-1} \mathrm{p_1 V_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{----------------------- (1)}$$

Work input will be minimum if $\frac{p_2}{p_1} = \frac{p_3}{p_2}$-----------------------------(2)

$$p_2^2 = p_1\ p_3$$

Dividing both sides by p_1^2 ,

$$\left(\frac{p_2}{p_1}\right)^2 = \frac{p_3}{p_1}\frac{p_2}{p_1} \Rightarrow \left(\frac{p_3}{p_1}\right)^{1/2}$$ ------------------------------- (3)

From (2), $\frac{p_3}{p_2} = \frac{p_2}{p_1} = \left(\frac{p_3}{p_1}\right)^{1/2}$ -------------------------------------- (4)

Substituting the equation (4) in equation (1), work input for a two stage compressor,

$$W_{min} = \frac{n}{n-1}\mathrm{p}_1\mathrm{V}_1\left[\left(\frac{p_3}{p_1}\right)^{½\left[\frac{n-1}{n}\right]} + \left(\frac{p_3}{p_1}\right)^{½\left[\frac{n-1}{n}\right]} - 2\right]$$

$$= \frac{n}{n-1}\mathrm{p}_1\mathrm{V}_1\left[2\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 2\right]$$

$$W_{min} = \frac{2n}{n-1}\mathrm{p}_1\mathrm{V}_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

or

$$W_{min} = \frac{2n}{n-1}m\mathrm{RT}_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

For a three stage compressor,

$$W_{min} = \frac{3n}{n-1}\mathrm{p}_1\mathrm{V}_1\left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{3n}} - 1\right]$$

or

$$W_{min} = \frac{3n}{n-1}m\mathrm{RT}_1\left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{3n}} - 1\right]$$

Generally, the minimum work input for a multistage reciprocating air compressor with x number of stages is given by,

$$W_{min} = \frac{xn}{n-1}p_1\mathrm{V}_1\left[\left(\frac{p_{x+1}}{p_1}\right)^{\frac{n-1}{xn}} - 1\right]$$

Minimum work input required for a two stage reciprocating air compressor with perfect intercooling is given by,

$$W_{min} = \frac{2n}{n-1}p_1\mathrm{V}_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right] kJ$$

But,from equation (4), $\left(\frac{p_3}{p_1}\right)^{1/2} = \frac{p_2}{p_1}$

Therefore,

$$\boldsymbol{W_{min} = \frac{2n}{n-1}p_1\mathrm{V}_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] kJ}$$

So, for maximum efficiency ie., for minimum work input, the work required for each stage is same.

For maximum efficiency, the following conditions must be satisfied:

1. The air is cooled to the initial temperature between the stages (Perfect cooling between stages).
2. In each stage, the pressure ratio is same. $\left(\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \cdots\right)$
3. The work input for each stage is same.

3.18 Rotary compressors:

Rotary compressors have a rotor to develop pressure. They are classified as
(1) Positive displacement compressors and (2) Non positive displacement (Dynamic) compressors

In positive displacement compressors, the air is trapped in between two sets of engaging surfaces. The pressure rise is obtained by the back flow of air (as in the case of Roots blower) or both by squeezing action and back flow of air (as in the case of vane blower). Example: (1) Roots blower, (2) Vane blower, (3) Screw compressor.

In dynamic compressors, there is a continuous steady flow of air. The air is not positively contained within certain boundaries. Energy is transferred from the rotor of the compressor to the air. The pressure rise is primarily due to dynamic effects.
Example: (1) Centrifugal compressor, (2) Axial flow compressor.

3.18.1 Roots blower:

The Roots blower is a development of the gear pump.

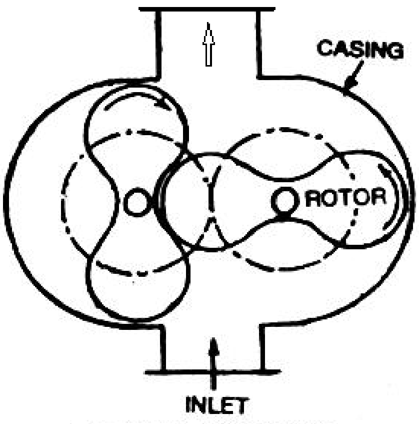


Fig:3.11 Roots blower

Construction: It consists of two lobed rotors placed in separate parallel axis of a casing as shown in fig:4.11. The two rotors are driven by a pair of gears (which are driven by the prime mover) and they revolve in opposite directions. The lobes of the rotor are of cycloid shape to ensure correct mating. A small clearance of 0.1 mm to 0.2 mm is provided between the lobe and casing. This reduces the wear of moving parts.

Working: When the rotor is driven by the gear, air is trapped between the lobes and the casing. the trapped air moves along the casing and discharged into the receiver. There is no increase in pressure since the flow area from entry to exit remains constant. But, when the outlet is opened, there is a back flow of high pressure air in the receiver. This creates the rise in pressure of the air delivered.These types of blowers are used in automobiles for supercharging.

3.18.2 Vane blower:

Construction: A vane blower consists of (1) a rotor, (2) vanes mounted on the rotor, (3) inlet and outlet ports and (4) casing. The rotor is placed eccentrically in the outer casing. Concentric vanes (usually 6 to 8 nos.) are mounted on the rotor. The vanes are made of fiber or carbon. Inlet suction area is greater than outlet delivery area.

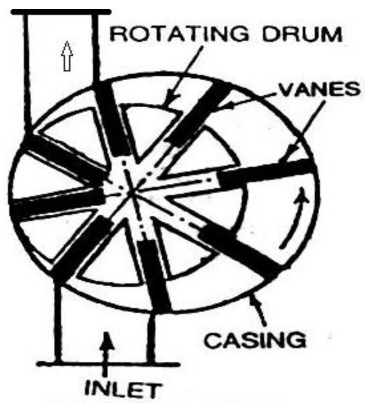


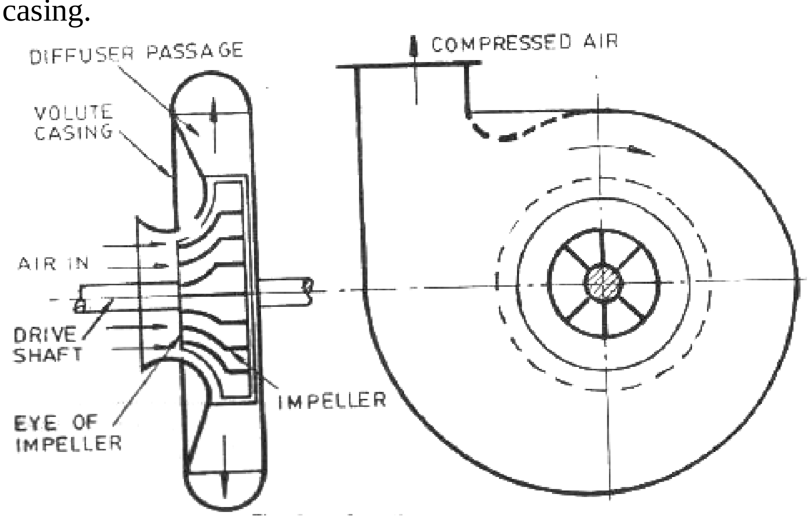
Fig: 3.12 Vane blower

Working: When the rotor is rotated by the prime mover, air is entrapped between two consecutive vanes. This air is gradually compressed due to decreasing volume between the rotor and the outer casing. This air is delivered to the receiver. This partly compressed air is further increased in pressure due to the back flow of high pressure air from the receiver.

Advantages: 1. Very simple and compact, 2. High efficiency 3. Higher speeds are possible

3.18.3 Centrifugal compressor

Construction: It consists of an impeller, a casing and a diffuser. The impeller consists of a number of blades or vanes, is mounted on the compressor shaft inside the casing. The impeller is surrounded by the casing.



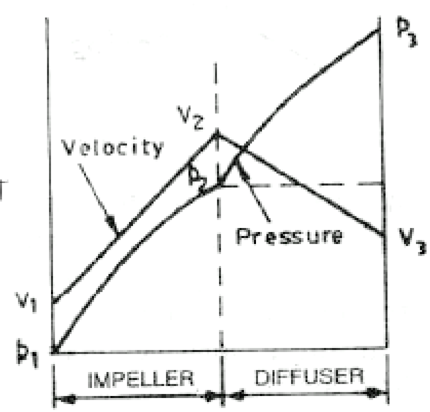


Fig: 3.13 Centrifugal compressor **Fig:3.14 Pressure – velocity Plot**

Working: In this compressor air enters axially and leaves radially. When the impeller rotates, air enters axially through the eye of the impeller with a low velocity. This air moves over the impeller vanes. Then, it flows radially outwards from the impeller. The velocity and pressure increases in the impeller. The air then enters the diverging passage known as diffuser. In the diffuser, kinetic energy is converted into pressure energy and the pressure of the air further increases. It is shown in fig:4.14. Finally, high pressure air is delivered to the receiver. Generally half of the total pressure rise takes place in the impeller and the other half in thediffuser.

Applications: Centrifugal compressors are used for low pressure units such as for refrigeration, supercharging of internal combustion engines, etc.

3.18.4 Axial flow compressor

In this air compressor, air enters and leaves axially.

Construction: It consists of two sets of blades: Rotor blades and stator blades. The blades are so arranged that the unit consists of adjacent rows of rotor blades and stator blades as shown in fig:4.15. The stator blades are fixed to the casing. The rotor blades are fixed on the rotating drum. The drum is rotated by a prime mover through a driving shaft. Single stage compressor consists of a row of rotor blades followed by a row of stator blades. Compression of air takes place in each pair of blades (one rotor blade and one stator blade). Hence there are many stages of compression in this type of compressor.

Working: When the switch is switched on, the prime mover rotates the drum. Air enters through the compressor inlet and passes through the rotor and stator blades. While passing through the blades, the air is compressed between the blades. The air is also compressed between the casing and the blades. The air flow passage area is gradually reduced from the inlet to the outlet of the compressor. This increases the pressure of the air considerably. Finally, high pressure air is delivered to the receiver.

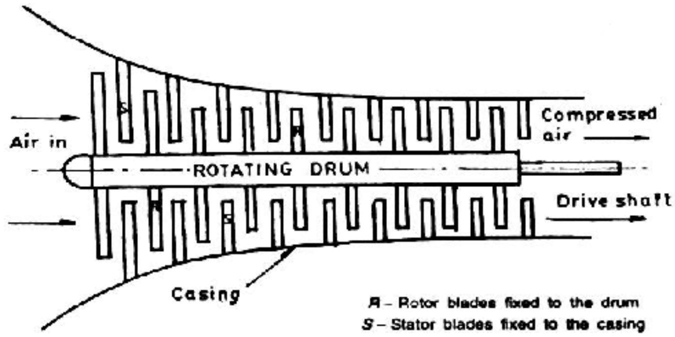


Fig:3.15 Axial flow compressor

Applications:

1. They are widely used in high pressure units such as industrial and marine gas turbine plants,
2. They are most suitable for aircraft work (Jet propulsion) since they require less frontal area

3.19 Comparison of Reciprocating and Rotary compressors

Reciprocating compressors	Rotary compressors
1. It is suitable for low rates of flow. Flow rate is limited to m^3/s	It is suitable for large rates of flow. Flow rate can be as large as 50 m^3/s.
2. It is used for high pressure rise. It can compress fluids up to 1000 bar.	It is used for medium pressure rise. The pressure rise is limited to 10 bar.
3. It cannot be coupled to turbines or I.C. engines.	It can be directly coupled to turbines or high speed internal combustion engines due to their higher speeds.
4. The flow of air is intermittent.	It gives uniform delivery of air.
5. The criterion of thermodynamic efficiency is isentropic.	The criterion of thermodynamic efficiency is isothermal.
6. Due to sliding parts it requires more lubrication.	No sliding parts. Hence needs lesser lubrication. It gives clean supply of air.
7. Maintenance cost is high because of large number of reciprocating parts.	Maintenance cost is less.
8. Complicated construction. It has more number of parts.	Simple in construction. It has less number of parts.
9. Torque is not uniform.	Uniform torque.

3.20 Free Air Delivery(FAD): It is the volume of air drawn into a compressor from the atmosphere. After compression and cooling the air is returned to the original temperature but it is at a higher pressure. Suppose atmospheric conditions are p_a ,T_a and V_a (the FAD) and the compressed conditions are p, Vand T.

Applying the gas law we have

$$\frac{pV}{T} = \frac{p_a V_a}{T_a}$$

$$V_a = \frac{pVT_a}{Tp_a} = F.A.D.$$

1. A Single cylinder, single acting air compressor has cylinder diameter 160mm and stroke length 300mm. It draws air into its cylinder at pressure of 100kpa at 27°C. The air is then compressed to a pressure of 650kpa. If the compressor runs at a speed of 2 rev/sec, Determine.

i) Mass of air compressed per cycle

ii) Work required per cycle

iii) Power required to derive the compressor in kW

Assume the compression process follows PV = constant.

Given data:

$D = 160mm = 0.16m$

$L = 300mm = 0.3m$

$P_1= 100kpa$

$T_1= 27°C= 27+ 273= 300K$

$P_2= 650kpa$

$N= 2rev/sec = 120rpm$

$PV^{\gamma}= C \quad \gamma= 1.4$

Solution:

Work done during Isothermal Compression (PV = C)

$W = mRT_1 \ln [P_2/ P_1]$

$W = P_1V_1 \ln [P_2/ P_1]$ [PV = mRT]

We know that,

$Vs=(\pi/4)D^2L =(\pi/4) *(0.16)^2*0.3$

$Vs=6.03X10^{-3}m^3 = V_1$ [clearance volume is neglected]

$Vs=6.03X10^{-3}m^3$

Substituting V_1 in work done equation

$W=100 X 6.03 X10^{-3} X \ln [650/100]$

$W= 1.13kJ$

Power = [W*N/60] = 1.13*120/60

P = 2026kW

We know that,

$P_1V_1= mRT_1$

$m = P_1V_1/R T_1 = [(100*6.03x10^{-3})/(0.287*300)]$

$m = 0.007kg$

Result:

i. m = 0.007kg

ii. $W=1.13kJ$
iii. $P = 2.26\ kW$

2. A Single cylinder, single acting reciprocating air compressor with a bore of 12cm and stroke of 16cm runs at 410rpm. At the beginning of compression, the pressure and temperature in the cylinder are 0.98bar and 40°C. the delivery pressure is 6bar. The index of compression is 1.32. the clearance is 6% of stroke volume. Determine the volume of air delivered referred to 1bar and 20°C. what is the power required?

Given data:

$D = 12cm = 0.12m$
$L= 16cm = 0.16m$
$N = 410rpm$
$P_1= 0.98\ bar= 98kpa$
$T_1= 40°C = 313K$
$P_2= 6bar = 600kpa$
$N = 1.32$
$Vc=6\%=0.06Vs$
$Po=1bar=100kpa$
$To=20°C=293K$

Solution:

We know that,

$Vs=(\pi/4)D^2L =(\pi/4) *(12)^2*16$
$Vs= 0.0018m^3$

We know that,

$V_1=Vc+Vs$
$V_1= 0.06Vs+Vs$
$V_1=1.06x0.0018$
$V_1=1.908x10^{-3}\ m^3$

Workdone on the single stage compressor with clearance volume

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

We know that,

$P_3V_3^n= P_4V_4^n$
$[V_4/ V_3]^n=[P_3/ P_4]$
$[V_4/ V_3]^n=[P_2/ P_1]$
$[V_4/ V_c]^n=[P_2/ P_1]$
$[V_4/ V_c]=[P_2/ P_1]^{1/n}$
$V_4 = V_c\ x[P_2/ P_1]^{1/n}$
$=0.06xVs[600/98]^{1/1.32}$
$=0.06x0.0018x[600/98]^{1/1.32}$
$V_4=4.26x10^{-4}m^3$

We know that,

$Va =V_1- V_4 = 1.908x10^{-3}- 4.26x10^{-4}$
$Va = 0.00148\ m^3$

Substituting Va value in work done equation

$W=[1.32/1.32-1]x98x0.00148[(600/98)^{1.32-1/1.32}-1]$
$W = 0.329\ kJ$

Power = WxN/60 = (0.329x410)/60

P = 2.25 kW

We know that,

$PoVo/To = P_2V_d/T_2$

$Vo = To/Po \times P_2V_d/T_2$

We know that,

$T_2/T_1=[P_2/P_1]^{n-1/n}$

$T_2=T_1x[P_2/P_1]^{n-1/n}$

$T_2=313x[600/98]^{1.32-1/1.32}$

T_2=485.6K

$[V_2/V_1]^n= P_1/P_2$

$V_2/V_1=[P_1/P_2]^{1/n}$

$V_2 = V_1 [P_1/P_2]^{1/n}$

$V_2 = 1.908x10^{-3} [98/600]^{1/1.32}$

$V_2 = 0.00048\ m^3$

We know that,

$V_d=V_2- V_3= V_2- V_c$=0.00048-(0.06x0.0018)

V_d = 0.000372m^3

Sub, To, Po, P_2, T_2, V_d values in ..(1)

Vo= (293/100)x (600/485.6)x0.000372

Vo = 0.0013 m^3

Result:

P = 2.25 kW

Vo = 0.0013 m^3

3. A single stage reciprocating compressor receives air at 25m^3/min at 1 bar, 15°C and discharges it at 15 bar. Assume the value of n for compression as 1.35 and volumetric efficiency as 0.75. determine i) theoretical power required ii) piston displacement per min ii) maximum air temperature. [Dec 2003]

Given data:

V_a =24m3/min

P_2 =15 bar =1500kpa

P_1 = 1 bar =100kpa

N = 1.35

T_1 = 15°C

η_{vol} = 0.75

Solution :

work done on the single stage compressor with clearance volume,

$W = n/n-1P_1 V_a [(P_2/P_1)^{n-1/n} -1]$

W = 9816.04 KJ/min = 163.6Kj/s

P = 163.6 KW

We know that ,

$\eta_{vol}= V_a/ V_s$

$0.75 = 25/ V_s$

V_s= 33.33 m3/min

We know that,

$T_2/T_1=[P_2/P_1]^{n-1/n}$

$T_2= T_1 x[P_2/P_1]^{n-1/n}$

$T_2 = 288 \text{ x}[\ 1500/100]^{1.35-1/1.35}$

$T_2 = 581.17K$

Result:

P = 163.6 KW

$V_s = 33.33$ m3/min

$T_2 = 581.17K$

4. A single stage reciprocating air compressor takes 1 m^3 of air per minute at 1bar and 15°C and delivers it at 7bar. The law of compression is $PV^{1.3}$== constant. Calculate the indicated power neglect clearance. If the speed of compressor is 300rpm and stroke to bore ratio is 1.5, calculate the cylinder dimensions. Find the power required if the mechanical efficiency of compressor is 85% and motor transmission efficiency is 90%

Given data:

$V_1 = 1m^3/min$

$P_1 = 1bar = 100kpa$

$T_1 = 15°C = 288K$

$P_2 = 7bar = 700kpa$

N=300rpm

L/D=1.5

$\eta_{\ mech} = 85\%$

motor efficiency= 90%

$PV^{1.3} = C$

Solution:

We know that, Workdone during polytropic compression

$W = (n/n\text{-}1)P_1\ V_a\ [\ (\ P_2/P_1)\ ^{n-1/n}\ \text{-}1]$

$W = (1.3/1.3\text{-}1)\text{x}100\text{x}1\text{x}[\ (\ 700/100)\ ^{1.3-1/1.3}\text{-}1]$

= 244.6kJ/min

Indicated Power = 4.07kW

We know that,

Stroke volume, $Vs = V_1 = (\pi/4)D^2L$

$1/300 = (\pi/4)\text{x}D^2\text{x}1.5D$

$1/300 = (\pi/4)\text{x}1.5D^3$

D = 0.141m

L= 1.5x0.141

L = 0.212m

Weknow that,

$\eta_{\ mech}$= (Indicated power/ Power input)

Power input = 4.07/0.85

Power input = 4.79kW

Motor efficiency = power input/ motor power

Motor power = 4.79/ 0.90

Motor power = 5.32kW

Result:

Indicated power = 4.07kW

Power input = 4.79kW

Motor power = 5.32kW

5. The free air delivered of a single cylinder single stage reciprocating air compressor 2.5 m^3/min. The ambient air is at STP conditions and delivery pressure is 7bar. The clearance volume is 5% of the stroke volume and law of compression and expansion is $PV^{1.25}=C$. if L= 1.2D and the compressor runs at 150rpm, determine the size of the cylinders.

Given data:

$V_a = 2.5\ m^3/min = 0.04166\ m^3/sec$

For STP condition, the pressure and temperature are

$V_1 = 1 m^3/min$

$P_1 = 1.013 bar = 101.3 kpa$

$T_1 = 15°C = 288K$

$P_2 = 7 bar = 700 kpa$

N=150rpm

L=1.2D

Vc=5%Vs = 0.05Vs

$PV^{1.25} = C$

n = 1.25

Solution:

The mass of free air delivered per second is given by

$m_a = PV/RT = (1.013x10^5 x0.04166)/(287x288) = 0.051 kg/sec$

We know that,

Work done, $W = (n/n-1)PV_a [(P_2/P_1)^{n-1/n} -1]$

$W = m_aRT(n/n-1) [(P_2/P_1)^{n-1/n} -1]$

$W = 0.054x0.287x288x (1.25/1.25-1) [(700/101.3)^{1.25-1/1.25} -1]$

W = 9.95kW

We know that,

Indicated power, IP =PmLAN/1000

$Pm = (n/n-1) P_1 x\eta_{vol}[(P_2/P_1)^{n-1/n} -1]$

But, $\eta_{vol} = 1+C-C (P_2/P_1)^{1/n}$

Where C = Vc/Vs

$\eta_{vol} = 1+ (Vc/Vs) – (Vc/Vs) (P_2/P_1)^{1/n}$

$\eta_{vol} = 1+ (0.05) – (0.05) (700/101.3)^{1/1.25}$

$\eta_{vol} = 0.815$

Substituting Pm value in eqn (2)

$Pm = (1.25/1.25-1) x1 x 0.815 x [(700/101.3)^{1.25-1/1.25} -1]$

Pm =1.923bar

Substituting Pm value in eqn (1)

Indicated Power IP (or) work out put

$1.95 = [1.923x10^5 x 1.2 D x (\pi/4)D^2 x 150/60] / 1000$

D = 0.28m

Result:

L = 1.2 D = 1.2x 0.28 = 0.336m

D = 0.28 m

L = 0.336m

6. A single stage double acting compressor has a free air delivery (FAD) of $14m^3$/min measured at 1.013bar and 15°C. the pressure and temperature in the cylinder during induction are 0.95bar and 32°C respectively. The delivery pressure is 7bar and index of compression and expansion, n=1.3. the clearance volume is 5% of the swept volume. Calculate the indicated power required and the volumetric efficiency.

