25

Modern Steam Turbines

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25.1. Introduction

In the last three chapters, we have discussed impulse turbines, reaction turbines and their performance. In these chapters, we discussed, apart from other things, power developed by the turbines and their efficiencies. The scientists and engineers, working in research centres all over the world, concentrated their attention to produce more power and to improve efficiencies of these turbines. They have listed a number of methods for this purpose, but the following are important from the subject point of view :

1. Reheating of steam, 2. Regenerative feed heating, and 3. Binary vapour plants.

All the above mentioned methods will be discussed, in detail, in this chapter.

25.2. Reheating of Steam

We have already discussed that efficiency of the ordinary Rankine cycle can be improved by increasing the pressure and temperature of the steam entering into the turbine. A little consideration will show, that the increase in the initial steam pressure will increase the expansion ratio, and steam will become quite wet at the end of expansion. As a matter of fact, it is not desirable that the steam may become wet at the end of expansion. The wet steam causes erosion of the turbine blades and increases internal losses. This will ultimately reduce the blade efficiency of the turbine.

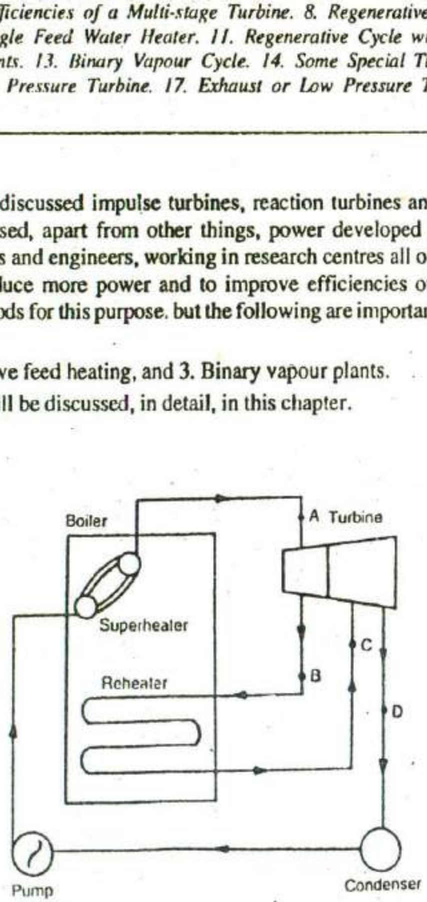


Fig. 25.1. Reheating of steam.

The above mentioned difficulty may be overcome by reheating of the steam. In this system, the steam is removed from the turbine when it becomes wet. It is then reheated at a constant pressure by the flue gases, until it is again in the superheated state. It is then returned to the next stage in the turbine. The schematic diagram of the process is shown in Fig. 25.1.

25.3. Advantages of Reheating of Steam

The reheating of steam in a turbine has the following advantages :

1. It increases the work done through the turbine.
2. It increases the efficiency of the turbine.
3. It reduces the erosion of the blades, because of increase in dryness fraction of steam at exhaust.
4. The amount of water required in the condenser of the turbine is reduced, due to reduction in the specific steam consumption.

25.4. Reheat Cycle

In a reheat cycle, the steam enters the turbine in a superheated state at point *A*. The steam then expands isentropically while flowing through the turbine, as shown by the vertical line *AB* in Fig. 25.2.

After expansion, the steam becomes wet, which is reheated at a constant pressure generally up to the same temperature as that at *A* shown by the point *C*, where it is again in superheated state. The steam again expands isentropically while flowing through the next stage of the turbine as shown by the vertical line *CD* in Fig. 25.2.

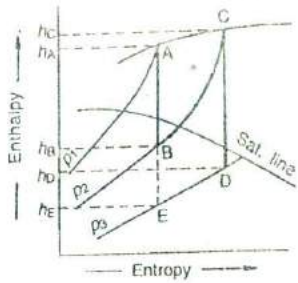


Fig. 25.2. Reheat cycle on *h-s* diagram

Now consider a steam turbine with a reheating system as shown in Fig. 25.2.

Let h_A = Enthalpy or total heat of steam at *A*,

h_B, h_C, h_D = Corresponding values at *B*, *C* and *D*,

h_{fD} = Enthalpy or sensible heat of water at *D*.

We know that the total heat supplied to steam is the sum of the total heat at *A* and the heat supplied during reheating between *B* and *C*.

$$\therefore \text{ Total heat supplied} = \text{Total heat at } A + \text{Heat supplied between } B \text{ and } C$$

$$= h_A + [(h_C - h_B) - h_{fD}]$$

We also know that work done

$$= \text{Total heat drop} = (h_A - h_B) + (h_C - h_D)$$

and efficiency,

$$\eta = \frac{\text{Work done}}{\text{Total heat supplied}} = \frac{(h_A - h_B) + (h_C - h_D)}{h_A + [(h_C - h_B) - h_{fD}]}$$

Notes : 1. If there had been no reheating of steam, the expansion through the turbine would have been along the line *AE*. In that case,

Efficiency, $$\eta = \frac{h_A - h_E}{h_A - h_{fE}}$$ ... (Rankine efficiency)

where h_E = Total heat of steam at *E*, and

h_{fE} = Total heat of water at *E*.

2. For simplicity, we have taken only one stage of reheating. But in actual practice, there may be more than one stage of reheating.

Example 25.1. *Steam at a pressure of 15 bar and 250° C is first expanded through a turbine to a pressure of 4 bar. It is then reheated at a constant pressure to the initial temperature of 250° C and is finally expanded to 0.1 bar. Using Mollier chart, estimate the work done per kg of steam flowing through the turbine and the amount of heat supplied during the process of reheat.*

Also find the work output when the expansion is direct from 15 bar to 0.1 bar without any reheat. Assume all expansion processes to be isentropic.

Solution. Given : $p_1 = 15$ bar ; $T_1 = 250°$ C ; $p_2 = 4$ bar ; $T_2 = 250°$ C ; $p_3 = 0.1$ bar

The reheating of steam is represented on the Mollier chart as shown in Fig. 25.3. From the chart, we find that

$h_A = 2930$ kJ/kg ; $h_B = 2660$ kJ/kg ; $h_C = 2965$ kJ/kg ; $h_D = 2345$ kJ/kg ; and $h_E = 2130$ kJ/kg

From steam tables, corresponding to a pressure of 0.1 bar, we find that sensible heat of water at *D*,

$$h_{fD} = h_{fE} = 191.8 \text{ kJ/kg}$$

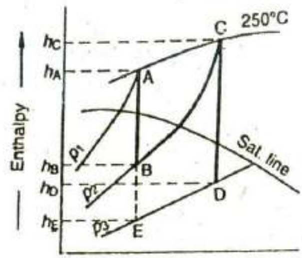


Fig. 25.3

Workdone per kg of steam

We know that workdone per kg of steam,

$$w = (h_A - h_B) + (h_C - h_D)$$

$$= (2930 - 2660) + (2965 - 2345) = 890 \text{ kJ/kg } \textbf{Ans.}$$

Heat supplied during the process of reheat

We know that the heat supplied during the process of reheat,

$$h = \text{Heat supplied between } B \text{ and } C$$

$$= (h_C - h_B) - h_{fD} = (2965 - 2660) - 191.8 = 113.2 \text{ kJ/kg Ans.}$$

Work output when the expansion is direct

The direct expansion from 15 bar to 0.1 bar is shown by the line *AE* in Fig. 25.3.

We know that work output

$$= \text{Total heat drop} = h_A - h_E = 2930 - 2130 = 800 \text{ kJ/kg } \textbf{Ans.}$$

Example 25.2. *In a thermal plant, the steam is supplied at a pressure of 30 bar and temperature of 300° C to the high pressure side of steam turbine where it is expanded to 5 bar. The steam is then removed and reheated to 300° C at a constant pressure. It is then expanded to the low pressure side of the turbine to 0.5 bar. Find the efficiency of the cycle with and without reheating.*

Solution. Given : $p_1 = 30$ bar ; $T_1 = 300°$ C ; $p_2 = 5$ bar ; $T_2 = 300°$ C ; $p_3 = 0.5$ bar

Efficiency of the cycle with reheating

The reheating of steam is represented on the Mollier chart as shown in Fig. 25.4. From the chart, we find that

$h_A = 2990$ kJ/kg ; $h_B = 2625$ kJ/kg ; $h_C = 3075$ kJ/kg ; $h_D = 2595$ kJ/kg ; and $h_E = 2280$ kJ/kg

From steam tables, corresponding to a pressure of 0.5 bar, we find that sensible heat of water at *D*,

$$h_{fD} = h_{fE} = 340.6 \text{ kJ/kg}$$

We know that efficiency of the cycle with reheating,

$$\eta_1 = \frac{(h_A - h_B) + (h_C - h_D)}{h_A + [(h_C - h_B) - h_{fD}]} = \frac{(2990 - 2625) + (3075 - 2595)}{2990 + [(3075 - 2625) - 340.6]}$$

$$= \frac{845}{3099.4} = 0.273 \text{ or } 27.3\% \textbf{ Ans.}$$

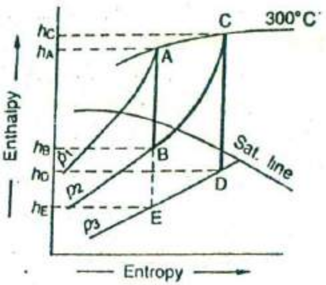


Fig. 25.4

Efficiency of the cycle without reheating

We know that efficiency of the cycle without reheating,

$$\eta_2 = \frac{h_A - h_E}{h_A - h_{fE}} = \frac{2990 - 2280}{2990 - 340.6}$$

$$= 0.268 \text{ or } 26.8\% \textbf{ Ans.}$$

25.5. Multi-stage Turbines

We have already discussed a two-stage impulse turbine in Art. 22.12. In this turbine, we have seen that the steam after leaving the moving blade is made to flow through a fixed ring and again it impinges on the blades fixed to the second moving ring. If the steam, from the second moving ring, is made to flow into the condenser, it is known as two-stage turbine. But sometimes, we make the steam to pass through a number of stages in order to get more work (or precisely to develop more power). Such a turbine is known as *multi-stage turbine.*

25.6. Reheat Factor

It is an important term used for the multi-stage turbines, which may be broadly defined as the ratio of cumulative heat drop to the isentropic heat drop. Now consider a multi-stage turbine (say three-stage turbine) whose reheat factor is required to be found out.

Let p_1 = Initial pressure of the steam,

p_2 = Pressure of steam leaving the first stage,

p_3 = Pressure of steam leaving the second stage,

p_4 = Final pressure of the steam.

t_1 = Initial temperature of the steam, and

η = Stage efficiency for each stage of the turbine.

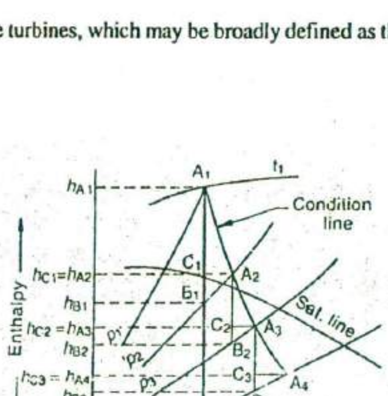


Fig. 25.5. Reheat factor for a three-stage turbine.

Now let us draw the expansion of steam in three stages on a Mollier chart, as shown in Fig. 25.5, as discussed below :

1. First of all, locate the point A_1 with the help of initial pressure (p_1) and temperature (t_1) of the steam. Now find out the enthalpy of steam at A_1 (*i.e.* h_{A1}).

2. From A_1, draw a vertical line A_1B_1 meeting the pressure line (p_2) at B_1 representing isentropic expansion of the steam in the first stage. Now find out the enthalpy of steam at B_1 (*i.e.* h_{B1}).

3. Now cut off B_1C_1 equal to the blade friction in the first stage [or in other words, cut off A_1C_1 equal to $\eta \times (h_{A1} - h_{B1})$]. Through C_1 draw a horizontal line C_1A_2 meeting the pressure line (p_2) at A_2, which gives the final condition of the steam discharged from the first stage. Now find out the enthalpy of steam at A_2 (*i.e.* h_{A2}).

4. Similarly, from A_2 draw a vertical line A_2B_2 to meet the pressure line (p_3) at B_2 representing isentropic expansion of steam in the second stage. Now cut off B_2C_2 equal to the blade friction in the second stage. Through C_2 draw a horizontal line C_2A_3 to meet the pressure line (p_3) at A_3, which gives the final condition of the steam discharged from the second stage. Now find out the enthalpy of steam at A_3 (*i.e.* h_{A3}).

5. Similarly, locate the point A_4 on the pressure line (p_4), which gives the final condition of the steam discharged from the third stage and find out the enthalpy of steam at A_4 (*i.e.* h_{A4}).

Now a draw a smooth curve through the points A_1, A_2, A_3 and A_4. Since these points represent the final condition of steam at the end of each stage, therefore the curve is known as *condition curve*. A little consideration will show, that if the friction is neglected, the isentropic heat drop through all these stages will be represented by the vertical line A_1D.

It will be interesting to know, from the Mollier diagram, that the pressure lines diverge from left to right, which shift the isentropic expansion lines at each stage slightly towards the right side in the diagram. Or in other words, enthalpy or heat drop (as represented by the lines A_1B_1, A_2B_2, A_3B_3) is slightly increased. The sum of increased heat drops is known as *cumulative heat drop*.

Now the ratio of cumulative heat drop to the isentropic heat drop is known as *reheat factor*. Mathematically, reheat factor,

$$\text{R.F.} = \frac{\text{Cumulative heat drop}}{\text{Isentropic heat drop}}$$

$$= \frac{A_1B_1 + A_2B_2 + A_3B_3}{A_1D} = \frac{\Sigma AB}{A_1D}$$

25.7. Efficiencies of a Multi-stage Turbine

The following efficiencies of a multi-stage turbine are important from subject point of view :

1. *Stage efficiency.* It is the ratio of the work done on the rotor (or useful heat drop) in a stage to the isentropic heat drop for the same stage. Mathematically, stage efficiency,

$$\eta_s = \frac{\text{Useful heat drop in one stage}}{\text{Isentropic heat drop for the same stage}}$$

$$= \frac{A_1C_1}{A_1B_1} = \frac{A_2C_2}{A_2B_2} = \frac{A_3C_3}{A_3B_3}$$

2. *Internal efficiency.* It is the ratio of the total work done on the rotor (or total useful heat drop) to the total isentropic heat drop. It accounts for all the losses due to friction etc. Mathematically, internal efficiency,

$$\eta_i = \frac{\text{Total useful heat drop}}{\text{Total isentropic heat drop}} = \frac{A_1C_1 + A_2C_2 + A_3C_3}{A_1D}$$

$$= \frac{(h_{A1} - h_{C1}) + (h_{A2} - h_{C2}) + (h_{A3} - h_{C3})}{h_{A1} - h_D} = \frac{h_{A1} - h_{C3}}{h_{A1} - h_D}$$

Note : The internal efficiency is sometimes known as *isentropic efficiency* or *turbine efficiency*.