Given data:

V_0=1**4**m^3/min = 0.233 m^3/sec
P_1= 0.95bar=95kpa
P_2= 7bar = 700kpa
T_1=32°C=305K
T_0=15°C=288K
P_0=1.013bar = 101.3kpa
Vc=5%Vs=0.05Vs   Vc/Vs= 0.05
n = 1.3

Solution:

Volumetric efficiency, $\eta_{vol}=1+C-C\,(P_2/P_1)^{1/n}$

$\eta_{vol}=1+(Vc/Vs)-(Vc/Vs)\,(P_2/P_1)^{1/n}$

$\eta_{vol}=1+(0.05)-(0.05)\,(700/95)^{1/1.3}$

$\eta_{vol}=0.818 = 81.8\ \%$

We know that,

$Po\ Vo/To = P_1\ V_1 / T_1$

$101.3\text{x}0.233/288 = 95\text{x}\ V_a/305$

$V_a = 0.263m^3/sec$

Work done or power,

$P = (n/n\text{-}1)P_1\ V_a\ [\ (P_2/P_1)^{n-1/n} -1]$

$P = (1.3/1.3\text{-}1)\text{x}95\text{x}0.263\ [\ (700/95)^{1.3-1/1.3} -1]$

P = 63.39 kW

Result:

$\eta_{vol}= 81.8\ \%$

Indicated power P = 63.39 kW

7. A single cylinder single acting reciprocating compressor takes in $6m^3$/min of air at 1bar and 15°C and compresses into 6 bar. Calculate the saving in the power required when the compression process in changed from adiabatic compression to isothermal compression

Given data:

V_1=6 m^3/min
P_1 =1 bar = 100kpa
T_1= 15° C = 288K
P_2= 6bar =600kpa

Solution:

Work done during isothermal compression (pv=c)
$W= P_1\ V_1 \ln[P_2/P_1]$
= 100*6*ln[600/100]
W = 1075.5kJ/min

Power , P = 17.91kW

Workdone during adiabatic process
W= [γ/γ-1] P_1 V_1 [(P_2/ P_1)$^{(\gamma-1/\gamma)}$-1]
W= [1.4/1.4-1] *100*6* [(600/ 100)$^{(1.4-1/1.4)}$-1]
W= 1403.87 kJ/min
P = 23.39kW
Saving power = 23.39-17.91
Saving power = 5.48 kW

8. Air is to be compressed in a single stage reciprocating compressor from 1.013bar and 15°C to 7bar. Calculate the indicated power required for a free air delivery of 0.3 m^3/min, when the compression process is i) Isentropic ii) polytropic with (n=1.45)

Given data:
P_1=1.013bar =101.3kpa
T_1=15°C = 288K
P_2=7bar=700kpa
Vo=0.3m^3/min
n=1.25

solution:
we know that, PoVo/To = P_1V_1/T_1
V_1= [PoVo/To] X [T_1/P_1] ………………. (1)
We know that, at atmospheric condition the pressure and temperature are
Po = 101.3kpa
To = 298 K
Substituting To,Po,Vo, P_1, V_1 values in eqn (1)
V_1 =[(101.3x0.3)/298]x[288/101.3]
V_1=0.289m^3/min
Work done duringisentropic Compression
W= [γ/γ-1] P_1 V_1 [(P_2/ P_1)$^{(\gamma-1/\gamma)}$-1]
W= [1.4/1.4-1] *101.3*0.289* [(700/ 101.3)$^{(1.4-1/1.4)}$-1]
W= 75.53kJ/min
W= 1.25kJ/s
P_{Iso}=1.25kW
Work done during polytropic compression
W= [n/n-1] P_1 V_1 [(P_2/ P_1)$^{(n-1/n)}$-1]
W= [1.25/1.25-1] x101.3x0.289x [(700/ 101.3)$^{(1.25-1/1.25)}$-1]
W = 69.08kJ/min
P_{poly}=1.15kW
Result:
P_{Iso}=1.25kWP_{poly}=1.15kW

9. Air enters a single stage double acting air compressor at 100kpa and 29°C. the compression ratio is 6:1. The speed of compression in 550rpm. The volume rate measured at suction condition is 5 m3/min. find the motor power required if the mechanical efficiency is 90%. If the volumetric efficiency is 80%. Find swept volume of cylinder.

Given data:
P_1=100kpa

T_1=29°C =302K
N=550rpm
V_1=5 m3/min
Compression ratio = 6:1
n = 1.3
η_{vol}=80%
η_{max}=90%

Solution:
Compression ratio =(total cylinder volume)/(clearance volume)= V_1/V_c
V_1/V_c =6
5/ V_c =6
V_c=0.833 m3/min
We know that,
$V_1= V_c+ V_s$
5= 0.833+ V_s
V_s=4.167 m3/kg
Workdone on the single stage compressor with clearance volume
W = (n/n-1)P_1 V_a [(P_2/P_1) $^{n-1/n}$ -1] ………(1)

Volumetric efficiency, η_{vol}=1+C-C (P_2/P_1) $^{1/n}$
C= V_c/V_s
η_{vol}=1+ (V_c/V_s)- (V_c/V_s) (P_2/P_1) $^{1/n}$
η_{vol}=1+ (0.833/ 4.167)- (0.833/ 4.167) (P_2/100) $^{1/1.3}$
P_2= 247.03kpa
We know that,
$\eta_{vol}= V_a/V_s$
0.8 = V_a/ 4.167
V_a=3.33 m^3/min
Applying V_a, P_2 values in eqn (1)
W = [1.3/1.3-1]x100x3.3 [(247.03/100) $^{1.3-1/1.3}$ -1]
W = 334.87kJ/min
W = 5.58kW
We know that,
Mech. efficiency =(power output of compressor)/(power supplied to compressor)
0.9 = (5.58)/ (power supplied to compressor)
Power Supplied To Compressor =6.2kW

Result:
V_s=4.167 m3/kg
Power Supplied To Compressor =6.2kW

10. A single stage single acting compressor delivers 15m^3 of free air per minute from 1bar to 8 bar. The speed of compressor is 300rpm. Assuming that compression and expansion follow the law $PV^{1.3}$ = constant and clearance is 1/16 th of swept volume, find the diameter and stroke of the compressor. Take L/D=1.5, the temperature and pressure of air at the suction are same as atmospheric air [Nov 2004]

Given data:
V_0=15m^3/min

P_1= 1bar=100kpa
P_2=8bar = 800kpa
N=300rpm
L=1.5D
$PV^{1.3}$= C
n = 1.3
L/D = 1.5

Solution:
We know that the volumetric efficiency
$\eta_{vol}=1 - (Vc/Vs)[(P_2/P_1)^{1/n}-1]$
$\eta_{vol}=1 - (1/16)[(8/1)^{1/1.3}-1]$
η_{vol}= 0.753 = 75.3%
We know that, free air delivered
Va = Vs x η_{vol} x 300
15 = Vs x 0.753 x300
Vs = 0.0664 m^3
Stroke volume = 0.0664m^3
We know that,
Vs = $(\pi/4)D^2L$ = 0.0664
$(\pi/4)D^2$x 1.5D = 0.0664
D = 0.3834 m
We know that,
L/D = 1.5
L =1.5 x 0.3834
L = 0.5751m

Unit -4 -
STEAM NOZZLES AND STEAM TURBINES – SMEA1404

Syllabus

Flow of steam through nozzles, Isentropic flow, ideal and actual expansion in nozzle, condition for maximum discharge, critical pressure ratio, Meta stable flow. Steam turbines, impulse and Reaction principles, Compounding, Velocity diagrams for impulse and reaction blades, Work done on turbine blades, optimization and efficiency.

STEAM NOZZLES AND TURBINES

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 superheated 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 (P2/P1) 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 respective to the temperature.

12. Convergent nozzle: The crossectional area of the duct decreases from inlet to the outlet side then it is called as convergent nozzle.

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

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.

Continuity and steady flow energy equations

Through a certain section of the nozzle: m.v = A.C m is the mass flow rate, v is thespecific 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 potentialenergies are negligible, w = 0 and q $\cong$ 0

$$h_1 + C_1^2/2 = h_2 + C_2^2/2$$

Types of Nozzles:

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

Convergent Nozzle:

A typical convergent nozzle is shown in fig. in a convergent nozzle, the cross sectionalarea 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.

Divergent Nozzle:

The cross sectional area of divergent nozzle increases continuously from its entrance toexit. It is used in a case, where the back pressure is less than the critical pressure ratio.

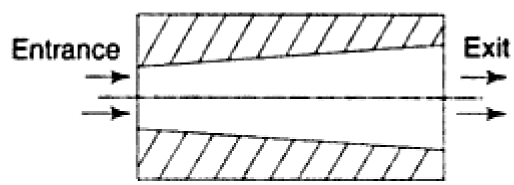


Fig. 4.1. Divergent Nozzle

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 iswidely used in many type of steam turbines.

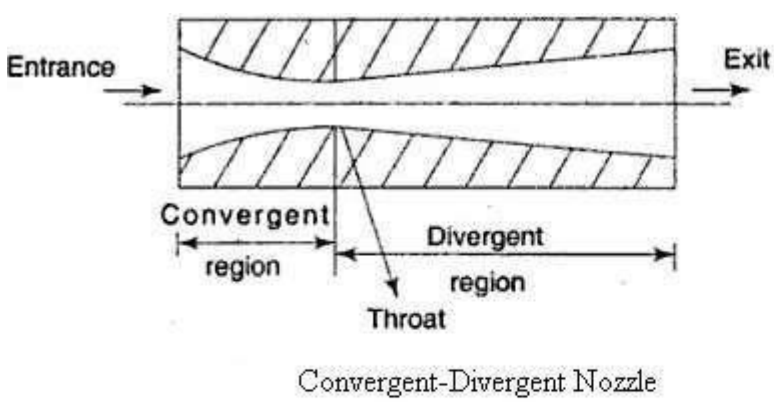
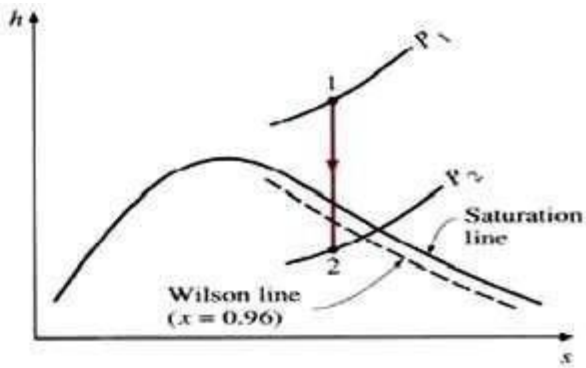


Fig. 4.2. Convergent-Divergent Nozzle

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:

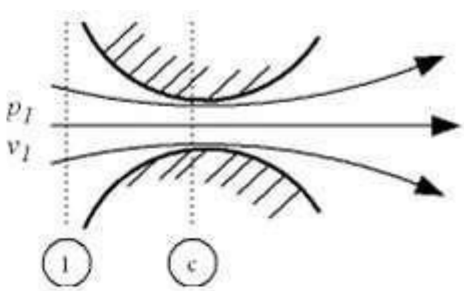


The h-s diagram for the isentropic expansion of steam in a nozzle.

Fig. 4.3 Steam isentropic expansion

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.



Critical flow nozzles

Fig. 4.4 Critical flow

The ratio between the critical pressure and the initial pressure for a nozzle can 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 polytrophic constant

For a perfect gas undergoing an adiabatic process the index – n – is the ratio of specific heats k = cp / cv. There isno 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.

$$\therefore \qquad V_2 = \sqrt{2000(H_1 - H_2)} = 44.72\sqrt{(H_1 - H_2)} \text{ m/s}$$

If frictional losses are taken into account then

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

Mass of steam discharged through 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 $\quad \frac{P_2}{P_1}=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$

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 P1 is the initial pressure of the steam in kpa and v1 is the specific volume of the steam in m3/kg at the initialpressure.

STEAM TURBINES: Normally the turbines are classified into types,

1. Impulse Turbine

2. Reaction Turbine

Impulse and Reaction Turbines:

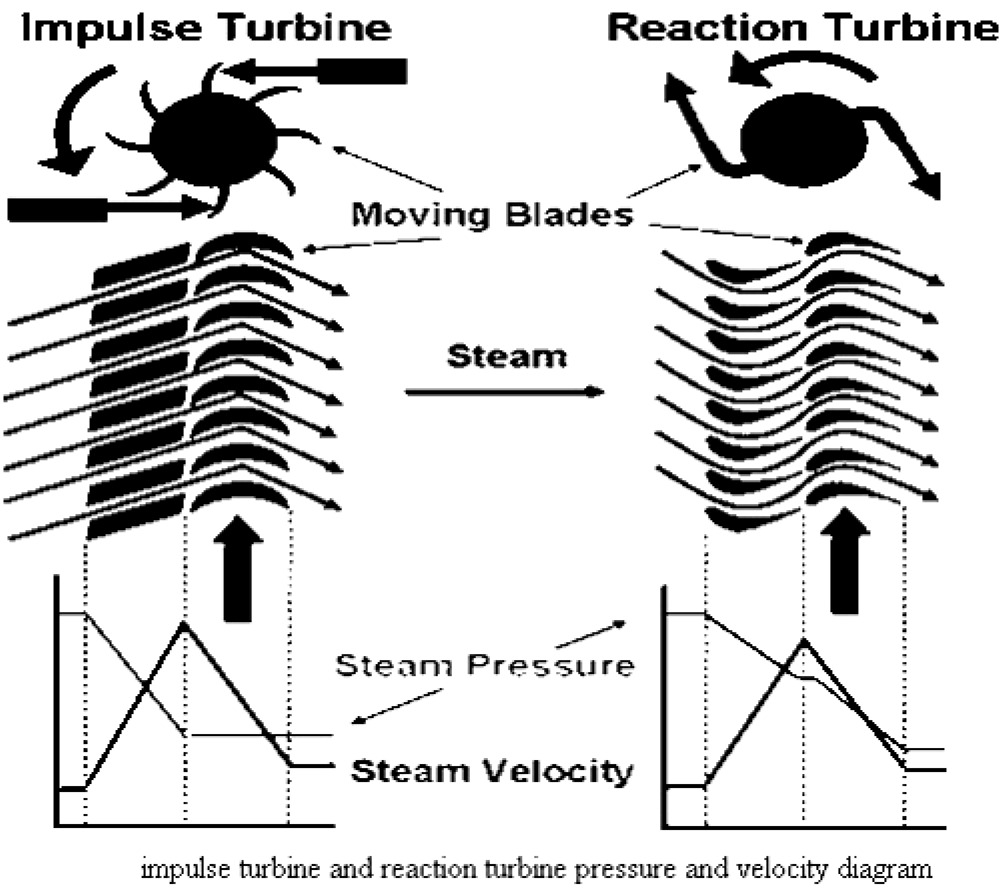


Fig. 4.5 Impulse and Reaction Turbines

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

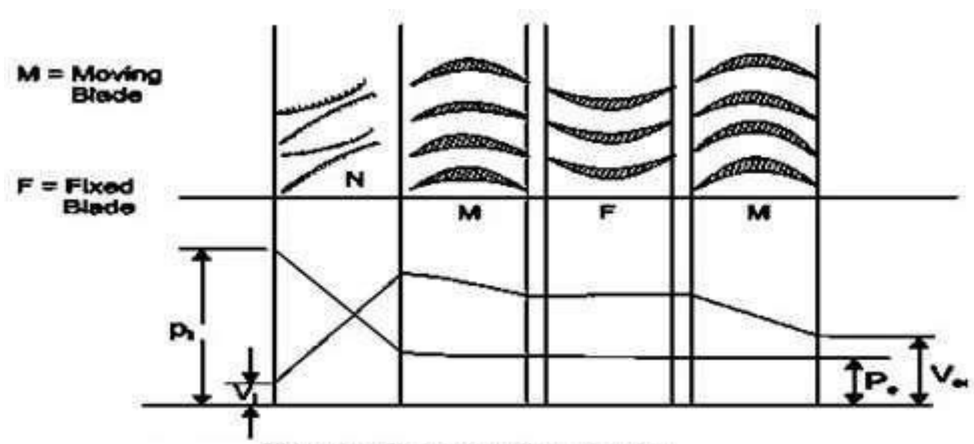
Reaction Turbines

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).

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.

Velocity Compounding:



Velocity Compounding diagram

Fig. 4.6. Velocity Compounding

Pi = Inlet Pressure, Pe= Exit Pressure, Vi =Inlet Velocity, Ve=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.

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.

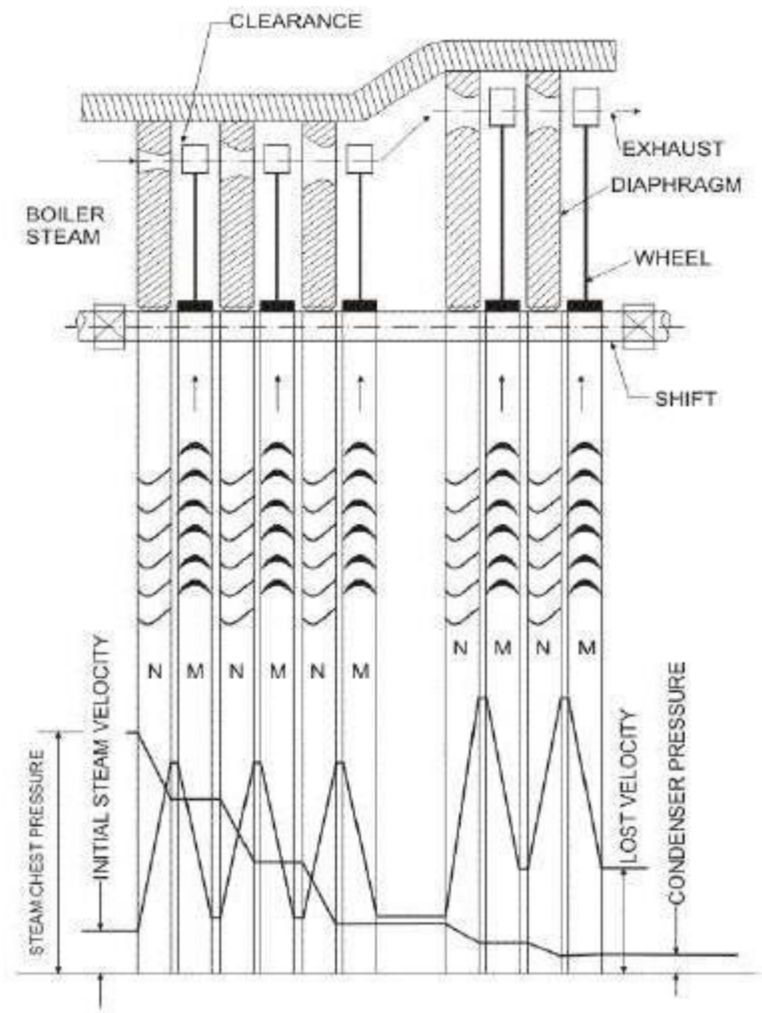


Fig. 4.7. Pressure Compounding

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 rowsof moving blades.

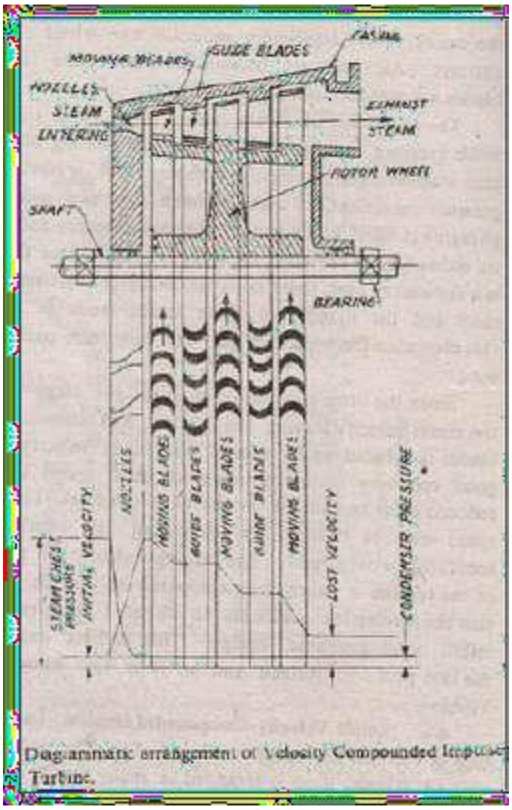


Fig 4.8 Pressure-Velocity Compounding

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

Velocity diagram of an impulse turbine:

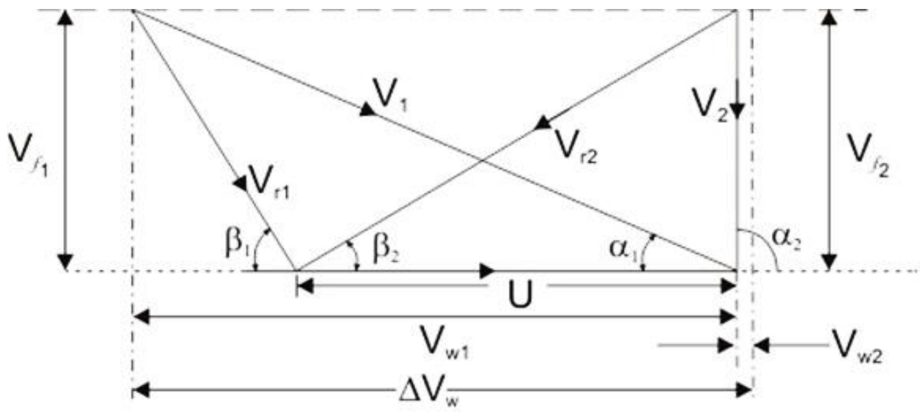


Fig 4.9 velocity diagram

V_1 and V_2 = Inlet and outlet absolute velocity

V_{r_1} and V_{r_2} = 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,

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

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,

$$\eta_b = \frac{2U\Delta V_w}{V_1^2}$$

stage efficiency $= \eta_s = \dfrac{\text{Work done by the rotor}}{\text{Isentropic enthalpy drop}}$

$$\eta_s = \frac{\dot{m} U \Delta V_w}{\dot{m}(\Delta H)_{isen}} = \frac{\dot{m} U \Delta V_w}{\dot{m}\left(\frac{V_1^2}{2}\right)} \cdot \frac{\dot{m}(V_1^2/2)}{\dot{m}(\Delta H)_{isen}}$$

or,

or, $\eta_s = \eta_b \times \eta_n \qquad [\eta_n = \text{Nozzle efficiency}]$

Optimum blade speed of a single stage turbine

$$\Delta V_w = V_{r1}\cos\beta_1 + V_{r2}\cos\beta_2$$

$$= V_{r1}\cos\beta_1 + \left(1 + \frac{V_{r2}}{V_{r1}} \cdot \frac{\cos\beta_2}{\cos\beta_1}\right)$$

$$= (V_1\cos\alpha_1 - U) + (1 + kc)$$

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

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

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

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

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

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

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

α_1 is of the order of 18^0 to 22^0

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

$\therefore$ The maximum value of blade efficiency

$$(\eta_b)_{\max} = 2(\rho\cos\alpha_1 - \rho^2)(1+kc)$$

$$= \frac{\cos^2\alpha_1}{2}(1+kc)$$

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

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

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

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

Velocity diagram of the velocity compounded turbines:

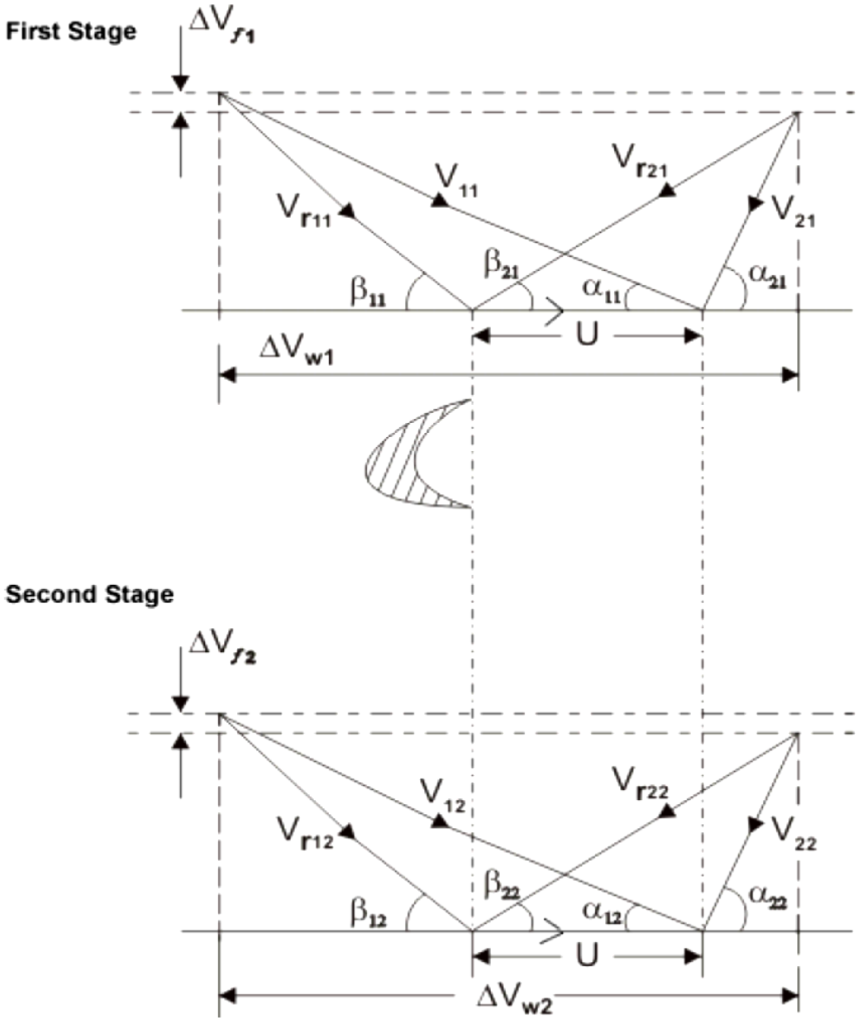
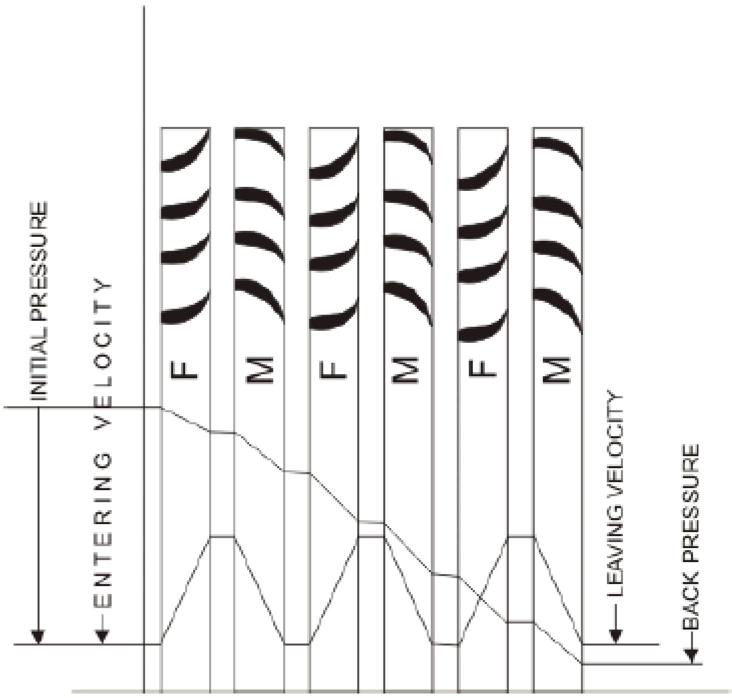


Fig 4.10 velocity diagram

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.

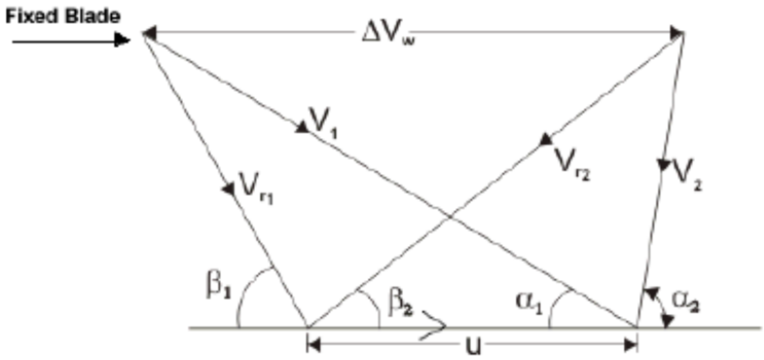


Pressure and enthalpy drop both in the fixed blade or stator and in the moving blade or **Rotor**

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

or,

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



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)

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

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

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

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

From the inlet velocity triangle we have,

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

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

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

Therefore, the Blade efficiency

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

Governing of Steam Turbine: The method of maintaining the turbine speed constant irrespective of the load is known as governing of tubines.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

Throttle Governing:

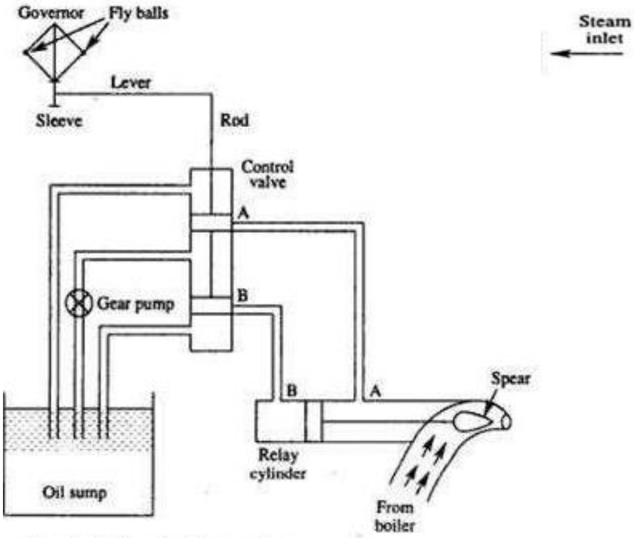


Fig 4 14 Throttle Governing

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 rely 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.

Nozzle Governing:

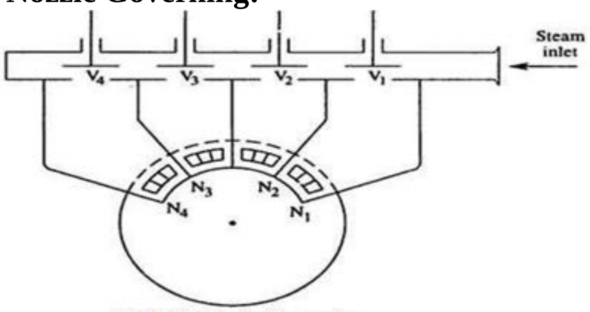


Fig 4 15 Nozzle Governing

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 180º 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.

1. Steam at 10.5 bar and 0.95 dryness is expanded through a convergent divergent nozzle. The pressure of steam leaving the nozzle is 0.85 bar. Find i) velocity of steam at throat for maximum discharge, ii) the area at exit iii) steam discharge if the throat area is 1.2cm2. assume the flow is isentropic and there are no friction losses. Take n= 1.135.
Given data:

P_1 = 10.5 bar

P_2 = 0.85 bar

Solution:

Area at throat A_t = 1.2 cm2

x_1 = 0.95

n = 1.135

solution:we know that, for n = 1.135

Throat pressure Pt = 0.577 x P1 = 0.577 x10.5 = 6.06 barproperties of steam from steam tables:

P1 = 10.5bar

hf = 772 KJ /kg

sf =2.159 KJ/kg

hfg = 2006 KJ/kg

sfg = 4.407 KJ/kg

Pt = 6.09 bar

hf = 673.25 KJ/kg

sf = 1.9375 KJ/kg

hfg = 2082.95 KJ/kg

sfg = 4.815 KJ/kg

vf = 0.01101 m3/kg

vg = 0.31556 m3 /kg

P2 = 0.85 bar

hf = 398.6 kJ/kg

hfg = 2269.8 kJ/kg

sf = 1.252 kJ/kgk

sfg = 6.163 kJ/kgk

vf = 0.001040 m3/kg

vg = 1.9721 m3/kg

s1= sf1 + x1 x sfg

= 2.159 +0.95 x 4.407 = 6.34565 kJ/kgk

h 1 = hf1 + x1 x hfg1

= 772 +0.95 x 4.407 = 6.34565 KJ/Kgk

1-t isentropic expansion between inlet and throat

S1 =sf = 6.34564 kJ/kg

St = sft + x_t x sfgt

6.34565 = 1.9375 + x_t x 4.815

X t = 0.915

h t = hft + x_t x hfgt

= 673.25 + 0.915 x 2082.95

= 2579.15 kJ/kg

Velocity of steam at throat:

Vt $=\sqrt{2000(\mathrm{h1}-\mathrm{ht})} = \sqrt{2000\ (2677.7-2579.15)}$

= 443.96 m/s

Vt = xt x vgt

= 0.915 x 0.31156 = 0.2887 m3 / kg

Mass of steam discharged:

= At x Vt/ v_t = 1.2x 10-4 x 443.96/0.28874

m = 0.1845 m3/kg

t-2 isentropic expansion between throat and exit

st = s2 = 6.34565 kJ/kgk

6.34565 = 1.252+x_2*6.162

X_2=0.83

V_2= $x_2 \times v_{g2}$ = 0.83 × 1.9721

= $1.637\frac{m^3}{kg}$

$h_2 = h_{f2} + x_2 \times h_{fg2}$

=398.6+0.83×2269.8

=2282.534 KJ/k

Velocity of steam at exit

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

$= \sqrt{200(2677.7 - 2282.534)}$

=889 m/sec

Accoding to mass balance , steam flow rate of throat is equal to flow rate at exit

$m_t = m_2$

$m_2 = \frac{A_2 \times v_2}{v_2}$

$0.1845 = A_2 \times 889/1.637$

$$A_2 = 3.397 \times 10^{-4} m^2$$

$$A_2 = 3.397 \times 10^{-4} cm^2$$

2. Dry saturated steam at 2.8 bar is expanded through a convergent nozzle to 1 .7 bar. The exit area is 3 cm^2. Calculate the exit velocity and mass flow rate for, i) isentropic expansion ii) supersaturated flow.

Given Data :

$P_1 = 2.8$ bar

$P_2 = 1.7$ bar

$A_2 = 3\ cm^2 = 3 \times 10^{-4} m^2$

Solution :

Properties of steam table

P_1 = 2.8 bar

$$h_1 = 2721.5\ KJ/kg$$

$s_1 = 7.014\ KJ/kgK$

$v_1 = 0.64600\ m^3/kg$

P_2 = 1.7 bar

$$h_f = 483.2 KJ/kg$$

$h_{fg} = 2215.6 KJ/kg$, $s_f = 1.475\ KJ/kgK$

$s_{fg} = 5.706 KJ/kgK$

$v_f = 0.001056\ m^3/kg$

$v_g = 1.0309\ m^3/kg$

For isentropic flow

$s_1 = s_2 = 7.014\ J/kgK$

$s_2 = s_{f2} + x_2 \times s_{fg2}$

$7.014 = 1.475 + x_2 \times 5.706$

$x_2 = 0.97$

$h_2 = h_{f2} + x_2 \times h_{fg2}$

$= 483.2 + 0.97 \times 2215.6$

$h_2 = 2634.152 KJ/kg$

$v_2 = x_2 \times v_{g2}$

$= 0.97 \times 1.0309 \quad = 1.00\ m^3/kg$

Velocity of steam at exit

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

$= \sqrt{200(2721.5 - 2631.15)}$

$\underline{V_2 = 418 m/sec}$

Mass flow rate at exit

$m_2 = \frac{A_2 \times v_2}{v_2}$

$= \frac{3 \times 10^{-4} \times 418}{1.00}$

$= 0.1257\ m^3/kg$

For super saturated flow

$$V_2 = \sqrt{\frac{2n}{n-1}} \times p_1 \times v_1 \left[1 - \left(1 - \frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$V_2 = \sqrt{\frac{2 \times 1.3}{1.3-1}} \times 2.8 \times 10^5 \times 0.6460 \quad \left[1 - \left(1 - \frac{1.7}{2.8}\right)^{\frac{1.3-1}{1.3}}\right]$$

$\underline{V2 = 413 m/sec}$

Mass flow rate at exit

$$m2 = \frac{A_2 \times v_2}{v_2} = \frac{3\times10^{-4}\times413}{0.94827}$$

= 0.1306 kg/sec.

3. Dry saturated steam at a pressure of 8 bar enters a C-D nozzle and leaves it a pressure of 1.5 bar. If the steam flow process is isentropic and if the corresponding expanding index is 1.135, Find the ratio of cross sectional area at exit and throat for maximum discharge.

Given Data:

$P_1 = 2.8$ bar

$P_2 = 1.7$ bar

n = 1.135

Solution:

We know that n= 1.135

Throat pressure $p_t = 0.577 \times p_1 = 0.577\times8 = 4.62$ bar

Properties of steam at steam table

At 8 bar

$h_1 = 2769.1$ KJ/kg

$s_1 = 6.6628$ KJ/kgK

$v_1 = 0.2404$ m³/kg

At 4.62 bar

$h_f = 626.7$KJ/kg

$h_{fg} = 2117.2$KJ/kg

$s_f = 1.829$ KJ/kgK

$s_{fg} = 5.018$KJ/kgK

$v_f = 0.001090$ m³/kg

$v_g = 0.40526$ m³/kg

At 1.5 bar

$h_f = 467.11$KJ/kg

$h_{fg} = 2226.5$KJ/kg

$s_f = 1.4336$ KJ/kgK

$$s_{fg} = 5.7897 KJ/kgK$$

$v_f = 0.001053\ m^3/kg$

$= 626.7+0.963\times2117.2$

$h_t=2666.18$ KJ/kg

$v_t = x_t \times v_{gt}$

$= 0.963\times040526 \quad =0.39\ m^3/kg$

Velocity of steam at throat

$V_t=\sqrt{2000(h_1-h_t)}$

$=\sqrt{200(2769.1-2666.18)}$

=477.749 m/sec

t-2 isentropic expansion

$s_t = s_2= 6.6628$ KJ/kgK

$s_2 = s_{f2} + x_2 \times s_{fg2}$

$6.6628 = 1.4336+ x_2 \times5.7897$

$x_2 = 0.903$

$v_2 = x_2 \times v_{g2}$

$=0.903 \times 1.1593= 1.04695\ m^3/kg$

$h_2 = h_{f2} + x_2 \times h_{fg2}$

$=467.11+0.903\times2226.5$

$h_2=2477.6395$ KJ/kg

Velocity of steam at exit

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

$=\sqrt{200(2769.1-2477.639)}$

=763.5 m/sec

According to mass balance

Mass flow rate of steam at throat = Mass flow rate of steam at exit

$$m_t = m_2$$

$$\frac{A_t \times V_t}{v_t} = \frac{A_2 \times V_2}{v_2}$$

$$\frac{A_2}{A_t} = \frac{1.04695 \times 477.749}{763.5 \times 0.39} = 1.68$$

4. Steam enters a group of CD nozzles at 21 bars and 270°C. The discharge pressure of the nozzle is 0.07 bars. The expansion is equilibrium throughout and the loss of friction in convergent portion of the nozzle is negligible, but the loss by friction in the divergent section of the nozzle is equivalent to 10% of the enthalpy drop available in that section. Calculate the throat and exit area to discharge 14 kg/sec of steam.

Given Data:

$P_1 = 21$ bar

$T_1 = 270°C$

$P_2 = 0.07$ bar

m = 14 kg / s.

since loss by friction is 10%.

The efficiency $\eta = 90\%$.

Solution :

Properties of steam (from Mollier Diagram)

$h_1 = 2980 \frac{KJ}{kg}$ (at 21 bar and 270°C)

Since the expansion is isentropic from

$h1 = \frac{2980kj}{kg}$ draw a vertical line in the moiller diagram up to 0.07bar pressure line now note the following values at that point.

h2 = 2052.213kj/kg(at 0.07bar)

v2 =16.1m3/kg

The critical pressure ratio when steam is initially super heated,

$pt\ /p_1 = 0.546$

Throat pressure

$pt = 0.546 *_{p1}$ =0.546*21=11.466bar

Properties of steam at throat

$v_t = \sqrt{}2000(h_{1-\ h_t}) = \sqrt{2000(2980 - 2805)} = 591.6 m/s$

Velocity of steam at exit

$$v_{2=\sqrt{2000(h_{t-h_2})}} * \eta = \sqrt{2000(2980 - 2052.213)} * 0.9 = \frac{1292.29m}{s}$$

Throat area of nozzle

$A_{2=}m*v_{2/}V_2$=(14*16.1)/1292.29=0.174m^2=1744.4cm^2

5.The following data refer to a single stage impulse turbine.

Isentropic nozzle entropy drop=200kj/kg
Nozzle efficiency=90%

Nozzle angle=25^0

Ratio of blade speed to whirl component of steam speed=0.5. blade coefficient =0.9. the velocity of steam entering the nozzle 30m/s. find(1).blade angles at the inlet and outlet if the steam enters the blade without shock and leaves the blade in the axial direction.(2). Blade efficiency (3).power developed (4).axial thrust if the steam flow rate is 10kg/s. [Nov 2003]

Given data:

$h_{t=}h_e$=200kJ/kg

η_N=90%

$\alpha=25^0$

$\frac{v_b}{v_{w1}}$=0.5

$\frac{vr2}{vr1}$=0.9

v_i=30m/s

$v_2 = v_{f2}$

v_{w2}=0

$\beta=90^0$ for axial discharge

Solution:

Actual enthalpy drop

$h_i - h_e = (h_i - h_e)\eta_N$

$h_i - h_e = 200 \times 0.9$

=180 KJ/kg

$V_e = \sqrt{2(h_i - h_e) + v_i^2} = \sqrt{2(1000-180) + 30^2}$

= 600.75 m/sec.

<u>Inlet velocity of steam to the turbine</u>

$v_1 = v_1 = 600 \frac{m}{sec}$

From triangle ABC , $v_{\omega 1} = v_1 \cos 25°$

$= 600.75 \cos 25°$

= 544.46 m/sec

$v_{f1} = v_1 \sin 25°$

$= 60.75 \times \sin 25° = 253.89$ m/sec

$v_b / v_{\omega 1} = 0.5$

$v_b = 0.5 \times 544.46 = 272.3$ m/sec

from triangle ACE $v_{r1} = \sqrt{[{v_{f1}}^2 + (v_{\omega 1} - v_b)^2]}$

$= \sqrt{[253.89^2 + (544.76 - 272.23)^2]}$

=372.25 m/sec

$\tan \frac{v_{f1}}{v_{\omega 1} - v_b}$

=253.89/(544.46-272.23)

Θ=43°

$v_{r2} = 0.9 \times v_{r1} = 0.9 \times 372.25 = 335.03$ m/sec

from triangle ABD,

$$\cos \varphi = \frac{AB}{AD} = \frac{v_b}{v_{r2}} = \frac{272.23}{335.03}$$

Φ=35°39·

$v_2 = \sqrt{\ } (v_{r2} - v_b)^2$

$v_2 = \sqrt{\ } (335.03^2 - 272.03^2)^2$

=195.28 m/sec

$v_f = v_2 = 195.28 \text{m/sec}$

Power developed P = m($V_{w1} + V_{w2}$) x V_b = 10(544.46 + 0) x 272.23

= 1482.18kW.

Blade efficiency:

$\eta_b = m (V_{w1} + V_{w2}) \times V_b / (1/20 (600.75)2 = 82.14\%$

Axial thrust

$Fy = m (V_{f1} – V_{f2}) = 10 (253.89 – 175.28)$

$Fy = 586.1N.$

6. Steam enters the blade row of an impulse turbine with a velocity of 600m/s at an angle of 25ºC to the plane of rotation of the blades. The mean blade speed is 250m/s. the plant angle at the exit side is 30º. The blades friction less is 10%. Determine

i) The blades angle at inlet

ii) The workdone per kg of steam

iii) The diagram efficiency

iv) The axial thrust per kg of steam per sec. [Nov 2003]

Given data:

$V_1 = 600m/s$

$\alpha = 25^o$

$V_b = 250$ m/s

$\emptyset = 30^o$

$V_{r2}/V_{r1} = 0.9$

Solution :

From Δ BCE,

$V_{w1} = V_1 \cos \alpha = 600 \cos 25^o = 543.79$m/s

$V_{f1} = V_1 \sin \alpha = 600 \sin 25^o = 253.\ 57$ m/s

From Δ ACE

$Tan\theta = V_{f1} / V_{w1}- V_b = 253.57/ 543.79- 250$

$$v_{r1} = \sqrt{[v_{f1}{}^2 + (v_{\omega 1} - v_b)^2}$$

$$v_{r1} = \sqrt{[253.57\ ^2 + (543.79\ \ - 250\ \ )^2}$$

=388.09 m/ sec

$v_{r2} = 0.9 \times v_{r1} = 0.9 \times 388.09$=349.28 m/sec

From Δ ADF

$V_b+v_{w2}=v_{r2} \cos 30°$

$250 + v_{w2} = 349.28 \cos 30°$

$v_{w2} = 52.49$ m/sec

$V_{f2} = Vr_2 \sin 30° = 349.28° \sin 30^0 = 174.64$ m/s

Work done $W = m(v_{w1}+v_{w2})\, v_b$

$W = 1(543.79+52.49) \times 250 = 149.07$ KW/kg.

Diagram efficiency$\eta_D = \frac{(vw1+vw2)\ vb}{mv_1{}^2/2}$

$= \frac{149.07 \times 1000}{1 \times 0.5 \times 600^2} = 82.82\%$

Axial thrust $F_y = m\,(v_{f1} - v_{f2})$

$= 1(253.57-174.64)$

$=79.73$ N/kg-sec

7. At a particular stage of a reaction turbine , the mean blade speed is 60 m/sec and the steam pressure is 3.5 bar with atemperature of 175°C . The identical fixed and moving blades have inlet angles 30° and outlet angle of 20°. Determine (i) The blade height if it is 1/10 of the blade ring diameter for a flow rate of 13.5 kg/sec.