3. *Efficiency ratio.* When the external losses due to friction at the bearing etc. are also considered with internal losses, the turbine efficiency is termed as *efficiency ratio.* It is defined as the ratio of the total work produced at the driving end of the shaft per kg of steam to the total isentropic heat drop across the turbine. Mathematically, efficiency ratio,

$$\text{E.R.} = \frac{\text{Total work produced at the driving end of shaft}}{\text{Total isentropic heat drop}}$$

$$= \frac{\text{Total useful heat drop}}{\text{Total isentropic heat drop}}$$

4. *Overall thermal efficiency.* It is the ratio of the work delivered at the turbine shaft to the heat supplied. This efficiency covers both the internal and external losses. Mathematically, overall thermal efficiency,

$$\eta_O = \frac{\text{Work delivered at the turbine shaft}}{\text{Heat supplied}} = \frac{\text{Total useful heat drop}}{\text{Heat supplied}}$$

$$= \frac{A_1C_1 + A_2C_2 + A_3C_3}{h_{A1} - h_{fD}}$$

$$= \frac{(h_{A1} - h_{C1}) + (h_{A2} - h_{C2}) + (h_{A3} - h_{C3})}{h_{A1} - h_{fD}} = \frac{h_{A1} - h_{C3}}{h_{A1} - h_{fD}}$$

5. *Rankine efficiency.* It is the ratio of the isentropic heat drop to the heat supplied. Mathematically, Rankine efficiency,

$$\eta_R = \frac{\text{Isentropic heat drop}}{\text{Heat supplied}} = \frac{h_{A1} - h_D}{h_{A1} - h_{fD}}$$

Example 25.3. *The steam is supplied to a three stage turbine at 30 bar and 350° C. The steam leaves the first stage at 7 bar ; second stage at 1 bar ; and finally at 0.1 bar. If each stage has an efficiency of 0.7, determine : 1. Rankine efficiency, 2. The final condition of steam, 3. Reheat factor, and 4. Overall thermal efficiency.*

Solution. Given : $p_1 = 30$ bar ; $T_1 = 350°$ C ; $p_2 = 7$ bar ; $p_3 = 1$ bar ; $p_4 = 0.1$ bar ; $\eta = 0.7$

Now let us draw expansion of the steam in three stages on the Mollier chart, as shown in Fig. 25.6, as discussed below :

1. First of all, locate a point A_1 at the intersection of initial pressure (30 bar) and initial temperature (350° C). We find that enthalpy of steam at A_1, *i.e.* $h_{A1} = 3120$ kJ/kg.

2. From A_1, draw a vertical line $A_1 B_1$ meeting the pressure line of 7 bar at B_1 representing isentropic expansion of steam in the first stage. We find that enthalpy of steam at B_1, *i.e.* $h_{B1} =$ 2790 kJ/kg.

3. Now cut off, A_1C_1 equal to $\eta\,(h_{A1} - h_{B1}) = 0.7\,(3120 - 2790) = 231$ kJ/kg. Therefore, enthalpy of steam at C_1 *i.e.* $h_{C1} = h_{A1} - 231 = 3120 - 231 = 2889$ kJ/kg. Through C_1, draw a horizontal line $C_1 A_2$ meeting the pressure line of 7 bar at A_2. Now the enthalpy of steam at A_2, *i.e.* $h_{A2} = h_{C1} = 2889$ kJ/kg.

4. Similarly from A_2, draw a vertical line A_2B_2 meeting the pressure line of 1 bar at B_2. We find that enthalpy of steam at B_2, *i.e.* $h_{B2} = 2535$ kJ/kg.

5. Now cut off A_2C_2 equal to $\eta\,(h_{A2} - h_{B2}) = 0.7\,(2889 - 2535) = 248$ kJ/kg. Therefore, enthalpy of steam at C_2, *i.e.* $h_{C2} = h_{A2} - 248 = 2889 - 248 = 2641$ kJ/kg. Through C_2, draw a

horizontal line C_2A_3 meeting the pressure line of 1 bar at A_3. Now the enthalpy of steam at A_3, *i.e.* $h_{A3} = h_{C2} = 2641$ kJ/kg.

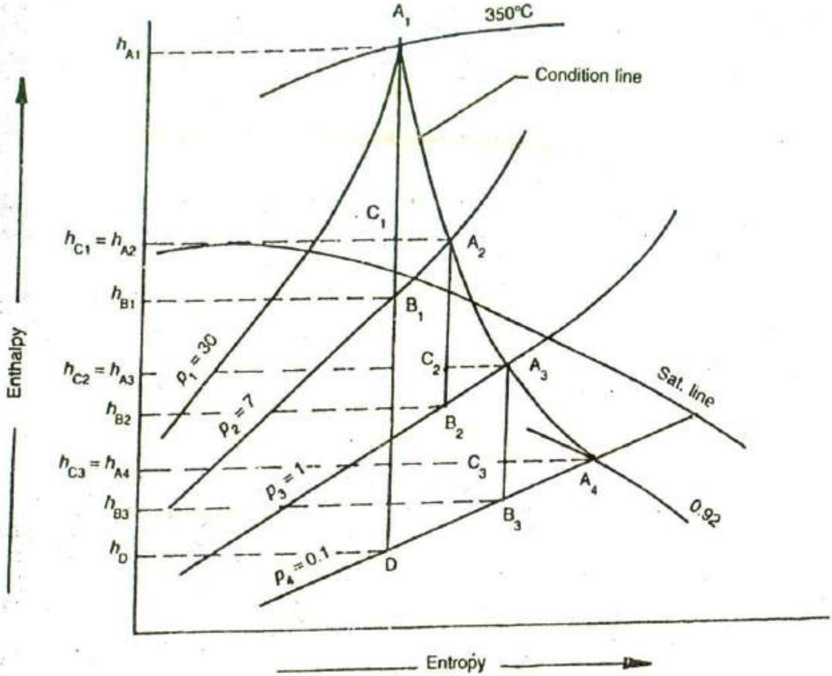


Fig. 25.6

6. Again from A_3, draw a vertical line A_3B_3 meeting the pressure line of 0.1 bar at B_3. We find that the enthalpy of steam at B_3, *i.e.* $h_{B3} = 2300$ kJ/kg.

7. Now cut off A_3C_3 equal to $\eta\,(h_{A3} - h_{B3}) = 0.7\,(2641 - 2300) = 239$ kJ/kg. Therefore enthalpy of steam at C_3 *i.e.* $h_{C3} = h_{A3} - 239 = 2641 - 239 = 2402$ kJ/kg. Through C_3, draw a horizontal line C_3A_4 meeting the pressure line of 0.1 bar at A_4. Now the enthalpy of steam at A_4 *i.e.* $h_{A4} = h_{C3} = 2402$ kJ/kg.

Now let us tabulate the values of enthalpy of steam as found above from the Mollier diagram for convenience.