(ii) The power developed by a pair

(iii) the specific enthalpy drop if the stage efficiency is 85%. [Apr 2004]

Given Data :

Mean blade speed vb= 60 m/sec

Steam pressure = 3.5 bar

Temperature = 175°C

For identical fixed and moving blade,

$\Theta=\beta=30°$, $\alpha=\varphi=20°$.

m=13.5 kg/sec.

$h = 1/10 \times d$

Solution ;

According to sine rule

ΔABC

$$\frac{v_1}{\sin 150} = \frac{v_{r1}}{\sin 20} = \frac{60}{\sin 10}$$

$V_{r1} = \frac{60}{\sin 10} \times \sin 20.$

= 118.2 m/sec

$v_{f1} = v_{r1} \times \sin 30° = 118.2 \times \sin 30°$

= 59.1 m/sec.

FA = $v_{r1} \times \cos 30° = 118.2 \times \cos 30°$

=102.4 m/sec.

$v_{w1}+v_{w2}$ = EA+AB+BF=102.4+60+102.4

= 264.8 m/sec.

Velocity flow at exit , $v_{f1} = 60$ m/sec.

Pressure of 3.5 bar and 175 ℃.

From steam table,

$V_{sup} = 0.73$ m^3/kg.

Mass of steam flow (m)

$13.5 = \frac{\pi(d+h)hV_{f1}}{V_{sup}} = \frac{\pi(10h+h)h\times 60}{0.73}$

13.5= 2838 h^2

h^2=13.5/2838

h=0.068m=68mm.

The power developed,

By a pair of fixed and moving blade rings

P = $m(v_{w1}+v_{w2})\, v_b$

= 13.5 (264.8)×60 = 214650W

=214.65 kW.

Heat Drop required for the efficiency of 85% Heat drop required

= 214.65/0.85 =252.52 kJ/sec.

UNIT – 5 - REFRIGERATION AND AIR CONDITIONING SYSTEMS – SMEA1404

Syllabus

Bell Coleman and vapour compression refrigeration systems, Performance calculations, vapour absorption refrigeration system, Air conditioning systems and types, working principle-Estimation of cooling load.

UNIT- 5 - REFRIGERATION CYCLES & REFRIGERANTS

5.1 INTRODUCTION

For specific applications, efficiencies of both living and non-living beings depend to a great extent on the physical environment. The nature keeps conditions in the physical environment in the dynamic state ranging from one extreme to the other. Temperature, humidity, pressure and air motion are some of the important environment variables that at any location keep changing throughout the year. Adaptation to these many a times unpredictable variations are not possible and thus working efficiently is not feasible either for the living beings or the non-living ones. Thus for any specific purpose, control of the environment is essential. Refrigeration and air-conditioning is the subject which deals with the techniques to control the environments of the living and non-living subjects and thus provide them comforts to enable them to perform better and have longer lives.

5.2 DEFINITIONS

Refrigeration:

Refrigeration is defined as a method of reducing the temperature of a system below that of the surroundings and maintaining it at the lower temperature by continuously extracting the heat from it.

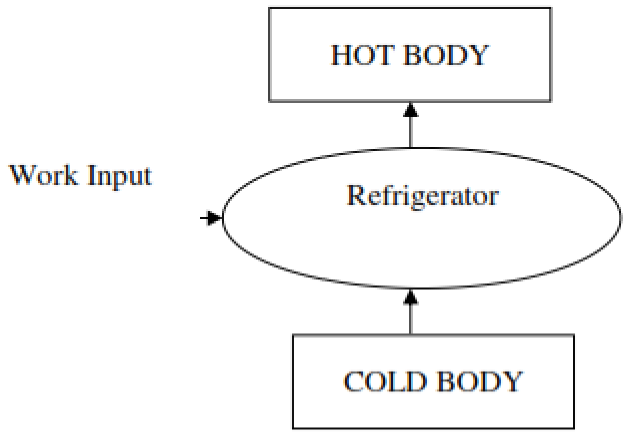


Fig 5.1

The principle of refrigeration is based on second law of thermodynamics. It states that heat does not flow from a low temperature body to a high temperature body without the help of an external work. In refrigeration process, the heat is continuously removed from a system at lower temperature and transfers it to the surroundings at a higher temperature. This operation according to second law of thermodynamics can only be performed by the aid of the external work. Therefore in a refrigeration system, power is to be supplied to remove heat continuously from the refrigerator to keep it cool at a temperature less than the surroundings. The refrigeration cycle is based on reversible Carnot cycle.

Refrigeration effect:

The rate at which the heat is absorbed in a cycle of from the interior space to be cooled is called refrigeration effect. It is defined as the quantity of heat removed to the time taken. It is also called as the capacity of a refrigerator.

Ton of Refrigeration (or) Unit of Refrigeration (TR):

The standard unit of refrigeration is *ton refrigeration* or simply*ton* denoted byTR. It is equivalent to the rate of heat transfer needed to produce1 ton (2000lbs) of ice at32^0F from water at 32^0Finoneday, i.e., 24hours.The enthalpy of solidification of water from and at32^0Fin British thermal unit is 144 Btu/lb. Thus

$$1 \text{ TR} = \frac{2000\,\text{lb} \times 144\,\text{Btu/lb}}{24\,\text{hr}}$$
$$= 12000 \text{ Btu/hr} = 200 \text{ Btu/min}$$

Ingeneral, 1TR means 200Btu of heat removal perminute.Thus if a refrigeration system is capable of cooling at the rate of 400 Btu/min, it is a 2 ton machine. A machine of 20 ton rating is capable of cooling ata rate of 20×200 = 4000Btu/min. This unit of refrigeration is currently in use in the USA, the UK and India. In many countries,the standard MKS unit of kcal/hr is used. In the MKS it can be seen that

$$1 \text{ TR} = 12000 \text{ Btu/hr} = \frac{12000}{3.968} = 3024.2 \text{ kcal/hr}$$
$$= 50.4 \text{ kcal/min} \approx 50 \text{ kcal/min}$$

If Btu ton unit is expressed into SI system, it is found to be **210 kJ/min or3.5 kW.**

Co-efficient of Performance (COP):

The Co-efficient of Performance is defined as the ratio of heat absorbed in a system to the work supplied.

The theoretical Coefficient of Performance (Carnot), (COP a standard measure of refrigeration efficiency of an ideal refrigeration system) depends on two key system temperatures: evaporator temperature T_e and condenser temperature T_c

COP is given as:$COP_{Carnot} = T_e / (T_c - T_e)$

This expression also indicates that higher COP_{Carnot} is achieved with higher evaporator temperatures and lower condenser temperatures. But COP is only a ratio of temperatures, and does not take into account the type of compressor. Hence the COP normally used in industry is calculated as follows:

$$COP = \frac{\text{Cooling effect (kW)}}{\text{Power input to compressor (kW)}}$$

Where the cooling effect is the difference in enthalpy across the evaporator and expressedas kW.

Ice making capacity:

It is the ability of the refrigeration system to make ice. In other words, it is the capacity of refrigeration system to remove heat from water to make ice.

Relative COP:

It is the ratio of actual COP to the theoretical COP of a refrigerator.Actual COP is measured during a test and theoretical COP is obtained by applying thelaws of thermodynamics.

5.3REFRIGERATOR AND HEAT PUMP

The vapor compression refrigeration cycle is a common method for transferring heat from a low temperature to a high temperature.

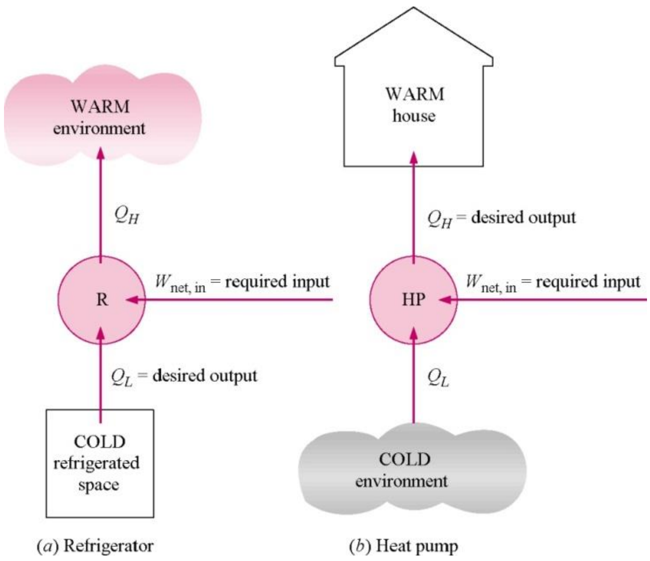


Fig 5.2

The above figure shows the objectives of refrigerators and heat pumps. The purpose of a refrigerator is the removal of heat, called the cooling load, from a low temperature medium. The purpose of a heat pump is the transfer of heat to a high temperature medium, called the heating load. When we are interested in the heat energy removed from a low temperature space, the device is called a refrigerator. When we are interested in the heat energy supplied to the high temperature space, the device is called a heat pump. In general, the term "heat pump" is used to describe the cycle as heat energy is removed from the low temperature space and rejected to the high temperature space.

The performance of refrigerators and heat pumps is expressed in terms of *coefficient of performance* (COP), defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

Both COP_R and COP_{HP} can be larger than 1. Under the same operating conditions, the COPs are related by

$$COP_{HP} = COP_R + 1$$

5.4 TYPES OF REFRIGERATION

Refrigeration is classified as based on working substance used

- Air refrigeration system (Bell-Coleman cycle)
- Water refrigeration system
- Ice refrigeration system
- Refrigeration by special fluid (low boiling point fluids – Refrigerants) (Reversed Carnot cycle)
    - Vapour compression refrigeration system (VCR)
    - Vapour absorbtion refrigeration system (VAR)
    - Vapour adsorbtion refrigeration system and etc.,

5.5Simple Vapour Compression Refrigeration System (VCR)

It consists of the following essential parts:

Compressor

The low pressure and temperature vapour refrigerant from evaporator is drawn intothe compressor through the inlet or suction valve A, where it is compressed to ahigh pressure and temperature. This high pressure and temperature vapourrefrigerant is discharged into the condenser through the delivery or dischargevalve B.

Condenser

The condenser or cooler consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed.The refrigerant, while passing through the condenser, gives up its latent heat to thesurrounding condensing medium which is normally air or water.

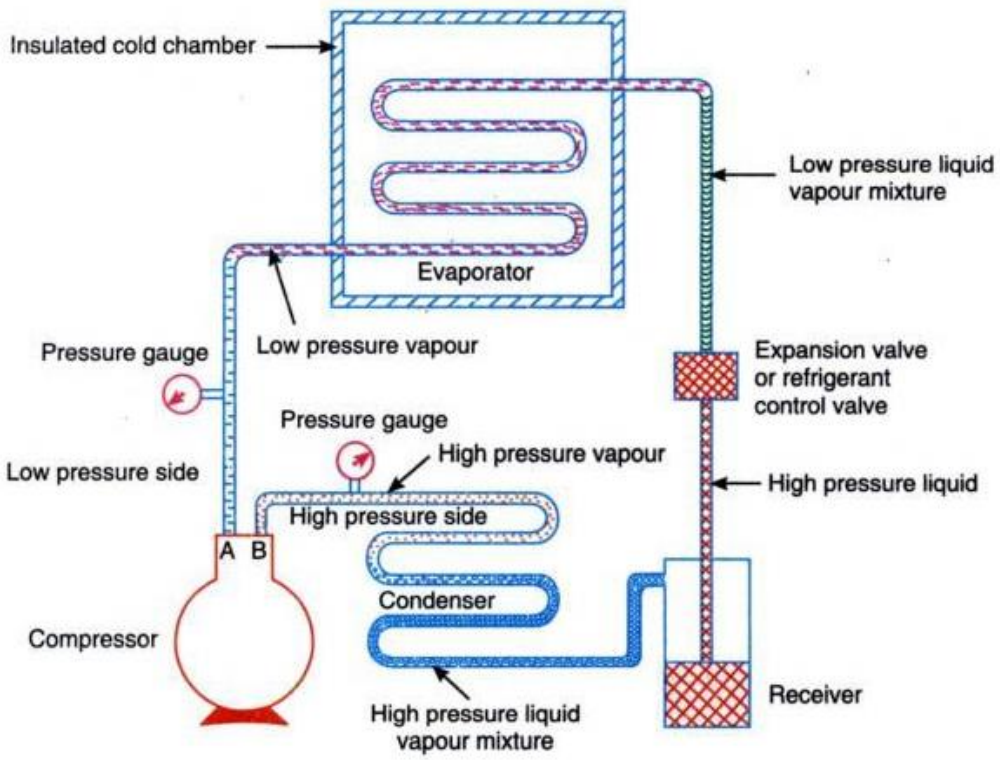


Fig 5.3

Receiver

The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from where it is supplied to the evaporator through the expansion valve orrefrigerant control valve.

Expansion Valve

It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure andtemperature to pass at a controlled rate after reducing its pressure and temperature.Some of the liquid refrigerant evaporates as it passes through the expansion valve,but the greater portion is vaporized in the evaporator at the low pressure andtemperature

Evaporator

An evaporator consists of coils of pipe in which the liquid-vapour. Refrigerant at low pressure and temperature is evaporated and changed into vapour refrigerant atlow pressure and temperature. In evaporating, the liquid vapour refrigerant absorbsits latent heat of vaporization from the medium (air, water or brine) which is to becooled.

The Simple Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle has four components: evaporator, compressor, condenser, and expansion (or throttle) valve. The most widely used refrigeration cycle is the *vapor-compression refrigeration cycle*. In an ideal or simple vapor-compression refrigeration cycle, the refrigerant enters the compressor as a saturated vapor and is cooled to the saturated liquid state in the condenser. It is then throttled to the evaporator pressure and vaporizes as it absorbs heat from the refrigerated space.

The ideal vapor compression cycle consists of four processes.

Ideal Vapor-Compression Refrigeration Cycle

Process	Description
1-2	Isentropic compression
2-3	Constant pressure heat rejection in the condenser
3-4	Throttling in an expansion valve
4-1	Constant pressure heat addition in the evaporator

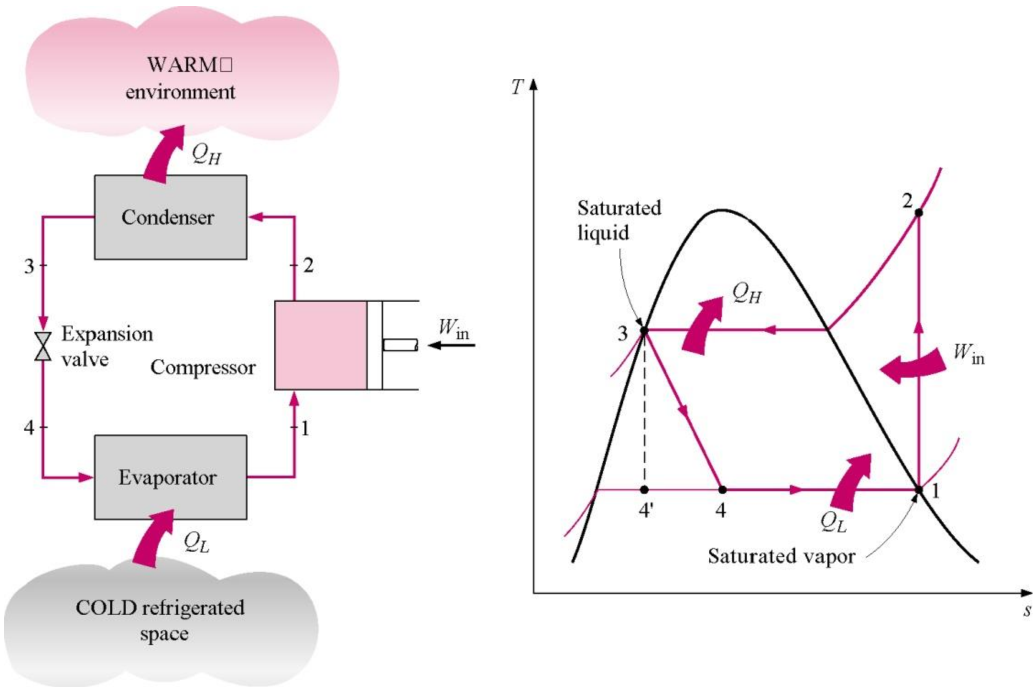


Fig 5.4

The P-h diagram is another convenient diagram often used to illustrate the refrigeration cycle.

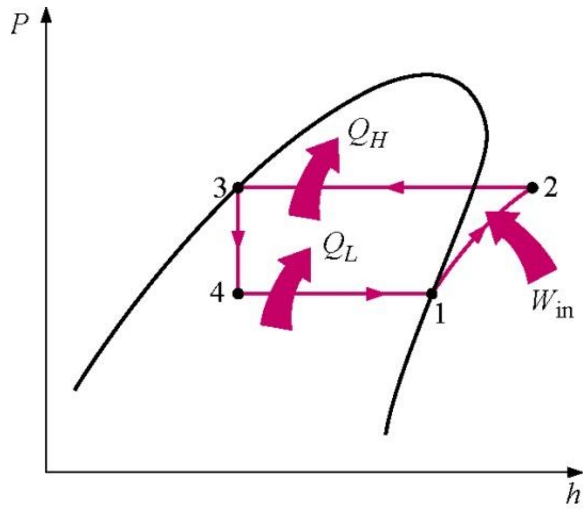


Fig 5.5

The ordinary household refrigerator is a good example of the application of this cycle.

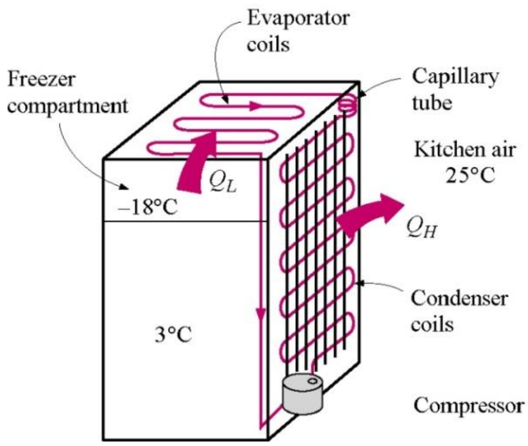


Fig 5.6

Results of First and Second Law Analysis for Steady-Flow

Component	Process	First Law Result
Compressor	s = Const.	$\dot{W}_{in} = \dot{m}(h_2 - h_1)$
Condenser	P = Const.	$\dot{Q}_H = \dot{m}(h_2 - h_3)$
Throttle Valve	$\Delta s > 0$ $\dot{W}_{net} = 0$ $\dot{Q}_{net} = 0$	$h_4 = h_3$
Evaporator	P = Const.	$\dot{Q}_L = \dot{m}(h_1 - h_4)$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

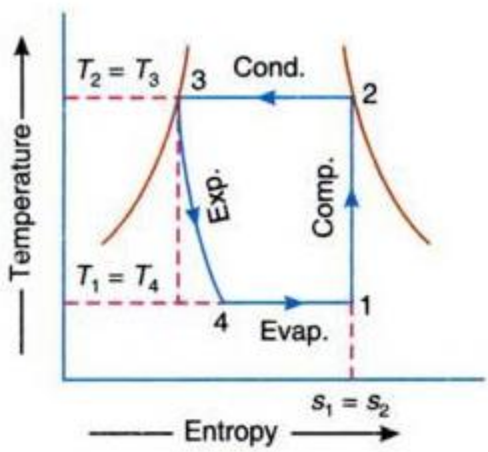
$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1}$$

Methods to enhance the COP of simple vapour compression refrigeration system

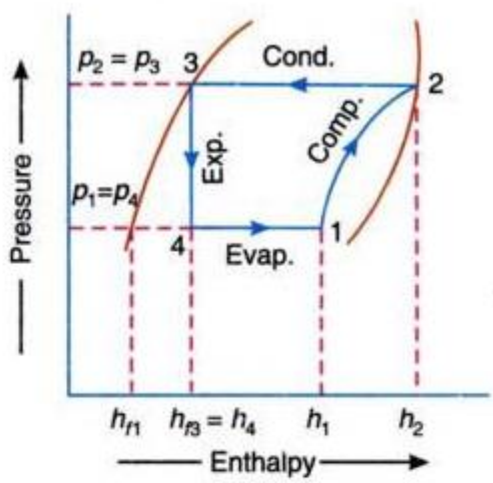
1. Cycle with dry saturated vapour after compression,
2. Cycle with wet vapour after compression,
3. Cycle with superheated vapour after compression,
4. Cycle with superheated vapour before compression, and
5. Cycle with undercooling or subcooling of refrigerant.

5.6 Theoretical Vapour Compression Cycle with Dry SaturatedVapour after Compression

A vapour compression cycle with dry saturated vapour after compression is shown in the following Figures (a) and (b) respectively. At point 1, let T_1, p_1, S_1 be the temperature, pressure and entropy of the vapour refrigerant respectively. The four processes of the cycle are as follows:



(a) T-s diagram.



(b) p-h diagram.

Fig 5.7

Compression Process

The vapour refrigerant at low pressure $p1$ and temperature$T1$ is compressed isentropic ally to dry saturated vapour as shown by the vertical line 1-2 on the T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rise from $p1$ to $p2$ and $T1$ to $T2$ respectively.

The work done during isentropic compression per kg of refrigerant is given by

$w = h2 - h1$

where $h1$ = Enthalpy of vapour refrigerant at temperature $T1$, i.e. at suction of the compressor, and

$h2$ = Enthalpy of the vapour refrigerant at temperature $T2$. i.e. at discharge of the compressor.

Condensing Process

The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure p2 and temperature T2 as shown by the horizontal line 2-3 on T-s and p-hdiagrams. The vapour refrigerant is changed into liquid refrigerant. The refrigerant,while passing through the condenser, gives its latent heat to the surrounding condensing medium.

Expansion Process

The liquid refrigerant at pressure p3 = p2 and temperature T3 = T2, is expanded by throttling process through the expansion valve to a low pressure p4 = p1 and Temperature T4 = T1 as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on p-h diagram. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

Vaporizing Process

The liquid-vapour mixture of the refrigerant at pressure p4 = p1 and temperatureT4 = T1 is evaporated and changed into vapour refrigerant at constant pressure andtemperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which, is to be cooled, This heat which is absorbed by the refrigerant is called refrigerating effect and it is briefly written asR_E. The process of vaporization continues up to point 1 which is the starting point and thus the cycle is completed.

We know that the refrigerating effect or the heat absorbed or extracted by the liquid-vapour refrigerant during evaporation per kg of refrigerant is given by

$$R_E = h_1 - h_4 = h_1 - hf_3$$

where hf3 = Sensible heat at temperature T3, i.e. enthalpy of liquid refrigerant leaving the condenser.

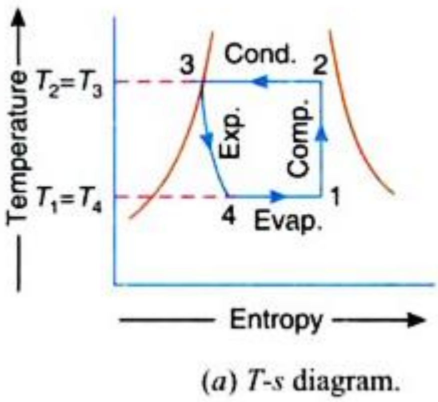
It may be noticed from the cycle that the liquid-vapour refrigerant has extracted heat during evaporation and the work will be done by the compressor for isentropic compression of the high pressure and temperature vapour refrigerant.

Coefficient of performance, C.O.P. = (Refrigerating effect)/(Work done)

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{h_1 - h_{f3}}{h_2 - h_1}$$

5.7 Theoretical Vapour Compression Cycle with Wet Vapour after Compression



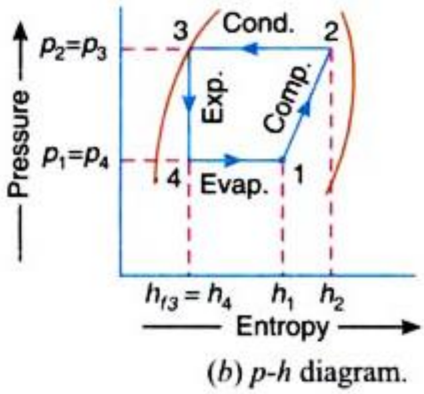


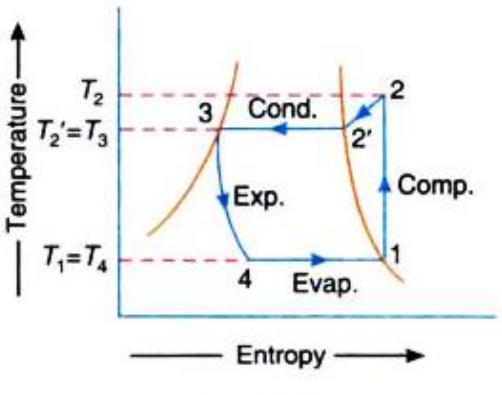
Fig 5.8

From the above figure, the end point of compression is lies in the region of wet (liquid and vapour). The enthalpy and entropy at this point is calculated by following formula

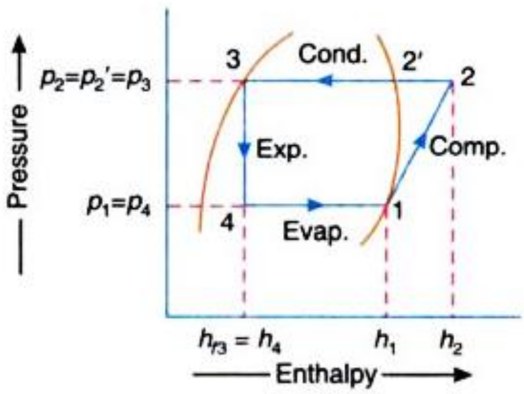
$$h_2 = h_{f2} + x_2 h_{fg2}$$

$$s_2 = s_{f2} + \frac{x_2 h_{fg2}}{T_2}$$

5.8 Theoretical Vapour Compression Cycle with Superheated Vapour after Compression



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

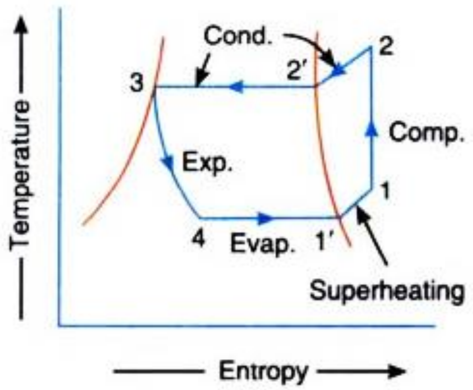
Fig 5.9

From the above figure, the end point of compression is lies in the region of superheated vapour. The enthalpy and entropy at this point is calculated by following formula

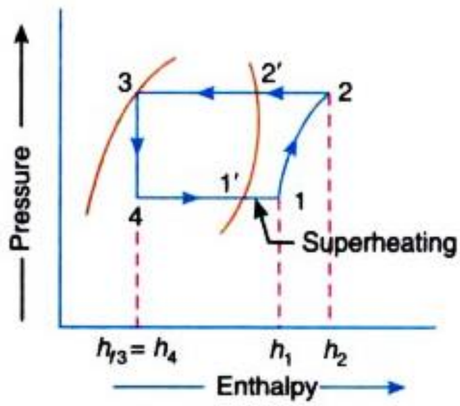
$$h_2 = h_{2'} + c_p \times \text{Degree of superheat} = h_{2'} + c_p\,(T_2 - T_{2'})$$

$$s_2 = s_{2'} + 2.3\, c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

5.9 Theoretical Vapour Compression Cycle with Superheated Vapour before Compression



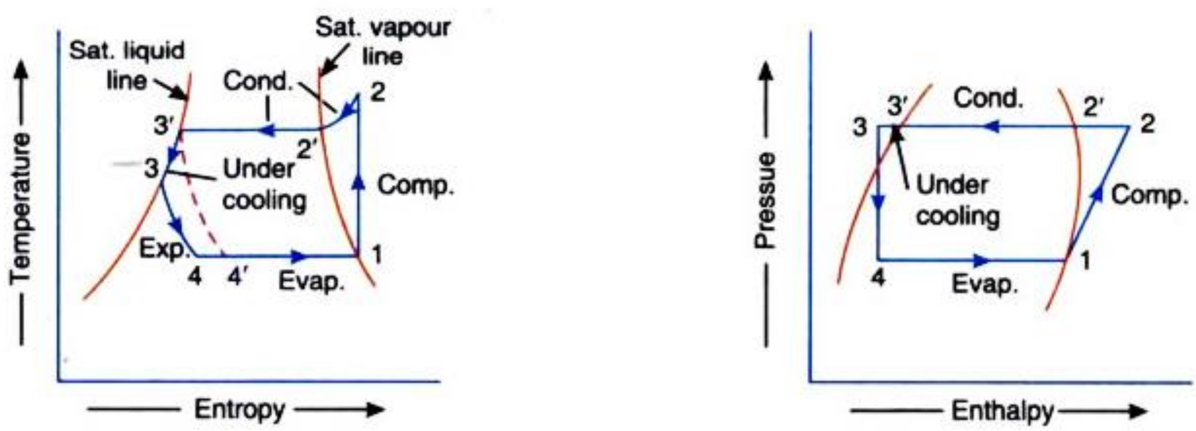
(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

Fig 5.10

5.10 Theoretical Vapour Compression Cycle with Sub-Cooling or Under cooling of Refrigerant



(*a*) *T-s* diagram.

(*b*) *p-h* diagram.

Fig 5.11

$$h_{f3} = h_{f3'} - c_p \times \text{Degree of undercooling}$$

The process 3-3', cooling of the refrigerant temperature below its saturation temperature value is called **sub cooling or under cooling**at the end of condensation process.

5.11 REFRIGERANTS

The working agent in a refrigerating system that absorbs carries or releases heat from the place to be cooled or refrigerated can be termed as a refrigerant. This heat transfer generally takes place through a phase change of the refrigerant. Amore complete definition of a refrigerant could be given as follows:

"Refrigerant is the fluid used for heat transfer in a refrigerating system that absorbs heat during evaporation from the region of low temperature and pressure, and releases heat during condensation at a region of higher temperature and pressure."

Primary and secondary refrigerants

Primary refrigerants are those which can be directly used for the purpose of refrigeration. If the refrigerant is allowed to flow freely into the space to be refrigerated and there is no danger of possible harm to human beings, then primary refrigerants are used. The refrigerants used in home refrigerators likeFreon-12 are primary refrigerants.

On the other hand, there may be certain situations in which we cannot allow the refrigerant to come in direct contact with the items being refrigerated, and thenthe refrigerant used is termed as a secondary refrigerant. As for example, we cannot allow a toxic refrigerant to be used for air conditioning in residential buildings. There are some refrigerants which are highly inflammable and so their direct use is forbidden for safety reasons. Again, it may so happen that if direct refrigeration, such as in cooling a big cold storage, is allowed, then the amount of refrigerant required may be so large that its cost becomes prohibitively high. These are some typical situations for which we favor the use of secondary refrigerants. Water and brine solutions are common examples of secondary refrigerants.

5.12Classification of refrigerants

Refrigerants can be broadly classified based on the following:

Working Principle

Under this heading, we have the primary or common refrigerants and the secondary refrigerants.

The primary refrigerants are those that pass through the processes of compression, cooling or condensation, expansion and evaporation or warming up during cyclic processes. Ammonia, R12, R22, carbon dioxide come under this class of refrigerants. On the other hand, the medium which does not go through the cyclic processes in a refrigeration system and is only used as a medium for heat transfer are referred to as secondary refrigerants. Water, brine solutions of sodium chloride and calcium chloride come under this category.

Safety Considerations

Under this heading, we have the following three sub-divisions.

Safe refrigerants

These are the non-toxic, non-flammable refrigerants such as R11, R12, R13, R14, R21, R22, R113, R114, methyl chloride, carbon dioxide, water etc.

Toxic and moderately flammable

Dichloroethylene methyl format, ethylchloride, sulphur dioxide, ammonia etc. come under this category.

Highly flammable refrigerants

The refrigerants under this category are butane, isobutene, propane, ethane, methane, ethylene etc.

Chemical Compositions

Halocarbon compounds

These are obtained by replacing one or more hydrogen atoms in ethane or methane with alogens.

Azeotropes

These are the mixtures of two or more refrigerants and behave as a compound.

Oxygen and Nitrogen Compounds

Refrigerants having either oxygen or nitrogen molecules in their structure, such as ammonia, are grouped separately and have a separate nomenclature from the halogenated refrigerants.

Cyclic organic Compounds

The compounds coming under this class are R316, R317 and R318.

Inorganic Compounds

These are further divided into two categories: Cryogenic and Non-cryogenic. Cryogenic fluids are those which are applied for achieving temperatures as low as – 160 $^{\circ}$C to – 273 $^{\circ}$C. Above this temperature range, we can use a multi-stage refrigeration system to realise the desired temperature. But below – 160°C, this is not possible since the COP of the cycle becomes very low. To attain temperatures below –160 $^{\circ}$C, we use refrigerants such as nitrogen, oxygen, helium, hydrogen etc. and for temperatures close to – 273°C, magnetic cooling is employed. The inorganic compounds which are employed above the cryogenic temperature ranges come under the remaining sub-division of inorganic refrigerants.

Unsaturated Compounds

Compounds such as ethylene, propylene etc., are grouped under this head and grouped under the 1000 series for convenience.

Miscellaneous

This group contains those compounds which cannot be grouped under the other components. They are indicated by the 700 series with the last numbers being their molecular weight. Examples include air, carbon dioxide, sulphur dioxide etc. As we can see from the above sub-divisions, they are not mutually exclusive. A compound may come under more than one sub-division. Hence, the importance of adopting the various naming conventions to designate the different refrigerants cannot be underestimated.

5.13 Designation of refrigerants

The American Society of Refrigerating Engineers (ASRE) has developed certain conventions for use in naming different types of refrigerants. These naming conventions differ according to the type of refrigerant. Each refrigerant type is denoted by a different series. Thus, we have separate series for halogenated refrigerants and other types. The naming conventions are simple and easy to follow. These conventions are now accepted worldwide and help to name the large variety of refrigerants available commercially nowadays.

Halocarbon Compounds

These are represented by a three digit nomenclature. Here, the first digit represents the number of carbon atoms in the compound minus one, the second digit stands for the number of hydrogen atoms plus one while thethird digit stands for the number of fluorine atoms. The remaining atoms arechlorine. As an example, let us consider the refrigerant having R22 as its three digit nomenclature.

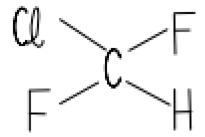
According to the above mentioned convention,

No. of C atoms in R22: C – 1 = 0 => C = 1

No. of H atoms in R22: H + 1 = 2 => H = 1

No. of F atoms in R22: F = 2

Since there is only one carbon atom in the compound, this compound hasoriginated from the methane series (CH). From the calculation, we can see there is one hydrogen atom and two fluorine atoms. The remaining valence bond of carbon will be balanced by chlorine. Thus, the substance is



Graphical Representation of Monochloro-Difluoro-Methane

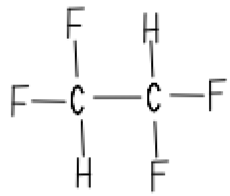
Therefore, chemical formula of R22 is $CHClF_2$and has the name Monochloro-difluoro-methane. Taking again the example of R134, we can calculate its chemical formula asabove which gives us

No. of C atoms: C – 1 = 1 => C = 2

No. of H atoms: H + 1 = 3 => H = 2

No. of F atoms: F = 4

Therefore, no. of Cl atoms: Cl = 0



Graphical Representation of Tetrafluoroethane

The compound is $C_2H_2F_2$and its name is Tetrafluoroethane.The non-halogenated refrigerants follow a different naming convention which is dependent upon the series of the refrigerant.

5.14 DESIRABLE PROPERTIES OF REFRIGERANTS

The vast number of refrigerants available in the market today allows us to choose a refrigerant depending upon the operating conditions of the refrigeration system. As such, there is no refrigerant that can be advantageously used under all operating conditions and in all types of refrigeration systems. In spite of that, we can state certain desirable properties that a refrigerant should posses. These properties can be divided into favorable thermodynamic, chemical and physical properties:

5.14.1 Thermodynamic Properties

Critical Temperature and Pressure

The critical temperature of the refrigerant should be as high as possible above the condensing temperature in order to have a greater heat transfer at a constant temperature. If this is not taken care of, then we will have excessive power consumption by the refrigeration system. The critical pressure should be moderate and positive. A very high pressure will make the system heavy and bulky whereas in case of very low pressures, there is a possibility of air leaking into the refrigerating system.

Specific Heat

The specific heat of the liquid should be as small as possible. This ensures that the irreversibilities associated with throttling are small and there is greater subcooling of the liquid. On the other hand, the specific heat of vapor should be high to have less superheating of the vapor.

Enthalpy of Vaporization

This should be as large as possible to minimize the area under superheat and the area reduction due to throttling. Also, the higher value of enthalpy of vaporization lowers the required flow rate per ton of refrigeration.

Conductivity

The conductivity of the refrigerant should be as high as possible so that the size of the evaporator and condenser is manageable. From this viewpoint, ammonia has a better conductivity than that of R12 or R22 and is more suitable than the latter. But, ammonia is toxic and this does not allow its use in home refrigeration systems.

Evaporator and Condenser Pressure

Both the evaporator and condenser pressures need to be above atmospheric pressure otherwise there is a possibility of air leaking into the system. Presence of air drastically reduces the capacity of the refrigeration system. Also, due to presence of moisture in air, acids or other corrosive compounds may form and this may affect the tubing of the refrigeration system.

Compression Ratio

The compression ratio needs to be as small as possible otherwise the leakage of refrigerant occurs across the piston. Also, the volumetric efficiency is affected.

Freezing Point

It should be as low as possible or else there will be a possibility of blockage of passages during flow of fluid through evaporator.

Volume of Refrigerant Handled Per Ton of Refrigeration

This should be as small as possible in order to have a small size of the compressor. The type of compressor is decided by this value. Forrefrigerants like R12, R500, R22 etc., a reciprocating compressor is suitable. For others like R11 and water, a centrifugal compressor is required to handle the large volume.

Coefficient of Performance

The Coefficient of performance or COP has a direct bearing on the running cost of the refrigeration system. Higher the magnitude of COP, lower will be the running cost. Since, the COP of any refrigeration system is limited by the Carnot COP, for large operating pressures a multi-stage refrigeration system should be employed. CO_2has a very low COP. Hence, it is not suitable for use as a refrigerant.

Density

The density of the refrigerant should be as large as possible. In reciprocating compressors, the pressure rise is accomplished by squeezing the entrapped fluid inside the piston-cylinder assembly. Hence, densitydecides the size of the cylinder. Again in centrifugal compressors pressurerise is related to the density of the vapor. A high value of density results in high pressure rise.

Compression Temperature

Whenever a refrigerant gets compressed, there is a rise in the temperature of the refrigerant resulting in the heating of the cylinder walls of the compressor. This necessitates external cooling of the cylinder walls to prevent volumetric and material losses. Refrigerants having lowest compression temperatures are thus better than others.

5.14.2 Chemical Properties

Chemical Stability and Inertness

It should be chemically stable for the operating ranges of temperature. Also, it should not react with the materials of the refrigeration system or with which it comes into contact. Further, it should be chemically inert and must not undergo polymerization reactions at either the lower or higher ranges of temperatures.

Action on Rubber or Plastics

Rubber and plastics are used extensively in the refrigeration system. These materials are mostly used in the seals and gaskets of the refrigeration system. They help to prevent the leakage of the refrigerant and ensure the smooth functioning of the compressor. The refrigerant should not react with them or else there might be leakage of refrigerant from the system or loss of functioning of the compressor.

Flammability

The refrigerant should be inert and not catch fire when subjected to high temperatures. From this viewpoint $CO_2$is the most suitable as it is not onlynon-flammable, but also acts as a fire-extinguisher. Ethane, butane,isobutene are highly undesirable as they catch fire quickly.

Effect on Oil

The refrigerant should not react with the lubricating oil else, there is a possibility of loss of lubricating action due to either thickening or thinning of the oil. It should not be soluble in the oil else there will be reduction in the viscosity of the lubricating oil.

Effect on Commodity

If the refrigerant is directly used for chilling, then it should not affect the commodity kept in the conditioned space. Also, in case where direct cooling is not employed, the refrigerant should still not affect the commodity if there is any leakage.

Toxicity

The refrigerant used in air conditioning, food preservation etc. should not be toxic as they will come into contact with human beings.

5.14.3 Physical Properties

Leakage and Detection

Since pressures higher than atmospheric are usually employed in refrigeration systems, there is a possibility of leakage of refrigerants after long period of operation. It is desirable to detect this leak early else the system would operate under reduced capacity or stop functioning altogether. Hence, it is desirable that the refrigerant has a pungent smell so that its leakage can be detected immediately.

Miscibility with Oil

The refrigerant should not be miscible with the oil else the lubricating strength will be reduced.

Viscosity

It should be as small as possible to ensure that the pressure drop in the system is as small as possible. A low viscosity refrigerant will require less energy for its circulation through the refrigeration system.

5.14.4 Safety Criteria

Under safety criteria, we consider the toxicity, flammability, action on perishablefood and formation of explosive compound on exposure to air. An ideal refrigerant should be non-toxic, non-flammable, have no effect on food product sand should not react with atmospheric air. No refrigerant satisfy these criteria fully. We can therefore, group refrigerants into different sub-groups based on their flammability and toxicity levels.

5.14.5 Economic Criteria

Apart from the thermodynamic, chemical, physical and safety criteria, there is another criterion by which we judge an ideal refrigerant. The economic criterion takes into account the cost of the refrigerant, the availability and supply levels of the refrigerant, cost of storage and handling. We discuss each of these in detail below.

Cost of Refrigerant

The cost of the refrigerant has a big impact on the overall cost of the refrigeration system. Hence, its cost should be as low as possible. From this view point, ammonia and water are ideally suited, but their low thermodynamic and chemical properties restrict their use in all types ofrefrigeration systems. Particularly, for flooded type evaporator or condenser, the refrigerant amount required is high and their cost needs to be factored in while making the initial investments.

Availability and Supply

The refrigerant should be easily available in the market and in abundant quantity. This ensures that the cost of the refrigerant is not prohibitive. Anabundant and free supply of the refrigerant ensures that refrigeration systems will be designed specifically for use with them.

Storage and Handling

The refrigerant should be such that it can be conveniently stored and handled during transportation and charging. It should be stored in as small a pressure vessel as possible. Also, if we have to handle a toxic or flammable refrigerant, then the cost involved will be higher compared to handling and storage cost of non-toxic and non-flammable refrigerant.

5.15 COMMON REFRIGERANTS

The refrigerants which are available commercially in the market are numerous. Some of them which are in common use are mentioned below:

Air

Air (molecular weight 28.97, specific heats Cp = 1.04 kJ/kgK and Cv = 0.712kJ/kg-K) is one of the earliest refrigerant to be used in the refrigeration systems. Its advantages are that it is available free of cost, is non-toxic and non-flammable and does not affect the commodity if pure. However, airsuffers from a number of drawbacks. Air contains moisture and this reacts with the material of the evaporator and condenser severely affecting their working capacity. Further, there is a possibility that the passages may be blocked by the formation of ice from this moisture. The COP of air is of the order of 0.6 and thus, not suitable for use in refrigeration systems on a commercial scale. It is mainly used for air conditioning in aircrafts where efficiency of operation is of secondary importance.

Ammonia

Ammonia (molecular weight 17) is one of the oldest refrigerants and it was commonly employed in places where toxicity effects were of secondary importance. Its advantages are its low cost, low specific volume, high COP(of the order of 4.0) and high refrigeration effect per unit mass of the refrigerant. Its primary drawback is its toxicity which prevents its use in airconditioning and food preservationsystems. Ammoniahasa boilingpointof-33^{0}C at atmospheric pressure.

Carbon Dioxide

Carbon dioxide (molecular weight 44) is a non-toxic and non-poisonous refrigerant. Also, it is not only non-flammable but and is an excellent extinguishing agent as well. Its other advantages are that it is chemically stable, immiscible with the lubricating oil and does not affect the metal used in the system. It has a low specific volume and this requires volume displacement per ton of refrigeration. However, its critical pressure is too high. Also, its critical temperature is only 31^{0}C which makes it unsuitable for use in countries with a hot climate like India. It is an excellent refrigerant for low temperature refrigeration.

Sulphur Dioxide

Sulphur dioxide (molecular weight 64) is a colourless, suffocating and irritating gas and is twice as heavy as air at atmospheric conditions. It was mostly used as a household refrigerant in the older days, but has since been discarded for better refrigerants. It suffers from a lot of disadvantages. Sulphur dioxide reacts with water forming sulphurous acid, which inpresence of oxygen becomes sulphuric acid, a corrosive compound for metals. It is non-flammable but attacks foodstuff on coming in contact with it. It is also partially miscible with the lubricating oil.

Hydrocarbons

This group consists of colourless fluids normally in gaseous state and made up of various combinations of carbon and hydrogen. Most of the refrigerants from this category are suitable for low temperature refrigeration. Isobutane falls in this category and has been suitable for domestic refrigeration. They are non-poisonous, but are flammable and highly explosive when exposed to air. The molecular weight and boiling point of each gas varies according to the number of hydrogen and carbon atoms. The larger the number of hydrogen and carbon atoms, the heavier isthe gas and higher is its boiling point.

Halocarbon Refrigerants

The halocarbon refrigerants are formed by replacing one or more of hydrogen atoms of methane or ethane by one or more atoms of the three halogens: fluorine, chlorine or bromine. Some of the refrigerants coming under this category are mentioned below:

Refrigerant R12

The refrigerant R12 is the most widely used refrigerant in the domestic and large commercial establishments. Its chemical formula is CCl_2F_2 and its boiling point is -30^{0}C at 1 bar. It is a non-flammable,non-explosive, non-irritating, non-toxic and odourless refrigerant. It remains chemically stable up to 550^{0}C. Also, it does not affect the material of the refrigeration system. It is available in abundance and is quite cheap. However, its use is being discontinued nowadays for its contribution to ozone depletion which will be discussed later.