$h_{A1} = 3120$ kJ/kg ; $h_{C1} = h_{A2} = 2889$ kJ/kg ; $h_{B1} = 2790$ kJ/kg ; $h_{C2} = h_{A3} = 2641$ kJ/kg ;

$h_{B2} = 2535$ kJ/kg ; $h_{C3} = h_{A4} = 2402$ kJ/kg ; $h_{B3} = 2300$ kJ/kg ; $h_D = 2140$ kJ/kg

From steam tables, corresponding to a pressure of 0.1 bar, we find that the enthalpy of water,

$$h_{fD} = 191.8 \text{ kJ/kg}$$

Rankine efficiency

We know that Rankine efficiency,

$$\eta_R = \frac{h_{A1} - h_D}{h_{A1} - h_{fD}} - \frac{3120 - 2140}{3120 - 191.8} = 0.335 \text{ or } 33.5\% \textbf{ Ans.}$$

2. *Final condition of steam*

From Mollier diagram, we find that the final condition or dryness fraction of steam at A_4 is 0.92. **Ans.**

3. *Reheat factor*

We know that reheat factor,

$$\text{R.F.} = \frac{A_1B_1 + A_2B_2 + A_3B_3}{A_1D} = \frac{(h_{A1} - h_{B1}) + (h_{A2} - h_{B2}) + (h_{A3} - h_{B3})}{h_{A1} - h_D}$$

$$= \frac{(3120 - 2790) + (2889 - 2535) + (2641 - 2300)}{3120 - 2140} = \frac{1025}{980}$$

$$= 1.046 \text{ Ans.}$$

4. *Overall thermal efficiency*

We know that overall thermal efficiency,

$$\eta_O = \frac{h_{A1} - h_{A3}}{h_{A1} - h_{fD}} = \frac{3120 - 2402}{3120 - 191.8} = 0.245 \text{ or } 24.5\% \text{ Ans.}$$

Example 25.4. *In a four stage impulse turbine, the steam is supplied at 350° C and at a pressure of 20 bar. The exhaust pressure is 0.05 bar and the overall turbine efficiency is 80%. Assuming that the work is shared equally between the stages and the condition line to be straight, determine (a) stage steam pressures ; (b) efficiency of each stage ; and (c) reheat factor.*

Solution. Given : $T_1 = 350°\text{C}$; $p_1 = 20$ bar ; $p_5 = 0.05$ bar ; $\eta_0 = 80\% = 0.8$

Stage steam pressures

Let p_2, p_3, p_4 = Steam pressures leaving the first, second and third stage respectively.

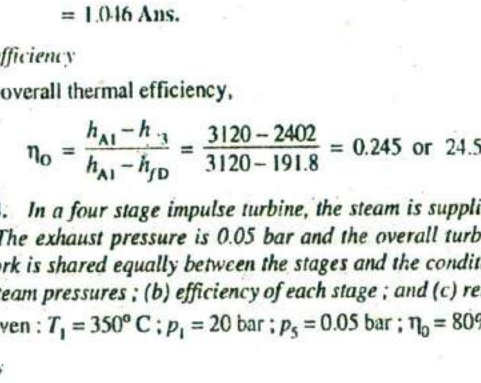


Fig. 25.7

Now let us draw the expansion of steam in four stages on the Mollier chart, as shown in Fig. 25.7, as discussed below :

1. First of all, locate the point A_1 at the intersection of initial pressure (20 bar) and initial temperature (350° C). We find that the enthalpy of steam at A_1, *i.e.* $h_{A1} = 3140$ kJ/kg.

2. From A_1, draw a vertical line A_1D meeting the pressure line of 0.05 bar at D representing the total isentropic expansion of the steam in all the four stages. We find that enthalpy of steam at D, *i.e.* $h_D = 2125$ kJ/kg. Thus we see that total enthalpy or heat drop,

$$h_{A1} - h_D = 3140 - 2125 = 1015 \text{ kJ/kg}$$

We know that overall thermal efficiency,

$$0.8 = \frac{h_{A1} - h_{C4}}{h_{A1} - h_D} = \frac{3140 - h_{C_4}}{1015}$$

$$\therefore \qquad h_{C4} = 2328 \text{ kJ/kg}$$

and total work done in all the four stages

$$= h_{A1} - h_{C4} = 3140 - 2328 = 812 \text{ kJ/kg}$$

$\therefore$ Work done in each stage $= 812 / 4 = 203$ kJ/kg

3. Now locate the point A_5 at the intersection of $h_{C4} = 2328$ kJ/kg and final pressure line of 0.05 bar. Now draw the condition line by joining A_1 and A_5 by a straight line (as given).

4. Now locate the point C_1 by cutting off A_1C_1 equal to 203 kJ/kg. Through C_1, draw a horizontal line meeting the condition line at A_2. From the Mollier diagram, we find that $p_2 = 9$ bar. **Ans.**

5. Similarly, locate the point C_2 by cutting off A_2C_2 equal to 203 kJ/kg. Through C_2, draw a horizontal line meeting the condition line at A_3. From the Mollier diagram, we find that $p_3 =$ 2.4 bar. **Ans.**

6. Again locate the point C_3 by cutting off A_3C_3 equal to 203 kJ/kg. Through C_3, draw a horizontal line meeting the condition line at A_4. From the Mollier diagram, we find that $p_4 =$ 0.5 bar. **Ans.**

Efficiency of each stage

From the Mollier diagram, we find that

$h_{A1} = 3140$ kJ/kg ; $h_{C1} = h_{A2} = 2937$ kJ/kg ; $h_{B1} = 2860$ kJ/kg ; $h_{C2} = h_{A3} = 2734$ kJ/kg ;

$h_{B2} = 2670$ kJ/kg ; $h_{C3} = h_{A4} = 2531$ kJ/kg ; $h_{B3} = 2475$ kJ/kg ; $h_{C4} = h_{A5} = 2328$ kJ/kg ;

$h_{B4} = 2285$ kJ/kg and $h_D = 2125$ kJ/kg

We know that efficiency of the first stage,

$$\eta_1 = \frac{A_1C_1}{A_1B_1} = \frac{h_{A1} - h_{C1}}{h_{A1} - h_{B1}} = \frac{3140 - 2937}{3140 - 2860} = 0.725 = 72.5\% \text{ Ans.}$$

Similarly,
$$\eta_2 = \frac{A_2C_2}{A_2B_2} = \frac{h_{A2} - h_{C2}}{h_{A2} - h_{B2}} = \frac{2937 - 2734}{2937 - 2670} = 0.76 = 76\% \text{ Ans.}$$

$$\eta_3 = \frac{A_3C_3}{A_3B_3} = \frac{h_{A3} - h_{C3}}{h_{A3} - h_{B3}} = \frac{2734 - 2531}{2734 - 2475} = 0.784 = 78.4\% \text{ Ans.}$$

and
$$\eta_4 = \frac{A_4C_4}{A_4B_4} = \frac{h_{A4} - h_{C4}}{h_{A4} - h_{B4}} = \frac{2531 - 2328}{2531 - 2285} = 0.825 = 82.5\% \text{ Ans.}$$

Reheat factor

We know that reheat factor,

$$\text{R.F.} = \frac{A_1B_1 + A_2B_2 + A_3B_3 + A_4B_4}{A_1D}$$

$$= \frac{(h_{A1} - h_{B1}) + (h_{A2} - h_{B2}) + (h_{A3} - h_{B3}) + (h_{A4} - h_{B5})}{h_{A1} - h_D}$$

$$= \frac{(3140 - 2860) + (2937 - 2670) + (2734 - 2475) + (2531 - 2285)}{3140 - 2125}$$

$$= \frac{1052}{1015} = 1.04 \textbf{ Ans.}$$

25.8. Regenerative Cycle

We have already discussed in chapter 10, Rankine and Carnot vapour cycles. The efficiency of Rankine cycle is less than that of Carnot cycle, because in the Rankine cycle, all the heat is not added at the higher temperature as is done in the case of Carnot cycle. Moreover, a large amount of additional heat is rejected to the condenser (shown by area *DEFG* in Fig. 25.8). This rejected heat is not compensated by the additional work represented by the area *ADE*. If, however, the working fluid enters the boiler, at some state, between *E* and *A*, the average temperature of heat supplied is increased. Thus the cycle becomes as efficient as Carnot cycle.

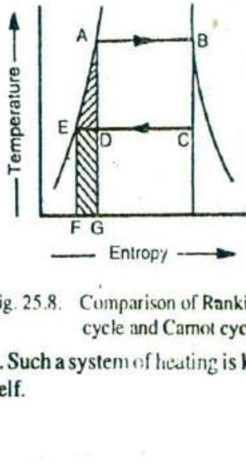


Fig. 25.8. Comparison of Rankine cycle and Carnot cycle.