Refrigerant R13

Its chemical formula is $CClF_3$. It is a non-flammable, non-toxic and stable refrigerant. It is very suitable for achieving low temperatures in a cascade refrigeration system. Its specific volume is high and therefore, it is suitable for centrifugal compressors. However, it also has a negative effect on ozone depletion.

Refrigerant R22

Its chemical formula is $CHClF_2$. It is also a non-toxic, non-flammable, non-corrosive and non-irritating refrigerant. It is the most common refrigerant for use in large refrigeration systems and is preferred toR12.

Refrigerant R114

Its chemical formula is $C_2Cl_2F_4$. Its boiling point corresponding to 1bar is about 3^{0}C. It has properties very similar to those of R12 with respect to water and oil combination. It is not suitable for low temperature refrigeration since it has negative evaporator pressure even at around 9^{0}C. It is non-toxic, non-explosive and non-corrosive even in the presence of water.

All refrigerants properties can be referred from standard refrigeration tables. Some refrigerant properties are listed below

5.16 VAPOUR ABSORPTION SYSTEM

Introduction

In a vapour absorption system the refrigerant is absorbed on leaving the evaporator, the absorbing medium being a solid or liquid. In order that the sequence of events should be continuous it is necessary for the refrigerant to be separated from the absorbent and subsequently condensed before being returned to the evaporator. The separation is accomplished by the application of direct heat in a ‘generator’. The solubility of the refrigerant and absorbent must be suitable and the plant which uses ammonia as the refrigerant and water as absorbent will be described.

Simple Vapour Absorption System

Refer Fig. 5.12 for a simple absorption system. The solubility of ammonia in water at low temperatures and pressures is higher than it is at higher temperatures and pressures. The ammonia vapour leaving the evaporator at point 2 is readily absorbed in the low temperature hot solution in the absorber. This process is accompanied by the rejection of heat. The ammonia in water solution is pumped to the higher pressure and is heated in the generator. Due to reduced solubility of ammonia in water at the higher pressure and temperature, the vapour is removed from the solution. The vapour then passes to the condenser and the weakened ammonia in water solution is returned to the absorber.

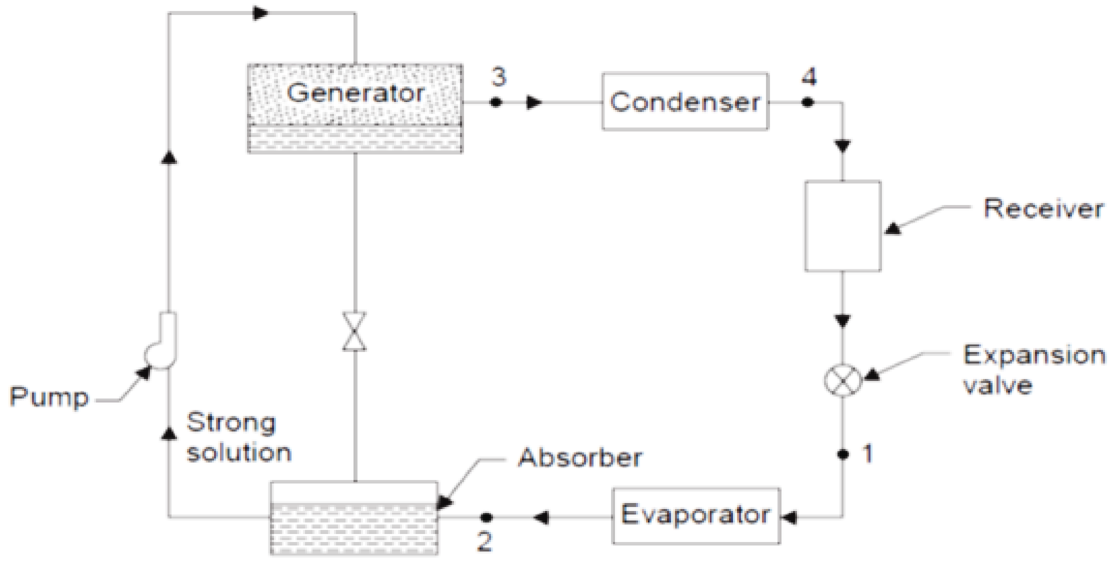
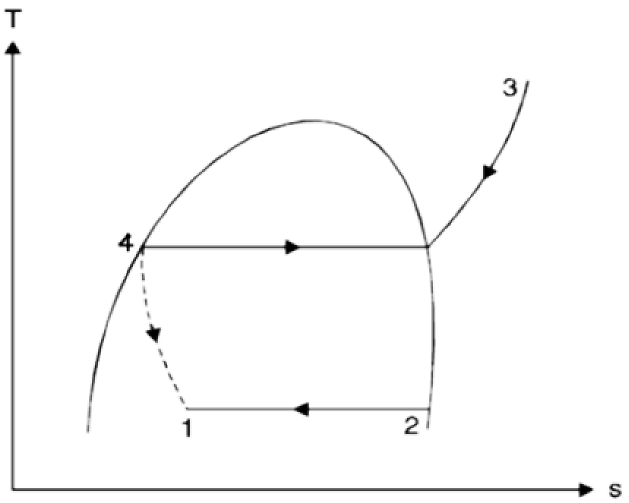


Fig 5.12



Simple vapour absorption system—T-s diagram.

Fig 5.13

In this system the work done on compression is less than in vapour compression cycle (since pumping a liquid requires much less work than compressing a vapour between the same pressures) but a heat input to the generator is required. The heat may be supplied by any convenient form e.g. steam or gas heating.

Practical Vapour Absorption System

Refer Fig. 5.14. Although a simple vapour absorption system can provide refrigeration yetits operating efficiency is low. The following accessories are fitted to make the system more practical and improve the performance and working of the plant.

1. Heat exchanger. 2. Analyser. 3. Rectifier.

1. **Heat exchanger.** A heat exchanger is located between the generator and the absorber.

The strong solution which is pumped from the absorber to the generator must be heated ; and the weak solution from the generator to the absorber must be cooled. This is accomplished by a heat exchanger and consequently cost of heating the generator and cost of cooling the absorber are reduced.

2. **Analyser.** An analyser consists of a series of trays mounted above the generator. Its main function is to remove partly some of the unwanted water particles associated with ammonia vapour going to condenser. If these water vapours are permitted to enter condenser they may enter the expansion valve and freeze ; as a result the pipe line may get choked.

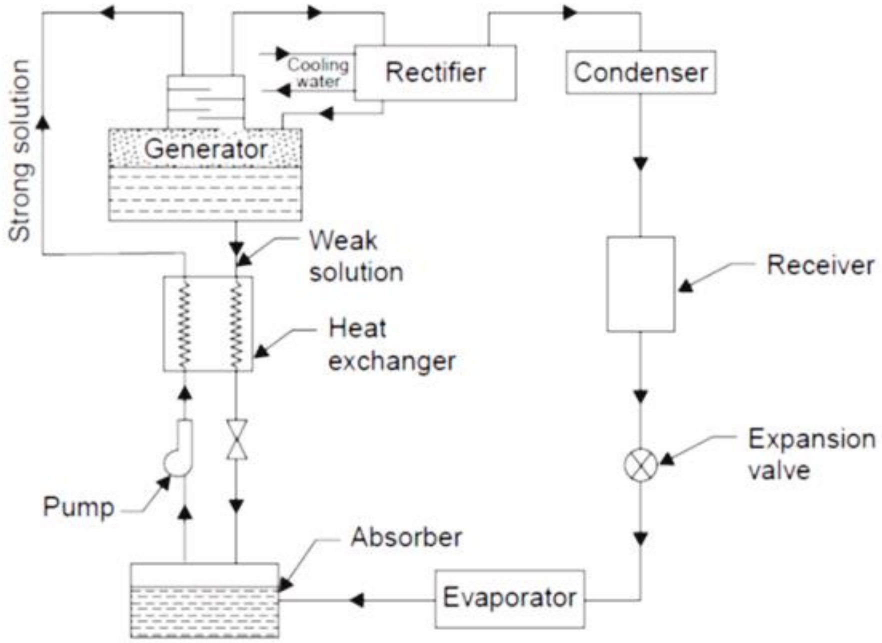


Fig 5.14

3. **Rectifier.** A rectifier is a water-cooled heat exchanger which condenses water vapour and some ammonia and sends back to the generator. Thus final reduction or elimination of the percentage of water vapour takes place in a rectifier. The co-efficient of performance (C.O.P.) of this system is given by :

$$\text{C.O.P.} = \frac{\text{Heat extracted from the evaporator}}{\text{Heat supplied in the generator + Work done by the liquid pump}} .$$

5.17 PSYCHROMETRY AND AIR – CONDITIONING

Psychrometric properties, Use of psychrometric chart, Psychrometric process – Sensible heat exchange process, Latent heat exchange process, Adiabatic mixing, Evaporative cooling, Property calculations of air-vapour mixtures.

Principles of air-conditioning, Types of air conditioning systems – summer, winter, year round air conditioners, Concept of RSHF, GSHF, ESHF, Simple problems.

5.17.1 CONCEPT OF PSYCHROMETRY AND PSYCHROMETRICS

Air comprises of fixed gases principally, nitrogen and oxygen with an admixture of water vapour in varying amounts. In atmospheric air water is always present and its relative weight averages less than 1% of the weight of atmospheric air in temperate climates and less than 3% by weight under the most extreme natural climatic conditions, it is nevertheless one of most important factors in human comfort and has significant effects on many materials. Its effect on human activities is in fact altogether disproportionate to its relative weights. *The art of measuring* the *moisture content of air is termed* "**psychrometry**". *The science which investigates the thermal properties of moist air, considers the measurement and control of the moisture content of air, and studies the effect of atmospheric moisture on material and human comfort may properly be termed* "**psychrometrics'**'.

DEFINITIONS

Some of the more important definitions are given below :

1. **Dry air.** The international joint committee on Psychrometric Data has adopted the following exact composition of air expressed in mole fractions (Volumetric) Oxygen 0.2095, Nitrogen 0.7809, Argon 0.0093, Carbon dioxide 0.0003. Traces of rare gases are neglected. Molecular weight of air for all air conditioning calculations will be taken as 28.97. Hence the gas constant, R_{air} = 0.287 kJ/kg K Dry air is never found in practice. Air always contains some moisture. Hence the common designation "air" *usually means moist* air. The term '*dry air*' is used to indicate the *water free contents* of air having any degree of moisture.

2. **Saturated air.** Moist air is said to be saturated when its condition is such that it can co-exist in natural equilibrium with an associated condensed moisture phase presenting a flat surface to it. For *a given temperature, a given quantity of air can be saturated with a fixed*

quantity of moisture. At higher temperatures, it requires a larger quantity of moisture to saturate it. At saturation, vapour pressure of moisture in air corresponds to the saturation pressure given in steam tables corresponding to the given temperature of air.

3. **Dry-bulb temperature (DBT).** It *is the temperature of air as registered by an ordinary thermometer* (*tdb*).

4. **Wet-bulb temperature (WBT).** *It is the temperature registered by a thermometer when the bulb is covered by a wetted wick and is exposed to a current of rapidly moving air* (*twb*).

5. **Adiabatic saturation temperature.** *It* is *the temperature at which the water or ice can saturate air by evaporating adiabatically into it*. It is numerically equivalent to the measured wet bulb temperature (as corrected, if necessary for radiation and conduction) (*twb*).

6. **Wet bulb depression.** It *is the difference between dry-bulb and wet bulb temperatures* (*tdb* – *twb*).

7. **Dew point temperature (DPT).** *It is the temperature to which air must be cooled at constant pressure in order to cause condensation of any of its water vapour*. It is equal to steam table saturation temperature corresponding to the actual partial pressure of water vapour in the air (*tdp*).

8. **Dew point depression.** *It* is *the difference between the dry bulb and dew point temperatures* (*tdb* – *tdp*).

9. **Specific humidity (Humidity ratio).** *It is the ratio of the mass of water vapour per unit mass of dry air in the mixture of vapour and air*, it is generally expressed as grams of water per kg of dry air. For a given barometric pressure it is a function of dew point temperature alone.

10. **Relative humidity (RH), (φ).** *It is the ratio of the partial pressure of water vapour in the mixture to the saturated partial pressure at the dry bulb temperature, expressed as percentage.*

11. **Sensible heat.** *It is the heat that changes the temperature of a substance when added*

to or abstracted from it.

12. **Latent heat.** *It is the heat that does not affect the temperature but changes the state*

of substance when added to or abstracted from it.

13. **Enthalpy.** *It is the combination energy which represents the sum of internal and flow energy in a steady flow process*. It is determined from an arbitrary datum point for the air mixture and is expressed as kJ per kg of dry air (*h*).

Note. When air is saturated DBT, WBT, DPT are *equal*.

5.17.2 PSYCHROMETRIC RELATIONS

Pressure

Dalton's law of partial pressure is employed to determine the pressure of a mixture of gases. This law states that the total pressure of a mixture of gases is equal to the sum of partial pressures which the component gases would exert if each existed alone in the mixture volume at the mixture temperature. Precise measurements made during the last few years indicate that this law as well as Boyle's and Charle's laws are only approximately correct. Modern tables of atmospheric air properties are based on the correct versions. For calculating partial pressure of water vapour in the air many equations have been proposed, probably Dr. Carrier's equation is most widely used.

$$p_v = (p_{vs})_{wb} - \frac{[pt - (p_{vs})_{wb}](t_{db} - t_{wb})}{1527.4 - 1.3\, t_{wb}}$$

where p_v = Partial pressure of water vapour,

p_{vs} = Partial pressure of water vapour when air is fully saturated,

p_t = Total pressure of moist air,

t_{db} = Dry bulb temperature (°C), and

t_{wb} = Wet bulb temperature (°C).

Specific humidity W :

$$\text{Specific humidity} = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}}$$

or

$$W = \frac{m_v}{m_a}$$

Also,

$$m_a = \frac{p_a V}{R_a T}$$

and

$$m_v = \frac{p_v \times V}{R_v \times T}$$

where p_a = Partial pressure of dry air,

p_v = Partial pressure of water vapour,

V = Volume of mixture,

R_a = Characteristic gas constant for dry air, and

R_v = Characteristic gas constant for water vapour.

From equations (10.2) and (10.3)

$$W = \frac{p_v \times V}{R_v \times T} \times \frac{R_a T}{p_a V} = \frac{R_a}{R_v} \times \frac{p_v}{p_a}$$

But $$R_a = \frac{R_0}{M_a} \quad \left(= \frac{8.3143}{28.97} = 0.287 \text{ kJ/kg K in SI units} \right)$$

$$R_v = \frac{R_0}{M_v} \quad \left(= \frac{8.3143}{18} = 0.462 \text{ kJ/kg K in SI units} \right)$$

where R_0 = Universal gas constant,

M_a = Molecular weight of air, and

M_v = Molecular weight of water vapour.

$$\therefore \quad W = \frac{0.287}{0.462} \cdot \frac{p_v}{p_a} = 0.622 \frac{p_v}{p_t - p_v}$$

$$W = 0.622 \frac{p_v}{p_t - p_v}$$

Degree of saturation (μ) :

$$\text{Degree of saturation} = \frac{\text{Mass of water vapour associated with unit mass of dry air}}{\text{Mass of water vapour associated with saturated unit mass of dry saturated air}}$$

i.e., $$\mu = \frac{W}{W_s}$$

where, W_s = Specific humidity of air when air is fully saturated

$$\therefore \quad \mu = \frac{0.622\left(\dfrac{p_v}{p_t - p_v}\right)}{0.622\left(\dfrac{p_{vs}}{p_t - p_{vs}}\right)} = \frac{p_v(p_t - p_{vs})}{p_{vs}(p_t - p_v)}$$

$$= \frac{p_v}{p_s}\left[\frac{\left(1 - \dfrac{p_{vs}}{p_t}\right)}{\left(1 - \dfrac{p_v}{p_t}\right)}\right]$$

where p_{vs} = Partial pressure of water vapour when air is fully saturated (p_{vs} can be calculated from steam tables corresponding to the dry bulb temperature of the air).

Relative humidity (RH), ϕ :

$$\text{Relative humidity, } \phi = \frac{\text{Mass of water vapour in a given volume}}{\text{Mass of water vapour in the same volume if saturated at the same temp.}}$$

$$= \frac{m}{m_{vs}} = \frac{\dfrac{p_v T}{R_v T}}{\dfrac{p_{vs} T}{R_v T}} = \frac{p_v}{p_{vs}}$$

Enthalpy of moist air

It is the sum of enthalpy of dry air and enthalpy of water vapour associated with dry air. It is expressed in kJ/kg of dry air

$$h = h_{air} + W \,.\, h_{vapour}$$
$$= c_p t_{db} + W \,.\, h_{vapour}$$

where h = Enthalpy of mixture/kg of dry air,

h_{air} = Enthalpy of 1 kg of dry air,

h_{vapour} = Enthalpy of 1 kg of vapour obtained from steam tables,

W = Specific humidity in kg/kg of dry air, and

c_p = Specific heat of dry air normally assumed as 1.005 kJ/kg K.

Also $$h_{vapour} = h_g + c_{ps} (t_{db} - t_{dp})$$

where h_g = Enthalpy of saturated steam at dew point temperature,

and c_{ps} = 1.88 kJ/kg K.

However, a better approximation is given by the following relationship :

$$h_{vapour} = 2500 + 1.88 t_{db} \text{ kJ/kg of water vapour}$$

where t_{db} is dry bulb temperature in °C, and the datum state is liquid water at 0°C.

$$\therefore \quad h = 1.005\, t_{db} + W(2500 + 1.88\, t_{db}) \text{ kJ/kg dry air.}$$

5.18 PSYCHROMETRIC CHARTS

The psychrometric charts are prepared to represent graphically all the necessary moist air properties used for air conditioning calculations. The values are based on actual measurements verified for thermodynamic consistency. For psychrometric charts the most convenient co-ordinates are dry bulb temperature of air vapour mixture as the abcissa and moisture content (kg/kg of dry air) or water vapour pressure as the ordinate. Depending upon whether the humidity contents is abcissa or ordinate with temperature co-ordinate, the charts are generally classified as Mollier chart and Carrier chart. Carrier chart having *tdb* as the abcissa and *W* as the ordinate finds a wide application.

The chart is constructed as under :

1. *The dry bulb temperature* (ºC) of unit mass of dry air for different humidity contents or humidity ratios are indicated by *vertical lines drawn parallel to the ordinate.*

2. *The mass of water vapour* in kg (or grams) per kg of dry air is drawn parallel to the abcissa for different values of dry bulb temperature. It is the *major vertical scale of the chart.*

3. *Pressure of water vapour in mm of mercury* is shown in the scale at left and is the absolute pressure of steam.

4. *Dew point temperatures* are temperatures corresponding to the boiling points of water at low pressures of water vapour and are shown in the scale on the *upper curved line.* The dew points for different low pressures are read on *diagonal co-ordinates*

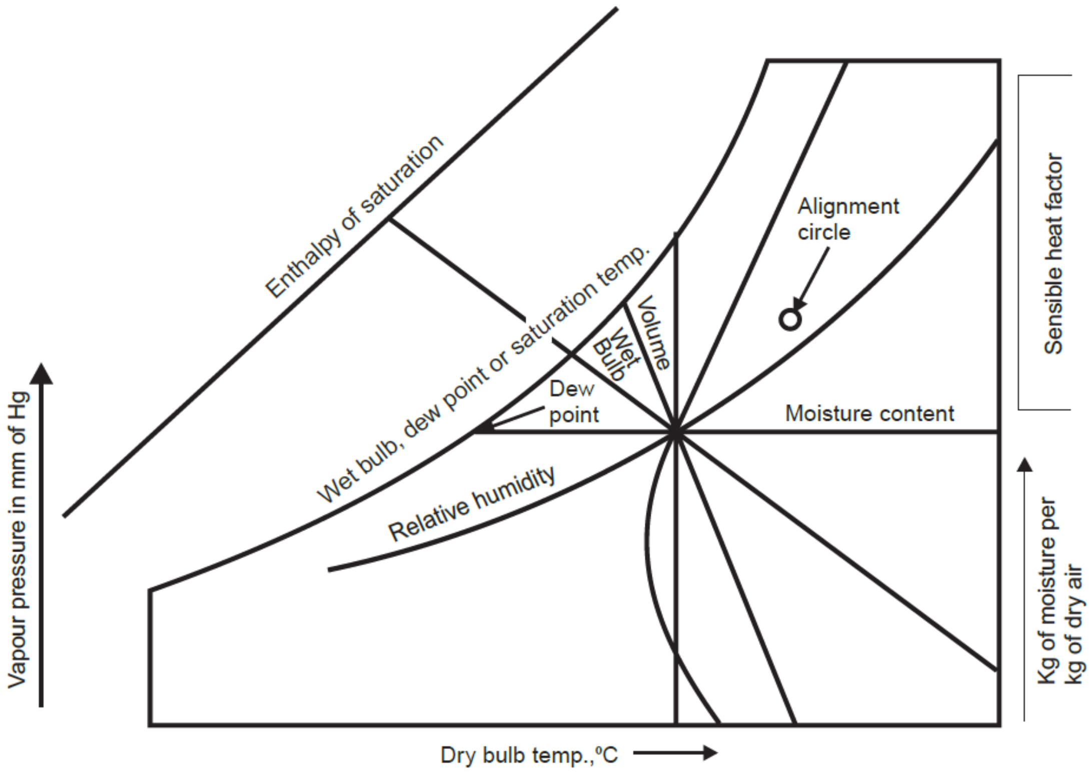


Fig 5.15

5. Constant relative humidity lines in per cent are indicated by marking off vertical distancesbetween the saturation line or the upper curved line and the base of the chart. The relative humidity curve depicts quantity (kg) of moisture actually present in the air as a percentage of the total amount possible at various dry bulb temperatures and masses of vapour.

6. *Enthalpy or total heat* at saturation temperature in kJ/kg of dry air is shown by a diagonal system of co-ordinates. The scale on the diagonal line is separate from the body of the chart and is indicated above the saturation line.

7. *Wet bulb temperatures* are shown on the diagonal co-ordinates coinciding with heat coordinates. The scale of wet bulb temperatures is shown on the saturation curve. The diagonals run downwards to the right at an angle of 30º to the horizontal.

8. The volume of air vapour mixture per kg of dry air (specific volume) is also indicated by a set of diagonal co-ordinates but at an angle of 60º with the horizontal. The other properties of air vapour mixtures can be determined by using formulae (already discussed).

In relation to the psychrometric chart, these terms can quickly indicate many things about the condition of air, for example :

1. If dry bulb and wet bulb temperatures are known, the relative humidity can be read from the chart.

2. If the dry bulb and relative humidity are known, the wet bulb temperature can be determined.

3. If wet bulb temperature and relative humidity are known, the dry bulb temperature can be found. 4. If wet bulb and dry bulb temperatures are known, the dew point can be found.

5. If wet bulb and relative humidity are known, dew point can be read from the chart.

6. If dry-bulb and relative humidity are known, dew point can be found.

7. The quantity (kg) of moisture in air can be determined from any of the following combinations :

(*i*) Dry bulb temperature and relative humidity ;

(*ii*) Dry bulb temperature and dew point ;

(*iii*) Wet bulb temperature and relative humidity ;

(*iv*) Wet bulb temperature and dew point temperature ;

(*v*) Dry bulb temperature and wet bulb temperature ; and

(*vi*) Dew point temperature alone.

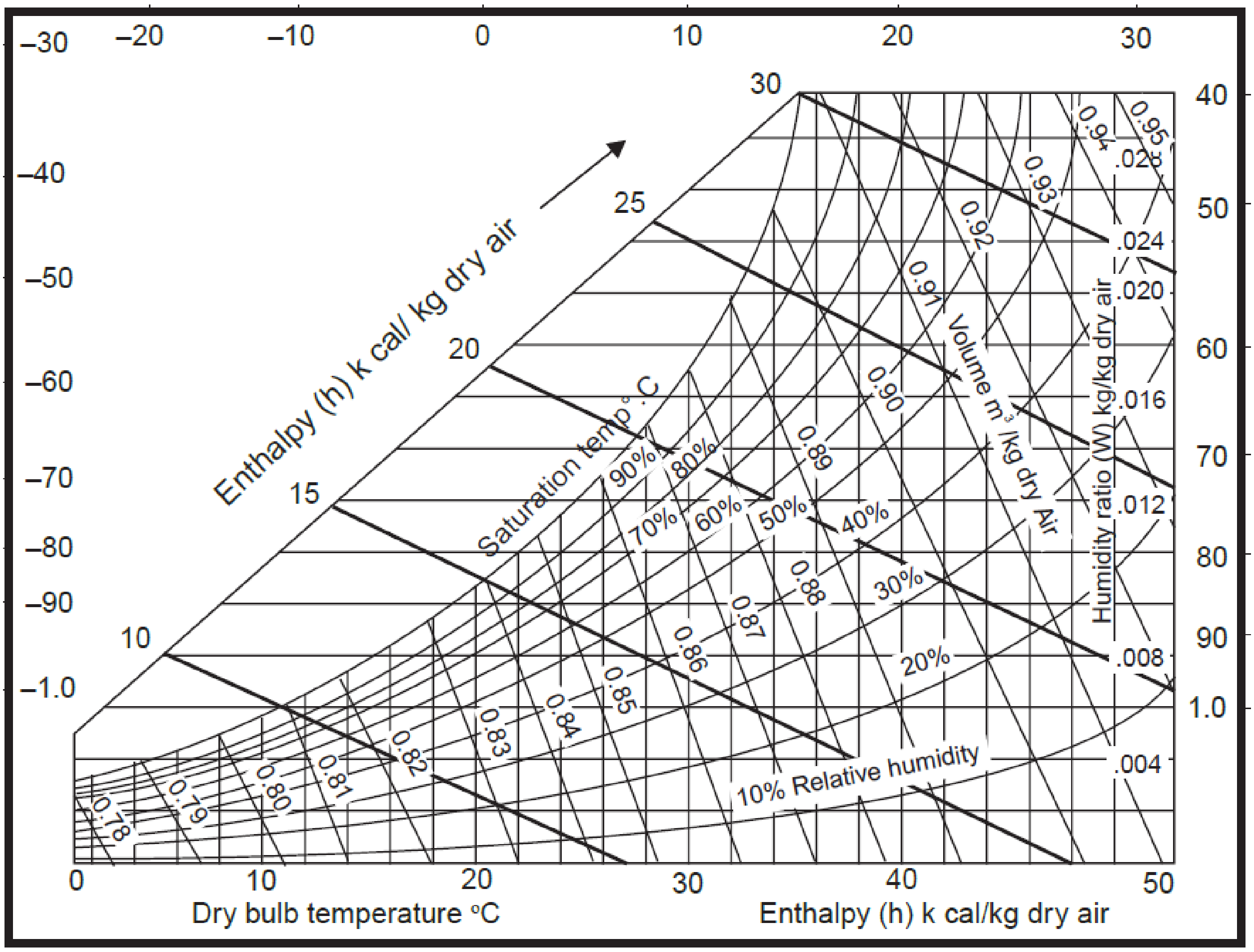


Fig 5.16 Carrier Psychrometric chart

5.19 PSYCHROMETRIC PROCESSES

In order to condition air to the conditions of human comfort or of the optimum control of an industrial process required, certain processes are to be carried out on the outside air available. The processes affecting the *psychrometric properties of air are called* ***psychrometric processes.***

These processes involve mixing of air streams, heating, cooling, humidifying, dehumidifying, adiabatic saturation and mostly the combinations of these.

The important psychrometric processes are enumerated and explained in the following text

1. Mixing of air streams

2. Sensible heating

3. Sensible cooling

4. Cooling and dehumidification

5. Cooling and humidification

6. Heating and dehumidification

7. Heating and humidification.

Mixing of Air Streams

Refer Figs. 5.17 and 5.18 Mixing of several air streams is the process which is very frequently used in air conditioning. This mixing normally takes place without the addition or rejection of

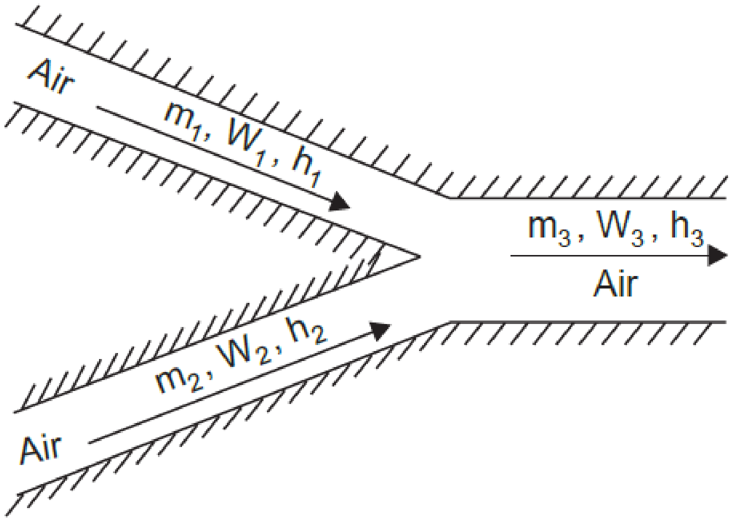


Fig 5.17 Mixing of Air streams

either heat or moisture, i.e., adiabatically and at constant total moisture content. Thus we can write the following equations:

$$m_1 + m_2 = m_3$$
$$m_1W_1 + m_2W_2 = m_3W_3$$
$$m_1h_1 + m_2h_2 = m_3h_3$$

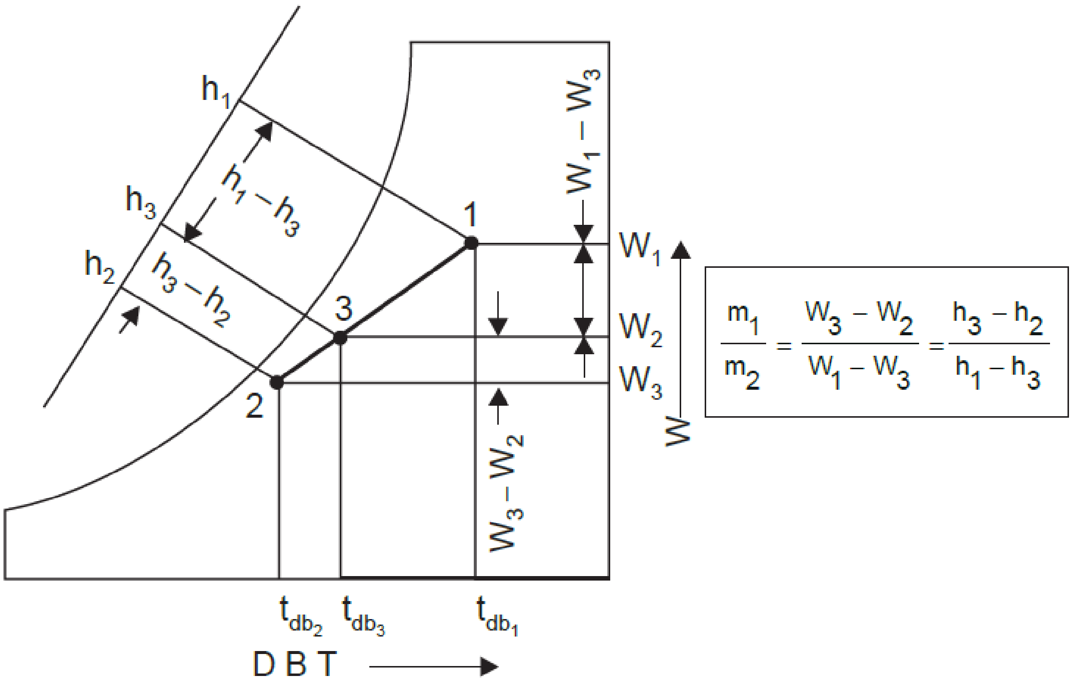


Fig 5.18

Rearranging of last two equations gives the following :

$$m_1(W_1 - W_3) = m_2(W_3 - W_2)$$
$$m_1(h_1 - h_3) = m_2(h_3 - h_2)$$
$$\frac{m_1}{m_2} = \frac{W_3 - W_2}{W_1 - W_3} = \frac{h_3 - h_2}{h_1 - h_3}$$

Sensible cooling:

During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil. For moisture content to remain constant, the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature. However, in practice, the exit air temperature will be higher than the cooling coil temperature. Figure 5.19 shows the sensible cooling process O-A on a psychrometric chart. The heat transfer rate during this process is given by:

$$Q_c = m_a(h_O - h_A) = m_a c_{pm}(T_O - T_A)$$

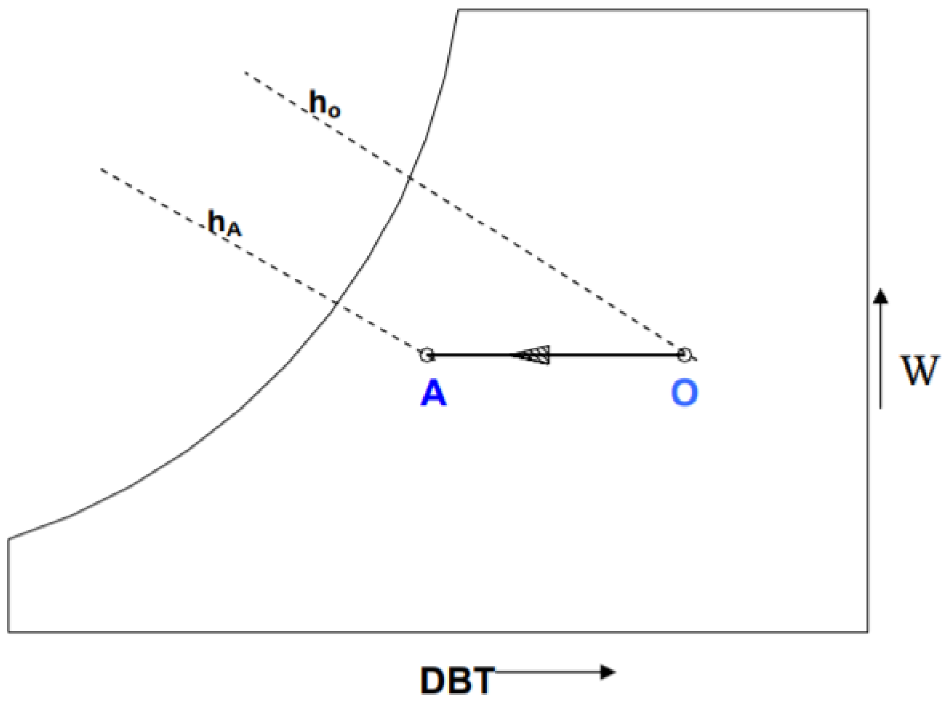


Fig 5.19 Sensible cooling process

Sensible heating (Process O-B):

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil. The heat transfer rate during this process is given by:

$$Q_h = m_a(h_B - h_O) = m_a c_{pm}(T_B - T_O)$$

where c_{pm} is the humid specific heat (≈1.0216 kJ/kg dry air) and ma is the mass flow rate of dry air (kg/s). Figure 5.20 shows the sensible heating process on a psychrometric chart.

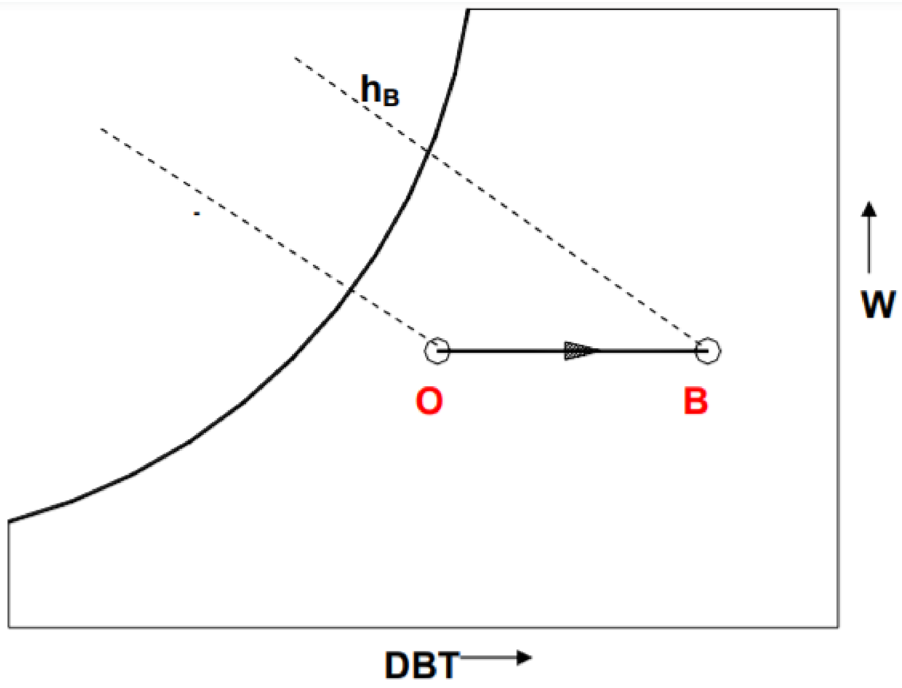


Fig 5.20 Sensible heating process

Cooling and dehumidification (Process O-C):

When moist air is cooled below its dew-point by bringing it in contact with a cold surface as shown in Fig.5.21, some of the water vapor in the air condenses and leaves the air stream as liquid, as a result both the temperature and humidity ratio of air decreases as shown. This is the process air undergoes in a typical air conditioning system. Although the actual process path will vary depending upon the type of cold surface, the surface temperature, and flow conditions, for simplicity the process line is assumed to be a straight line. The heat and mass transfer rates can be expressed in terms of the initial and final conditions by applying the conservation of mass and conservation of energy equations as given below: By applying mass balance for the water:

$$m_a.w_O = m_a.w_C + m_w$$

By applying energy balance:

$$m_a.h_O = Q_t + m_w.h_w + m_a.h_C$$

from the above two equations, the load on the cooling coil, Qt is given by:

$$Q_t = m_a(h_O - h_C) - m_a(w_O - w_C)h_w$$

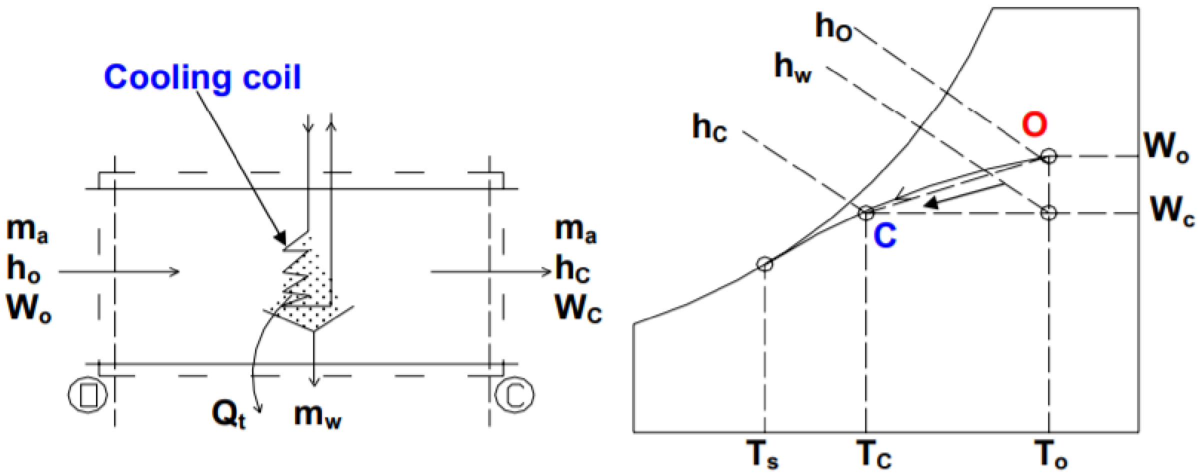


Fig 5.21 Cooling and Dehumidification process

the 2 term on the RHS of the above equation is normally small compared to the other terms, so it can be neglected. Hence,

$$Q_t = m_a(h_O - h_C)$$

It can be observed that the cooling and de-humidification process involves both latent and sensible heat transfer processes, hence, the total, latent and sensible heat transfer rates (Qt, Ql and Qs) can be written as:

$$\begin{aligned} & Q_t = Q_l + Q_s \\ \text{where} \quad & Q_l = m_a(h_O - h_w) = m_a.h_{fg}(w_O - w_C) \\ & Q_s = m_a(h_w - h_C) = m_a.c_{pm}(T_O - T_C) \end{aligned}$$

By separating the total heat transfer rate from the cooling coil into sensible and latent heat transfer rates, a useful parameter called Sensible Heat Factor (SHF) is defined. SHF is defined as the ratio of sensible to total heat transfer rate, i.e.,

$$SHF = Q_s / Q_t = Q_s / (Q_s + Q_l)$$

From the above equation, one can deduce that a SHF of 1.0 corresponds to no latent heat transfer and a SHF of 0 corresponds to no sensible heat transfer. A SHF of 0.75 to 0.80 is quite common in air conditioning systems in a normal dry-climate. A lower value of SHF, say 0.6, implies a high latent heat load such as that occurs in a humid climate.

By pass factor

$$BPF = \frac{T_C - T_S}{T_O - T_S}$$

It can be easily seen that, higher the by-pass factor larger will be the difference between air outlet temperature and the cooling coil temperature. When BPF is 1.0, all the air by-passes the coil and there will not be any cooling or de-humidification. In practice, the by-pass factor can be increased by increasing the number of rows in a cooling coil or by decreasing the air velocity or by reducing the fin pitch.

Alternatively, a contact factor(CF) can be defined which is given by:

CF=1-BPF

Heating and Humidification (Process O-D):

During winter it is essential to heat and humidify the room air for comfort. As shown in Fig.5.22., this is normally done by first sensibly heating the air and then adding water vapour to the air stream through steam nozzles as shown in the figure.

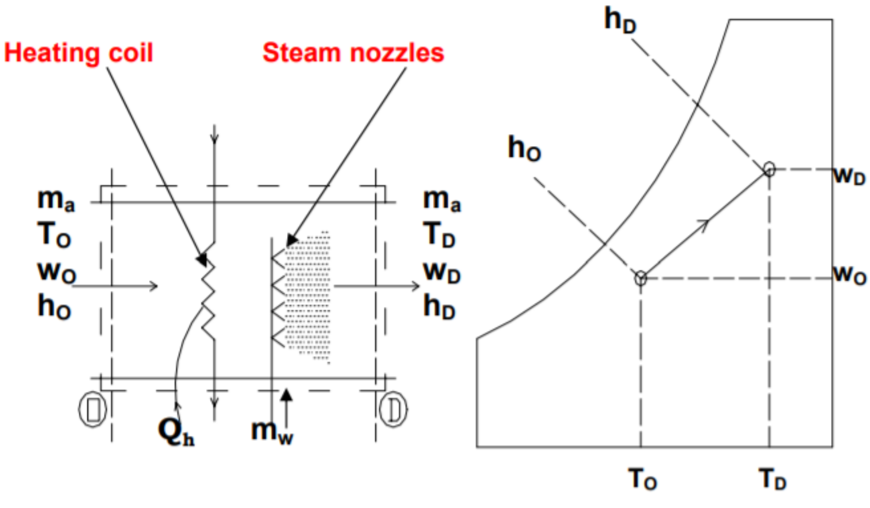


Fig 5.22

Mass balance of water vapor for the control volume yields the rate at which steam has to be added, i.e., mw:

$$m_w = m_a(w_D - w_O)$$

where ma is the mass flow rate of dry air.

From energy balance:

$$Q_h = m_a(h_D - h_O) - m_w h_w$$

where Q_h is the heat supplied through the heating coil and h_w is the enthalpy of steam. Since this process also involves simultaneous heat and mass transfer, we can define a sensible heat factor for the process in a way similar to that of a cooling and dehumidification process.

5.20 AIR CONDITIONING SYSTEMS

Air conditioning systems require basic arrangement for getting refrigeration effect through cooling coil followed by subsequent humidification/dehumidification and heating etc. in order to provide air conditioned space with air at desired temperature and humidity. Air conditioning systems require different arrangements depending upon the atmospheric air condition and comfort condition requirement. Such as summer air conditioning systems and inter air conditioning systems are different. These systems have different arrangement if outdoor conditions are hot and humid, hot and dry etc. Summer air conditioning system for hot and dry outdoor condition is given in Fig. 5.23. Here the comfort conditions may require delivery of air to air-conditioned space at about 25ºC DBT and 60% relative humidity where the outdoor conditions may be up to 40–44º C DBT and 20% relative humidity in Indian conditions. Generic arrangement has air blower which blows air across the air filter between (1) and (2). Air coming out from filter passes over cooling coils and is subsequently sent for humidification between states (3) and (4). Large size water particles carried by air are retained by water eliminator. Air finally coming out at state (5) is sent to air conditioned space. Here psychrometric representation is made considering negligible change in humidity in water eliminator.