This is achieved during an ideal regenerative cycle as shown in Fig. 25.9. The dry saturated steam, from the boiler, enters the turbine at a higher temperature, and then expands isentropically to a lower temperature in the same way as that in the Rankine and Carnot cycle. Now the condensate, from the condenser, is pumped back and circulated around the turbine casing, in the direction opposite to the steam flow in the turbine. The steam is thus heated before entering into the boiler. Such a system of heating is known as *regenerative heating*, as the steam is used to heat the steam itself.

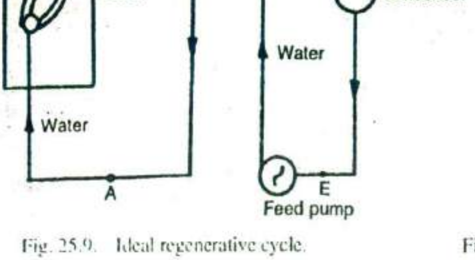


Fig. 25.9. Ideal regenerative cycle.

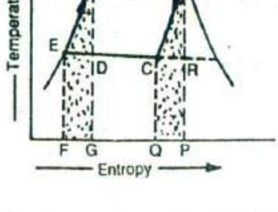


Fig. 25.10. Regenerative cycle on *T-s* diagram.

It may be noted that due to loss of heat, the expansion in the steam turbine is no more isentropic. It follows the path *BC*, which is exactly parallel to *EA*, as shown in Fig. 25.10.

A little consideration will show, that the heat transferred to the liquid (area *EAGF*) is equal to the heat transferred from the steam (area *BPQC*). Moreover, the heat is supplied to the working fluid at constant temperature in the process shown by curve *AB*. This is represented by the area *ABPG*. The heat is rejected from the working fluid at constant temperature shown by curve *CE*. This is represented by the area *CQFE* which is equal to the area *RPGD*. The area *RPGD* represents, to some scale, the heat rejected in the Carnot cycle. Thus, the ideal regenerative cycle has an efficiency equal to the efficiency of Carnot cycle with the same heat supply and heat rejection temperatures.

It will be interesting to know that the ideal regenerative cycle is only of academic interest as it is impossible to achieve the cycle, in actual practice, due to the following two reasons :

1. It is not possible to effect the necessary heat transfer from the steam in the turbine to the liquid feed water.
2. The moisture content of the steam leaving the turbine is considerably increased as a result of the heat transfer.

Note : Since it is impossible to achieve the ideal regenerative cycle, in practice, therefore some advantage of this principle is taken in the form of bleeding a part of the steam, which is discussed below :

25.9. Bleeding

The process of draining steam from the turbine, at certain points during its expansion and using this steam for heating the feed water (in feed water heaters) and then supplying it to the boiler is known as *bleeding*, and the corresponding steam is said to be bled. A feed water heater is a simple form of heat exchanger. It, usually, consists of tubes through which the feed water flows. These tubes are surrounded by a casing, containing the heating steam. The steam condenses and transfers its latent heat to the feed water in the tubes. There is no theoretical limit to the number of heaters, which will yield increased efficiency. But due to the cost of each heater and complicated and costly net work of pipe lines and fittings, its number is restricted to six in the larger modern turbo-generators. The effects of bleeding are :

1. It increases the thermodynamic efficiency of the turbine.
2. The boiler is supplied with a hot water.
3. A small amount of work is lost by the turbine, which decreases the power developed.

25.10. Regenerative Cycle with Single Feed Water Heater

Consider a regenerative cycle with single feed water heater as shown in Fig. 25.11.

The steam (at pressure p_1) enters the turbine at point *A*. Let a small amount of wet steam (say m kg) after partial expansion (at pressure p_2) be drained from the turbine at point *B* and enter the feed water heater. The remaining steam (at pressure p_3) is further expanded in the turbine and leaves at point *C* as shown on the Mollier diagram in Fig. 25.12 (Example 25.5).

This steam is then condensed in the condenser. The condensate from the condenser, is pumped into the feed water heater, where it mixes up with the steam extracted from the turbine. The proportion of steam extracted is just sufficient to cause the steam leaving the feed water heater to be saturated. Now consider 1 kg of steam entering the turbine at point *A*.

Let h_1 = Enthalpy or total heat of steam entering the turbine at *A*,

h_2 = Enthalpy or total heat of bled steam,

h_3 = Enthalpy or total heat of steam leaving the turbine at *C*,

h_{f2} = Enthalpy or sensible heat of feed water leaving the feed water heater,

h_{f3} = Enthalpy or sensible heat of steam leaving the condenser, and

m = Amount of bled steam per kg of steam supplied.

We know that heat lost by bled steam

$$= \text{Heat gained by feed water}$$

or

$$m(h_2 - h_{f2}) = (1-m)(h_{f2} - h_{f3})$$

$$mh_2 - m\,h_{f2} = h_{f2} - h_{f3} - m\,h_{f2} + m\,h_{f3}$$

$$\therefore \quad m = \frac{h_{f2} - h_{f3}}{h_2 - h_{f3}} \qquad \ldots(i)$$

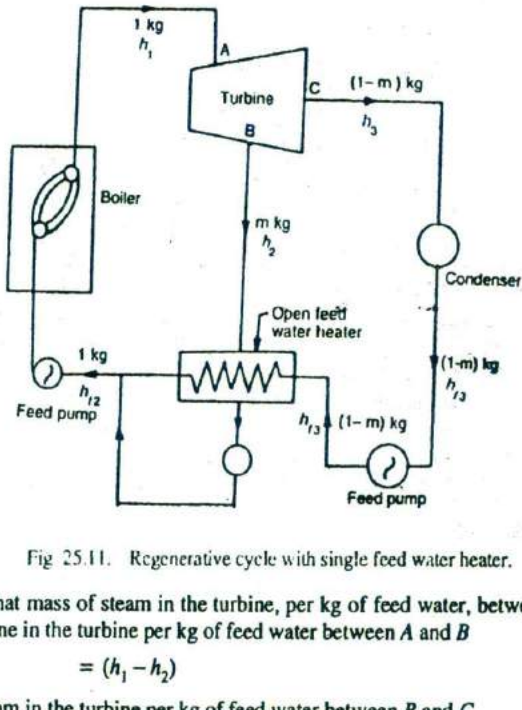


Fig 25.11. Regenerative cycle with single feed water heater.

We know that mass of steam in the turbine, per kg of feed water, between A and B is 1 kg. Therefore work done in the turbine per kg of feed water between A and B

$$= (h_1 - h_2)$$

Mass of steam in the turbine per kg of feed water between B and C

$$= (1-m)\text{ kg}$$

$\therefore$ Work done in the turbine between B and C

$$= (1-m)(h_2 - h_3)$$

Total work done $= (h_1 - h_2) + (1-m)(h_2 - h_3)$

and total heat supplied per kg of feed water

$$= h_1 - h_{f2}$$

$\therefore$ Efficiency of the cycle including the effect of bleeding,

$$\eta = \frac{\text{Total work done}}{\text{Total heat supplied}} = \frac{(h_1 - h_2) + (1-m)(h_2 - h_3)}{h_1 - h_{f2}}$$

Note : If there had been no regenerative feed heating (or in other words, $m = 1$), then the efficiency of the cycle will be the same, as that of Rankine cycle. In this case, Rankine efficiency,

$$\eta = \frac{h_1 - h_3}{h_1 - h_{f3}}$$

Example 25.5. *In a regenerative cycle, having one feed water heater, the dry saturated steam is supplied from the boiler at a pressure of 30 bar and the condenser pressure is 1 bar. The steam is bled at a pressure of 5 bar. Determine the amount of bled steam per kg of steam supplied and the efficiency of the cycle. What would be the efficiency without regenerative feed heating ? Also determine the percentage increase in efficiency due to regeneration.*

Solution. Given : $p_1 = 30$ bar ; $p_3 = 1$ bar ; $p_2 = 5$ bar

From Mollier diagram, as shown in Fig. 25.12, we find that

Enthalpy of steam at 30 bar, $h_1 = 2800$ kJ/kg

Enthalpy of steam at 5 bar, $h_2 = 2460$ kJ/kg

Enthalpy of steam at 1 bar, $h_3 = 2220$ kJ/kg

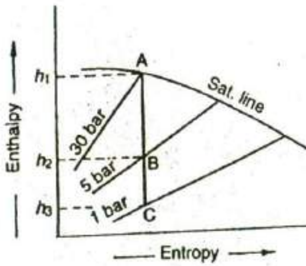


Fig. 25.12

From steam tables, we also find that enthalpy or sensible heat of water at 5 bar,

$$h_{f2} = 640.1 \text{ kJ/kg}$$

and enthalpy or sensible heat of water at 1 bar,

$$h_{f3} = 417.5 \text{ kJ/kg}$$

Amount of bled steam per kg of steam supplied

We know that amount of bled steam per kg of steam supplied,

$$m = \frac{h_{f2} - h_{f3}}{h_2 - h_{f3}} = \frac{640.1 - 417.5}{2460 - 417.5} = 0.109 \text{ kg } \textbf{Ans.}$$

Efficiency of the cycle

We know that efficiency of the cycle,

$$\eta_1 = \frac{(h_1 - h_2) + (1 - m)(h_2 - h_3)}{h_1 - h_{f2}}$$

$$= \frac{(2800 - 2460) + (1 - 0.109)(2460 - 2220)}{2800 - 640.1} = \frac{553.84}{2159.9}$$

$$= 0.256 \text{ or } 25.6\,\% \textbf{ Ans.}$$

Efficiency of the cycle without regenerative feed heating

We know that efficiency of the cycle,

$$\eta_2 = \frac{h_1 - h_3}{h_1 - h_{f3}} = \frac{2800 - 2220}{2800 - 417.5} = \frac{580}{2382.5}$$

$$= 0.243 \text{ or } 24.3\% \textbf{ Ans.}$$

Percentage increase in efficiency due to regeneration

We know that percentage increase in efficiency due to regeneration

$$= \frac{0.256 - 0.243}{0.243} = 0.0535 \text{ or } 5.35\% \textbf{ Ans.}$$

25.11. Regenerative Cycle with Two Feed Water Heaters

In this case, the steam is removed from the turbine at two points B and B_1. It is then fed into two open feed water heaters 1 and 2 as shown in Fig. 25.13.

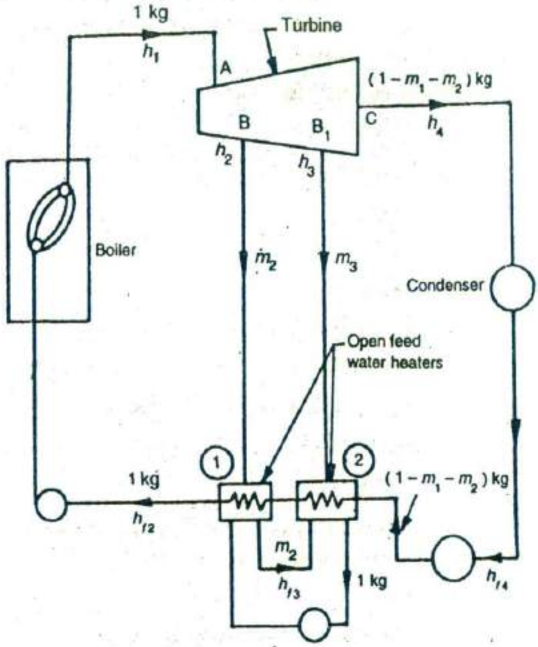


Fig. 25.13. Regenerative cycle with two feed water heaters.

The steam (at pressure p_1) enters the turbine at point A. Let a small amount of steam (say m kg) after partial expansion (at pressure p_2) be drained from the turbine at point B and enter the feed water heater 1. Similarly, let another small amount of steam (say m_2 kg) after further expansion (at pressure p_3) be drained from the turbine at point B_1 and enter the feed water heater 2. The remaining steam equal to $(1 - m_1 - m_2)$ kg (at pressure p_4) is further expanded in turbine, and leaves it at point C as shown on the Mollier diagram in Fig. 25.14.

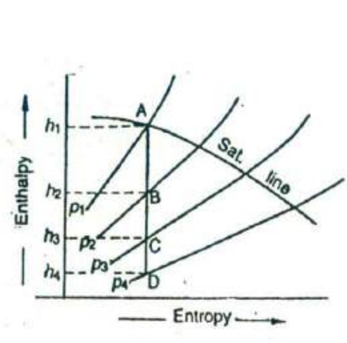


Fig. 25.14

The steam is then condensed in the condenser. The condensate from the condenser is pumped into the feed water heater, where it mixes up with the steam extracted from the turbine. Now consider 1 kg of steam entering into the turbine at A as shown in Fig. 25.13.

Let h_1 = Enthalpy of steam entering the turbine at A,

h_2 = Enthalpy of steam bled at B,

h_3 = Enthalpy of steam bled at B_1,

h_4 = Enthalpy of steam leaving the turbine at C,

h_{f2} = Enthalpy of feed water leaving the feed water heater 1,

h_{f3} = Enthalpy of feed water leaving the feed water heater 2,

h_{f4} = Enthalpy of steam leaving the condenser,

m_1 = Amount of steam bled at B per kg of steam supplied, and

m_2 = Amount of steam bled at B_1 per kg of steam supplied.

We know that heat lost by bled steam at B

= Heat gained by feed water

$$m_1 (h_2 - h_{f2}) = (1 - m_1)(h_{f2} - h_{f3})$$

$$m_1 h_2 - m_1 h_{f2} = h_{f2} - h_{f3} - m_1 h_{f2} + m_1 h_{f3}$$

$$\therefore \quad m_1 = \frac{h_{f2} - h_{f3}}{h_2 - h_{f3}}$$

Similarly, heat lost by bled steam at B_1

= Heat gained by feed water

$$\therefore \quad m_2 (h_3 - h_{f3}) = (1 - m_1 - m_2)(h_{f3} - h_{f4})$$

$$m_2 h_3 - m_2 h_{f3} = h_{f3} - h_{f4} - m_1 h_{f3} + m_1 h_{f4} - m_2 h_{f3} + m_2 h_{f4}$$

or

$$m_2 = \frac{(1 - m_1)(h_{f3} - h_{f4})}{h_3 - h_{f4}}$$

We know that the mass of steam in the turbine per kg of feed water between A and B is 1 kg.

$\therefore$ Work done in the turbine per kg of feed water between A and B

$$= h_1 - h_2 \quad \ldots (i)$$

and mass of steam in the turbine per kg of feed water between B and B_1

$$= (1 - m_1) \text{ kg}$$

$\therefore$ Work done in the turbine between B and B_1

$$= (1 - m_1)(h_2 - h_3) \quad \ldots (ii)$$

Similarly, mass of steam in the turbine per kg of feed water between B_1 and C

$$= (1 - m_1 - m_2) \text{ kg}$$

$\therefore$ Work done in the turbine between B_1 and C

$$= (1 - m_1 - m_2)(h_3 - h_4) \quad \ldots (iii)$$

Thus total work done per kg of feed water

$$= (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$$

and total heat supplied per kg of feed water

$$= h_1 - h_{f2}$$

$\therefore$ Efficiency of the plant including the effect of bleeding,

$$\eta = \frac{\text{Total work done}}{\text{Total heat supplied}}$$

Note : When the bleeding takes place at more than two points, the efficiency of the plant may be obtained by proceeding in the same way as explained above.