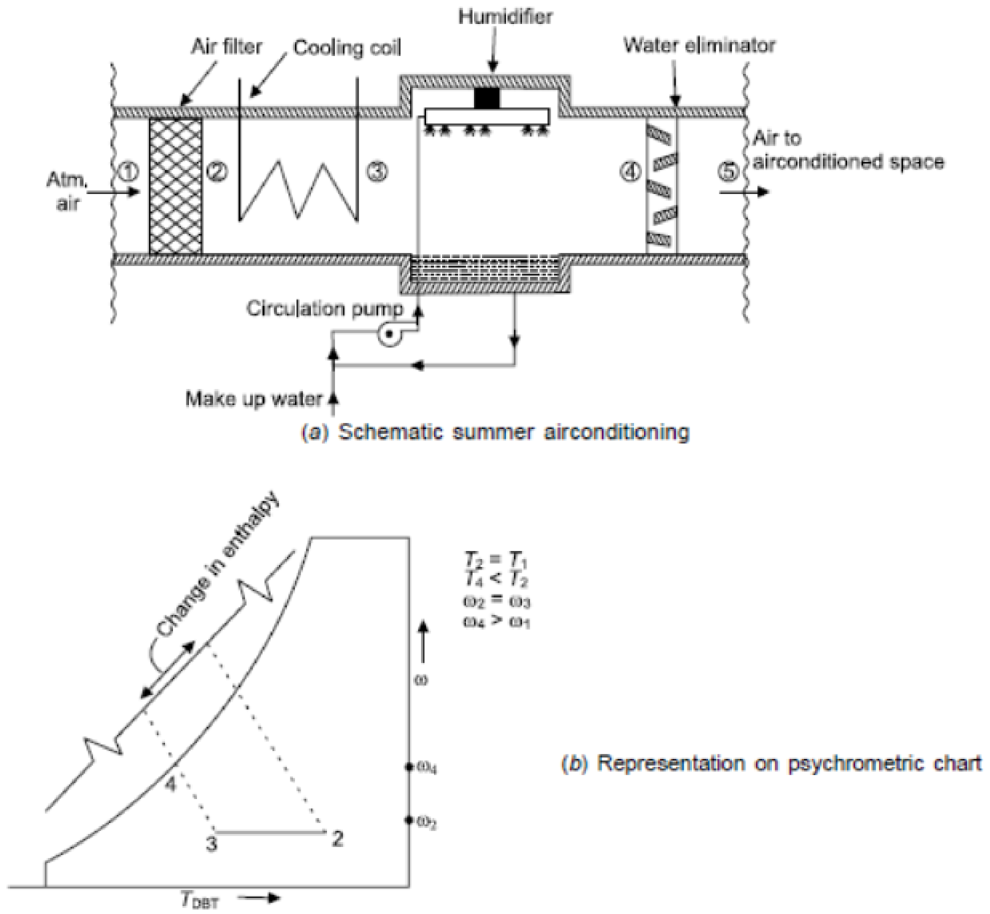


Fig 5.23 Summer air conditioning system

Winter Air Conditioning System:

In winter AC System, the inlet is heated by the heater, and in winter season due to less present in the air, we also need to add the moisture particle to the air, generally, a humidification system is added to maintain the moisture quantity.

Working of Winter Air Conditioning System:

In winter air conditioning, the air is heated and is accompanied by humidification. The outside air flows through a damper and mixes up with the recirculated air which is obtained from the conditioned space. The mixture here passes through a filter to remove dirt, dust, and other impurities.

The air now passes through a preheat coil to prevent possible freezing of water due to which dry bulb temperature increases to a very high value and the relative humidity drops to a low value.

This air is being pumped into the humidifier.

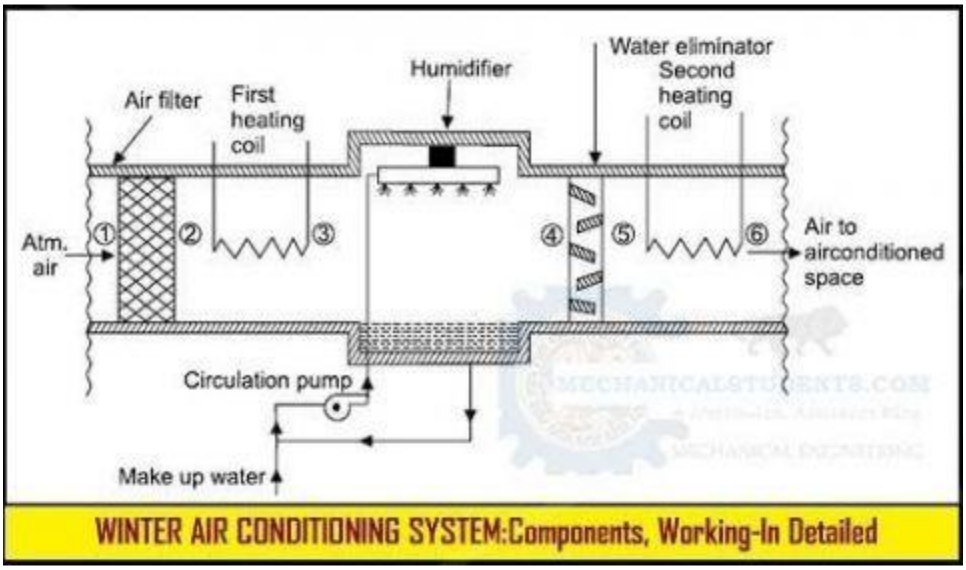


Fig 5.24 Winter air conditioning system

So, humidification of air (addition of moisture) is done and then the air is made to pass through a reheat coil to bring the air to the designed dry bulb temperature.Now the conditioned air is supplied to the conditioned space by mea fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators.The remaining part of the air known as recirculated air is again conditioned as shown in the figure.So it is again humidified due to which it reaches a point of 80% or 100% RH where the DBT is very low. So in order to get the desired dry bulb temperature, again the process of reheating is done where the desired percentage 40% RH is also obtained. A damper is used in order to control the area and have an intake of the required amount of air.

1. A 5 tonne refrigerator plant uses RR as refrigerant. It enters the compressor at -5°C as saturated vapour. Condension takes place at 32°C and there is no under cooling of refrigerant liquid. Assuming isentropic compression,determine COP of the plant, mass flow of refrigerant, power required to run the compressor in kw. The properties of R-12 are given table.

T(°C)	P(bar)	Enthalpy(kw/kg)		Entropy(KJ/kgk)
		hf	hg	Sg
32	7.85	130.5	264.5	1.542
-5	2.61	-	249.3	1.557

Solution:

Beginning of compression in dry and end of compression is superheated. So the P-h and T-S diagrams are

From table, at point 1

T_1=-5°C=268K

hg_1=2493kJ/Kg , Sg_1=1.557KJ/Kgk

At point 2

T_2=32°C=305k, hf_2=130.5KJ/Kg, hg_2=264.5, Sg_2=1.542KJ/KgK

From ph diagram, At point eqn(1) (dry).

At -5°C, i.e at 268k

$hg_{1=}$249.3KJ/Kg=hg

h_1=249.3KJ/Kg

At 32°C, i.e at 305K

$hg_{2=}$264.5KJ/Kg=h_2'

h_2'=264.5KJ/Kg

Entropy is constant during the compression process so,

$S_1=S_2$

From T- S diagram

At point (1) dry,

S_1=Sg at -5°C

$S_1=Sg_1$=1.557KJ/Kgk

$S_1=S_2$=1.557KJ/Kgk

At point (2) (super heated)

$S_1=S_2' + Cp \ln (T_2/T_2')$

$1.557=S_2'+1.884 \ln(T_2/305)$ --------(1)

$S_2' = Sg$ at 32°C.

$Sg_2 = 1.542 = S_2$

$S_2' = 1.542$ KJ/Kg k

$1.884 \ln (T_2/305) = 0.015$

$T_2 = 307.44$k

For super heated vapour the enthalpy is

$h_2 = h_2' + C_p(T_2 - T_1')$

$h_2 = 264.5 + 1.884\ (307.44 - 305)$

$h_2 = 269.1$ KJ/Kg

From P-h diagram, we know that,

$h_3 = h_4$

$h_3 = h_f$ at 32°C

$h_{f\,2} = 130.5 = h_f$

We Know that,

COP=Refrigeration effect / Work done= $(h_1 - h_4)/\ (h_2 - h_1)$

$= (2493-130.5)/(269.1-249.3) = 6$

Refrigeration effect = m× (h1- h_4)

$m = (2\times210)/\ (249.3-130.5)$

m = 8.84Kg/min

Work done = Refrigeration effect/ cop

$= (2\times210)/6 = 175$ KJ/min

Power = 2.92kw.

2. **A refrigerator works between -7°C and 27°C the vapour is dry at the end of adiabatic compression. Assuming there is no under cooling determine (i) cop (ii) power of the compressor to remove a heat load of 12140KJ/hr.The properties of refrigerant are given in table.**

T(°C)	sensible Heat (h_f)	Latent heat(h_{fg}) KJ/Kgk)	Entropy of liquid (KJ/Kgk)	Entropy of vapour Sg (KJ/Kgk)
-7	-29.3	1297.9	-0.109	4.748
27	1117.23	1172.3	0.427	4.333

Solution:

The vapour is dry at end of compression i.e, beginning of compression is wet and of compression is dry saturated.

At point (1)

T_1=-7°C=266k, h_{fg1}=1297.9 KJ/Kg, S_{f1}=-0.109 KJ/KgK

h_{f1}=-29.3 KJ/Kg, S_{fg1}=4.478 KJ/KgK

At point (2)

T_2=27°C=300k, h_{fg2}=1172.3 KJ/Kg, S_{f2}= 0.427KJ/KgK

h_{f2}=117.23 KJ/Kg, S_{fg2}=4.333KJ/KgK

We point $S_1=S_2$

At point (1) (wet)

$S_1=S_{wet}=S_{f1}+x_1+S_{fg1}$

$S_1= -0.109+x_1(S_{fg1}- S_{f1})$ ($S_{fg}=S_g-S_f$)

$S_1= -0.109+x_1(4.857)$

At point (2) (dry)

$S_2=S_{g2}$=4.33KJ/KgK

S_2=4.33KJ/KgK

$S_1=S_2$ So,4.33= $-0.109+x_1(4.857)$

Dryness fraction

x_1=0.913

At point (1) (wet)

$h_1 = h_{f1} + x_1 \times h_{fg1}$

$h_1 = -29.3 + 0.913 \times 1297.3$

$h_1 = 1156.3$ KJ/Kg

At point (2) (dry)

$h_2 = h_{f2} + h_{g2}$

$h_2 = 117.23 + 1172.3$

$h_2 = 1289.53$ KJ/Kg

From P-h diagram

$h_3 = h_4$

$h_3 = h_{f2}$

$h_3 = 1172.3$ KJ/Kg

$h_4 = 117.23$ KJ/Kg

COP= $(h_1 - h_4) / (h_2 - h_1)$ = (1156.3-117.23) / (1289.53-1156.3) = 7.7

Work done = Heat removed/ COP

= 12140/7.7

Power = 0.43 KJ/hr

3.Air enters the compressor of air craft system at 100kpa, 277k and is compressed to 300kpa with an isentropic efficiency of 72%. After being cooled to 328k and air expands is 100kpa and an η_{Isen}=78% the load is 3 tons and find COP, power,mass flow rate.

Given data:

P_1= 100kpa, T_3=38k

T_1= 277k, P_4=100kpa

P_2= 300kpa, η_T=78%

η_c= 72%

Solution:

process 1-2 Isentropic compression

$T_2= (P_2/P_1)^{\gamma-1/\gamma} \times T_1$

$T_2= (300/100)^{1.4-1/1.4}$

$T_2=379.14k$

$\eta_c=(T_2-T_1)/(T_2'-T_1)$

$0.72= (379.14-277)/ (T_2'-277)$

$T_2=418.86k$

Process 3-4 isentropic compression

$T_3/T_4= (P_3/P_4)^{\gamma-1/\gamma}$

$328/T_4= (300/100)^{\gamma-1/\gamma}$

$T_4=239.64k$

$\eta_t=(T_3-T_4')/(T_3-T_4)$

$0.78= (328-T_4')/ (328-239.64)$

$T_4'= 259.08k$

$COP= (T_1-T_4')/ (T_2'-T_1)$

$COP= (277-259.08)/ (418.86-277) =0.17$

1 tonne= 3.5kw of heat

3tonne= 3×3.5=10.5kw

Energy balance.

Heat energy absorbed by I_{ce}=Heat rejected by air

$= m\times C_p\times (T_1-T_4')$

$10.5= m_a\times 1.005\times(277-259.08)$

Mass of air, $m_a= 0.583$Kg/sec.

Power, $P=m_a \times Cp_a \times (T_2'-T_1)$

$= 0.583 \times 1.005 \times (418.86-277)$

$= 83.12kW$

4. **An ammonia refrigerator process 20tons of ice per day from and at 0°C.The condensation and evaporation takes at 20°C and -20°C respectively the temperature of the vapour at the end of Isentropic compression is 50°C and there is no under cooling of the liquid.COP=70% of theoretical COP. Determine (i) Rate of NH_3 circulation (ii) size of compressor, N=240rpm,L=D, η_{vol}=80%. Take Laten heat of I_{cc}=335kJ/Kg, C_p= 2.8 kJ/Kg,**

Vs_1=0.624m^{3}kg. Use the following properties of ammonia.

Sat.Temp(°C)	Enthalpy(kJ/Kg)		Entropy(kJ/Kgk)		
	h_f	h_g	S_f	S_g	
20	**274.98**	**1461.58**	**1.0434**	**5.0919**	
20	**89.72**	**1419.05**	**0.3682**	**5.6204**	**[Apr 2003]**

Given data: 20 tons of Icc per day at °C

T_3= 20°C

T_1=-20°C

T3=50°C

COP =70% of theoretical cop

N=240rpm

L=D

h_v=80%

Latent heat of Ice=335KJ/Kg

C_p=2.8KJ/Kgk

V_{s1}=0.624m^3/kg

Solution:-

The refrigeration effect=20×3.5=77.55kw

h_1=1419.05KJ/Kg

h_{g2}=1461.58KJ/Kg at 20°C

h_{f3}=274.98 KJ/Kg

$h_2=h_{g2}+C_p\ (T_2\text{-}20)$

$h_2=1461.58+2.8(50\text{-}20)$

$h_2=1545.58$ KJ/Kg

COP= $(h_1\text{-}h_{f3})/(h_1\text{-}h_2)$

= (1419.05-274.98)/(1545.58-1419.05)

=9.04.

5. Explain the working principle of vapor compression refrigeration system with neat sketch.

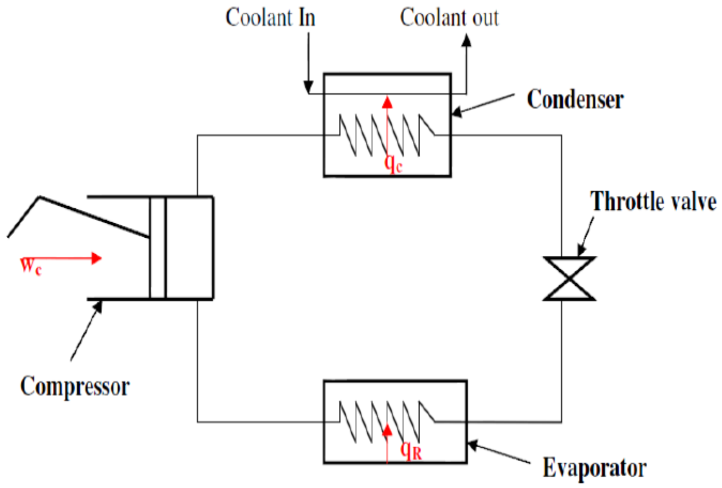
Schematic layout of vapour compression system

Process 1-2: Isentropic compression of the refrigerant from state 1 to state 2. During this process work done is done on the refrigerant by the surroundings. At the end of the process the refrigerant will be in super heated vapour state.

Process 2-3: Constant pressure condensation of the refrigerant in the condenser till it becomes a saturated liquid.

Process 3-4: Throttling expansion of the refrigerant from condenser pressure to the evaporator pressure. Process 4-1: Constant pressure vapourisation of the refrigerant in the evaporator till it becomes a dry saturated vapour. During this process heat is absorbed by the refrigerant from the place to be refrigerated. Applying steady flow steady state energy equation to the evaporator and neglecting the changes in kinetic and potential energies we have Refrigeration effect = QR = m (h1 – h4) Since process 3-4 is a throttling process, h4 = h3. Hence QR = m (h1 – h3) Similarly, by applying steady flow, steady state energy equation to compressor we getCompressor work input = Wc = m (h2 – h1)

Hence COP = Qr / Wc = (h1 – h4) / (h2 – h1)



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

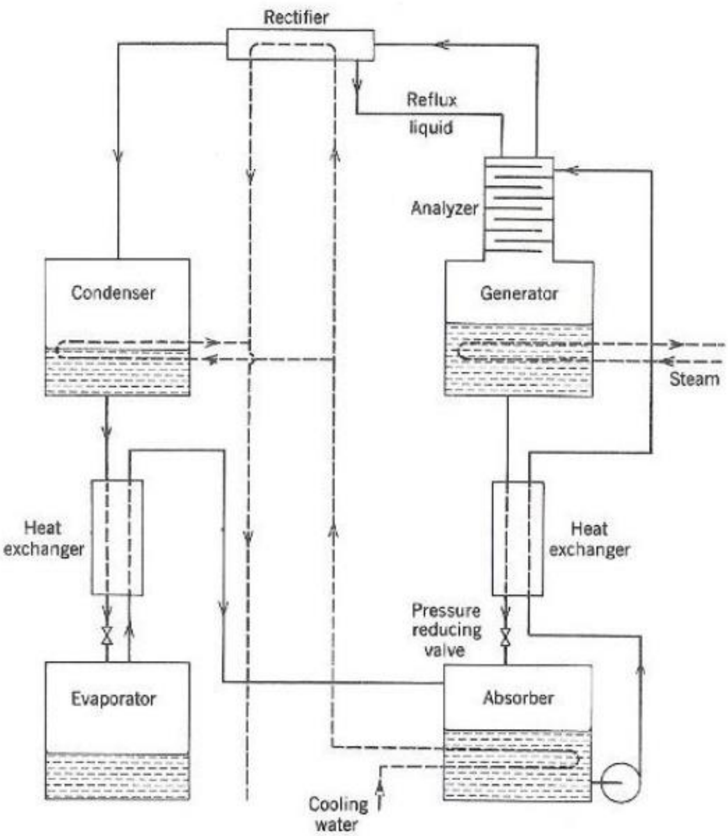
6.Explain and working principles of Ammonia – water vapour absorption refrigeration system with neat sketch. [May 2011]

1) Evaporator: It is in the evaporator where the refrigerant pure ammonia (NH3) in liquid state produces the cooling effect. It absorbs the heat from the substance to be cooled and gets evaporated. From here, the ammonia passes to the absorber in the gaseous state.

2) Absorber: In the absorber the weak solution of ammonia-water is already present. The water,

used as the absorbent in the solution, is unsaturated and it has the capacity to absorb more ammonia gas. As the ammonia from evaporator enters the absorber, it is readily absorbed by water and the strong solution ofammonia-water is formed. During the process of absorption heat is liberated which can reduce the ammonia absorption capacity of water; hence the absorber is cooled by the cooling water. Due to absorption of ammonia, strong solution of ammonia-water is formed in the absorber.

3) Pump: The strong solution of ammonia and water is pumped by the pump at high pressure to the generator.



Schematic layout of Ammonia –water vapour absorption system

4) Generator: The strong solution of ammonia refrigerant and water absorbent are heated by the external source of heat such as steam or hot water. It can also be heated by other sources like natural gas, electric heater, waste exhaust heat etc. Due to heating the refrigerant ammonia gets vaporized and it leaves the generator. However, since water has strong affinity for ammonia and its vaporization point is quite low some water particles also get carried away with ammonia refrigerant, so it is important to pass this refrigerant through analyzer.

5) Analyzer: One of the major disadvantages of the ammonia-water vapor absorption refrigeration system is that the water in the solution has quite low vaporizing temperature, hence when ammonia refrigerant gets vaporized in the generator some water also gets vaporized. Thus the ammonia refrigerant leaving the generator carries appreciable amount of water vapor. If this water vapor is allowed to be carried to the evaporator, the capacity of the refrigeration system would reduce. The water vapor from ammonia refrigerant is removed by analyzer and the rectifier. The analyzer is a sort of the distillation column that is located at the top of the generator. The analyzer consists of number of plates positioned horizontally. When the ammonia refrigerant along with the water vapor particles enters the analyzer, the solution is cooled. Since water has higher saturation temperature, water vapor gets condensed into the water particles that drip down into the generator. The ammonia refrigerant in the gaseous state continues to rise up and it moves to the rectifier.

6) Rectifier or the reflex condenser: The rectifier is a sort of the heat exchanger cooled by the water, which is also used for cooling the condenser. Due to cooling the remaining water vapor mixed with the ammonia refrigerant also gets condensed along with some particles of ammonia. This weak solution of water and ammonia drains down to the analyzer and then to the generator.

7) Condenser and expansion valve: The pure ammonia refrigerant in the vapor state and at high

pressure then enters the condenser where it is cooled by water. The refrigerant ammonia gets converted into the liquid state and it then passes through the expansion valve where its temperature and pressure falls down suddenly. Ammonia refrigerant finally enters the evaporator, where it produces the cooling effect. This cycle keeps on repeating continuously. Meanwhile, when ammonia gets vaporized in the generator, weak solution of ammonia and water is left in it. This solution is expanded in the expansion valve and passed back to the absorber and its cycle repeats.

7. Comparison between vapour compression and vapour absorption systems.

Compression systems	**Absorption systems**
Work operated	Heat operated
High COP	Low COP
Performance very sensitive to evaporator temperatures.	Performance not very sensitive to evaporator temperatures.
System COP reduces considerably at part loads.	COP does not reduce significantly with load.
Liquid at the exit of evaporator may damage compressor.	Presence of liquid at evaporator exit is not a serious problem.
Performance of sensitive to evaporator superheat.	Evaporator superheat is not very important.
Many moving parts	Very few moving parts
Regular maintenance required	Very low maintenance required
Higher noise and vibration	Less noise and vibration.

8. List out the Properties of Refrigerants

Toxicity: It obviously desirable that the refrigerant have little effect on people.

Inflammability: Although refrigerants are entirely sealed from the atmosphere, leaks are bound to develop. If the refrigerant is inflammable and the system is located where ignition of the refrigerant may occur, a great hazard is involved.

Boiling Point: An ideal refrigerant must have low boiling temperature at atmospheric pressure.

Freezing Point: An ideal refrigerant must have a very low freezing point because the refrigerant should not freeze at low evaporator temperatures.

Evaporator and condenser pressure: In order to avoid the leakage of the atmosphere air and also to enable the detection of the leakage of the refrigerant, both the Evaporator and condenser pressure should be slightly above the atmosphere pressure.

Chemical Stability: An ideal refrigerant must not decompose under operating conditions.

Latent heat of Evaporation: The Latent heat of Evaporation must be very high so that a minimum amount of refrigerant will accomplish the desired result; in other words, it increases the refrigeration effect.

Specific Volume: The Specific Volume of the refrigerant must be low. The lower specific volume of the refrigerant at the compressor reduces the size of the compressor.

Specific heat of liquid vapour: A good refrigerant must have low specific heat when it is in liquid state and high specific heat when it is vaporized.

Viscosity: The viscosity of the refrigerant t both the liquid and vapour state must be very low as improved the heat transfer and reduces the pumping pressure.

Corrosiveness: A good refrigerant should be non-corrosive to prevent the corrosion of the metallic parts of the refrigerator.

Odour: A good refrigerant must be odourless, otherwise some foodstuff such as meat, butter, etc loses their taste.

Oil solvent properties: A good refrigerant must be not react with the lubricating oil used in the refrigerator for lubricating the parts of the compressor.

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