Example 25.6. *In a steam turbine plant, the steam is generated and supplied to the turbine at 50 bar and 370° C. The condenser pressure is 0.1 bar and the steam enters the condenser with dryness fraction of 0.9. Two feed heaters are used, the steam in the heaters being bled at 5 bar and 0.5 bar. In each heater, the feed water is heated to saturation temperature of the bled steam. The condensate is also pumped at this temperature into the feed line immediately after the heater. Find the masses of the steam bled in the turbine per one kg of steam entering the turbine. Assuming the condition line for the turbine to be straight, calculate the thermal efficiency of the cycle.*

Solution. Given : $p_1 = 50$ bar ; $T_1 = 370°$ C ; $p_4 = 0.1$ bar ; $x_4 = 0.9$; $p_2 = 5$ bar ; $p_3 = 0.5$ bar

First of all, let us draw the Mollier diagram and condition line for the cycle, as shown in Fig. 25.15. From this diagram, we find that

$$h_1 = 3110 \text{ kJ/kg} \; ; \; h_2 = 2780 \text{ kJ/kg} \; ;$$

$$h_3 = 2510 \text{ kJ/kg} \; ; \; h_4 = 2350 \text{ kJ/kg}$$

From steam tables, we also find that

$$h_{f2} = 640.1 \text{ kJ/kg (at 5 bar)}$$

$$h_{f3} = 340.6 \text{ kJ/kg (at 0.5 bar)}$$

$$h_{f4} = 191.8 \text{ kJ/kg (at 0.1 bar)}$$

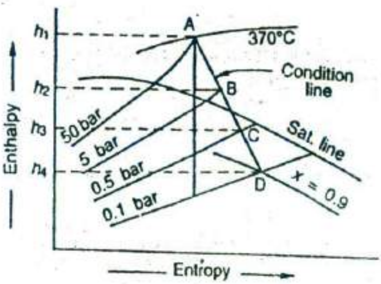


Fig. 25.15

Mass of steam bled in the turbine

We know that mass of steam bled at B,

$$m_1 = \frac{h_{f2} - h_{f3}}{h_2 - h_{f3}} = \frac{640.1 - 340.6}{2780 - 340.6} = 0.123 \text{ kg} \quad \textbf{Ans.}$$

and mass of steam bled at B_1,

$$m_2 = \frac{(1 - m_1)(h_{f3} - h_{f4})}{h_3 - h_{f4}} = \frac{(1 - 0.123)(340.6 - 191.8)}{2510 - 191.8}$$

$$= 0.056 \text{ kg} \quad \textbf{Ans.}$$

Thermal efficiency of the cycle

We know that work done from A to B per kg of feed water

$$= h_1 - h_2 = 3110 - 2780 = 330 \text{ kJ/kg} \qquad \ldots (i)$$

Similarly, work done from B to B_1 per kg of feed water

$$= (1 - m_1)(h_2 - h_3) = (1 - 0.123)(2780 - 2510) \text{ kJ/kg}$$

$$= 236.8 \text{ kJ/kg} \qquad \ldots (ii)$$

and work done from B_1 to C per kg of feed water

$$= (1 - m_1 - m_2)(h_3 - h_4) = (1 - 0.123 - 0.056)(2510 - 2350) \text{ kJ/kg}$$

$$= 131.4 \text{ kJ/kg} \qquad \ldots (iii)$$

$\therefore$ Total work done $= 330 + 236.8 + 131.4 = 698.2$ kJ/kg

Heat supplied $= h_1 - h_{f2} = 3110 - 640.1 = 2469.9$ kJ/kg

∴ Thermal efficiency of the cycle,

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{698.2}{2469.9} = 0.283 \quad \text{or} \quad 28.3\% \quad \textbf{Ans.}$$

25.12. Binary Vapour Plants

We have already discussed that the maximum possible efficiency of any steam engine (known as Carnot efficiency) is given by the equation :

$$\eta = \frac{T_1 - T_3}{T_1} = 1 - \frac{T_3}{T_1}$$

where T_1 is the higher temperature at which heat is absorbed and T_3 is the lower temperature at which the heat is rejected.

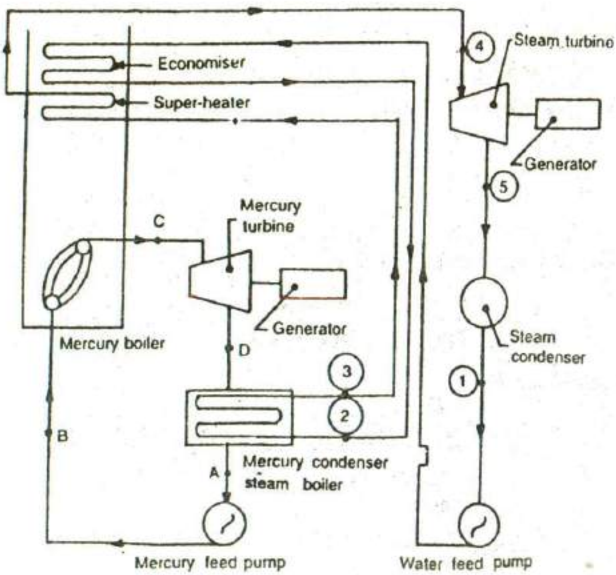


Fig. 25.16. Binary vapour plant.

We know that the value of lower temperature T_3 is fixed by atmospheric conditions. Thus the thermal efficiency of a steam plant can only be improved by increasing the value of T_1. In a steam plant, if T_1 is increased, it will also increase corresponding pressure, which is one of the limiting factors in its design. Thus a substance, other than steam, is used in the high *temperature range. By using mercury vapour, instead of steam, in the high temperature range of the cycle, an increased value of T_1 is obtained without any increase in maximum pressure. The heat rejected by the mercury, in condensing, is utilised in raising superheated steam for the lower temperature range of the cycle. A power plant using binary vapour (mercury and steam) is known as *binary vapour plant*.

* The critical temperature of mercury vapour is 588.4° C at a critical pressure of 21 bar. Moreover, the critical temperature of water vapour is 374.15° C at a critical pressure of 225.65 bar.

A diagrammatic view of a binary vapour plant is shown in Fig. 25.16. The liquid mercury passes from the mercury liquid heater to the mercury vapour boiler, where it is evaporated. It then flows to the mercury turbine through which it is expanded to its low pressure limit. It, now, exhausts to the mercury condenser steam boiler, where its latent heat is given out to the hot feed water. This operation condenses the mercury, while the feed water is evaporated into steam. The mercury is then returned to the mercury liquid heater, thus completing its cycle.

The feed water from the economiser is evaporated into steam in the mercury condenser steam boiler. It then passes to the superheater, where it is superheated by the hot flue gases. Now the superheated steam passes to the steam turbine in which it expands to the condenser pressure. It now passes to the economiser, thus completing the cycle of water and steam.

25.13. Binary Vapour Cycle

The binary vapour cycle on a *T-s* diagram is shown in Fig. 25.17. The line *AB* represents the evaporation of liquid mercury plotted to the same temperature scale as that of steam. But the scale for the corresponding pressures for the mercury is lower. The mercury vapour at *B* has a much higher temperature than the steam at the same pressure. The mercury vapours are now expanded isentropically in a mercury turbine as represented by the line *BC* in Fig. 25.17. The condensation of mercury is shown by the line *CD*. During condensation, the latent heat is utilized for evaporating a corresponding amount of steam. The line *DA* represents the heating of mercury. Thus the mercury has completed a cycle *ABCD*.

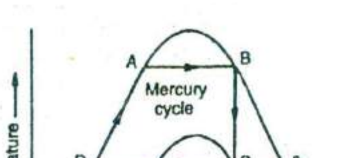


Fig. 25.17. Binary vapour cycle on *T-s* diagram.

The steam cycle is represented by 1-2-3-4-5 as shown in Fig. 25.17. The line 1-2 represents the evaporation by the condensing mercury. The line 2-3 represents the superheating of the steam by the flue gases. The steam is now expanded isentropically through a steam turbine as shown by the line 3-4. The condensation of the exhaust steam is represented by the line 4-5. The heating of feed water is represented by the line 5-1. This completes the steam cycle.

Notes : **1.** It is found that 8.196 kg of mercury is required per kg of steam to obtain the correct amount of heat from the condensing mercury for evaporating the steam. It may be seen from the areas of *T-s* diagram that the use of mercury for the high temperature range of the cycle gives a higher efficiency than that obtained from steam with the same addition of heat.

2. The relation between the two cycles is that the heat lost by mercury in *CD* is equal to the heat gained by steam in 1-2. Mathematically, heat lost by mercury at *C* is equal to heat gained by steam at 1.

25.14. Some Special Turbines

The following are some special turbines commonly used in industry :

1. Pass-out or extraction turbine ;
2. Back pressure turbine ; and
3. Exhaust or low pressure turbine.

The function of these turbines, aside from power generation, is to supply steam for manufacturing processes or heating or to utilise steam that would otherwise be wasted.

The above mentioned turbines are discussed, in detail, in the following pages.

25.15. Pass-out or Extraction Turbine

A pass-out or extraction turbine is of the type used in central stations, in which steam is extracted at different stages and used in heating the feed water for the boiler or processing work such as paper making, textile, dying, sugar refining etc. Thus, a pass-out or extraction turbine supplies the required power and also the low pressure steam needed for the purpose.

The typical arrangement of a pass-out or extraction turbine is shown in Fig. 25.18. The high pressure steam from the boiler enters the H.P. stage of the turbine where it expands and the pressure is reduced to such a value as required for processing work. A part of this low pressure steam leaving the high pressure stage is supplied to the processing work while the remaining steam expands further in the L.P. stage. The exhaust steam from the processing plant and the low pressure turbine is condensed in a condenser and pumped back to the boiler.

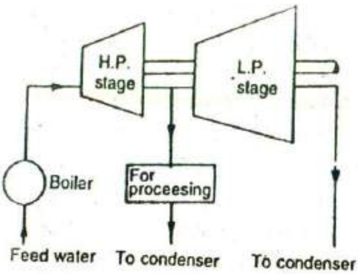


Fig. 25.18. Pass-out or extraction turbine.

25.16. Back Pressure Turbine

The back pressure turbine is also used in applications where combined power and heat in steam for process work is required. The typical arrangement of such a turbine is shown in Fig. 25.19.

In this turbine, the steam leaves the turbine at a higher pressure than in normal turbine and is generally superheated. The superheated steam is not suitable for process work due to the following reasons :

1. The control of its temperature is impossible, and
2. The rate of heat transfer from the superheated steam to the heating surface is lower than that of saturated steam.

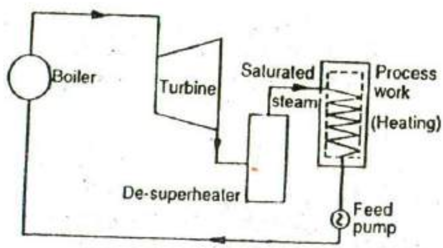


Fig. 25.19. Back pressure turbine.

Thus, the exhaust is passed through a de-superheater to make the steam saturated. The saturated steam is then passed through a heater where it is fully condensed.

25.17. Exhaust or Low Pressure Turbine

The exhaust or low pressure turbine is chiefly used where there are several reciprocating steam engines which work intermittently and are non-condensing such as rolling mill and colliery engines. The arrangement of such a turbine is shown in Fig. 25.20.

In this turbine, the exhaust steam from the engine is expanded in a exhaust or low pressure turbine and then condensed in a condenser. In this turbine, some form of heat accumulator is required to collect the more

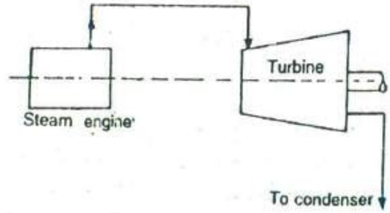


Fig. 25.20. Exhaust or low pressure turbine.

or less irregular supply of low pressure steam for the non-condensing steam engines and deliver it to the turbine at the required rate.

25.18. Future Power Plants

The depleting resources of oil, gas and coal (the conventional fuels used in power plants) along with atmospheric pollution problems have drawn the attention of the scientists and engineers all over the world to find out other sources for the generation of electric power. These sources of energy are going to attain the nerve centre of the future power plants. Though atomic and nuclear power plants have been developed on conventional lines, but lot of work is yet to be done. Efforts are being made to convert atomic and nuclear energy directly into electric power with the help of magneto-hydrodynamic generator and other equipments.

At certain places research is being conducted to use solar energy and gases for the generation of power. It is hoped that our scientists will be able to save the humanity from the power crisis in the near future.

EXERCISES

1. In a power plant, steam at a pressure of 31.5 bar and temperature 370° C is supplied by a boiler. The steam is removed after expansion to 5.6 bar in high pressure turbine, reheated at 370° C and is used in the low pressure turbine. Exhaust is at 0.035 bar. Calculate the gain in efficiency by reheating. **[Ans.** 1.78%]

2. Steam at a pressure of 150 bar and 550° C is expanded through a turbine to a pressure of 40 bar. It is then reheated to a temperature of 550° C after which it completes its expansion through the turbine to an exhaust pressure of 0.1 bar. Calculate the ideal efficiency of the plant and the work done,

   (*a*) taking the reheating into account, and

   (*b*) if the steam was expanded direct to the exhaust pressure without reheating.

   [**Ans.** (*a*) 43.75%, 1640 kJ/kg ; (*b*) 41.8%, 1370 kJ/kg]

3. Steam supplied to a three stage turbine is at 20 bar and 350° C. The steam leaves the first stage at 6 bar and the second stage at 1 bar. The steam finally leaves at 0.1 bar. Each stage has an efficiency of 0.8. Find : (*i*) Rankine efficiency, (*ii*) Final condition of steam, (*iii*) Overall thermal efficiency, (*iv*) Efficiency ratio, and (*v*) Reheat factor. [Ans. 35.2% ; 0.89 ; 28.3% ; 0.804 ; 1.008]

4. A three stage pressure compounded impulse turbine operates under initial steam conditions of 28 bar and 350° C and condenser pressure of 50 mm of Hg. Assuming an internal efficiency of 0.78 and equal work is done in all the stages, estimate the probable stage efficiencies and the reheat factor of the turbine.

   [Ans. 71.7%, 74.7%, 78.1% ; 1.043]

5. Steam is supplied at 17.5 bar and 300° C to a 5-stage turbine and exhausts at 0.04 bar. The enthalpies after actual expansion and isentropic expansion in kJ/kg are as follows :

Stage No.	1	2	3	4	5
Actual expansion	2901	2746	2592	2416	2232
Isentropic expansion	2842	2700	2550	2374	2207

If the reheat factor for the turbine is 1.04, determine the stage efficiencies and the overall efficiency.

[Ans. 68% ; 77% ; 78.6 %, 80.7%, 88% ; 89.2 %]

6. A steam power plant equipped with regenerative as well as reheat arrangement is supplied with steam to the high pressure turbine at 80 bar and 470° C. A part of the steam is extracted at 7 bar for feed heating and the remaining steam is reheated to 350° C in a reheater and expanded in low pressure turbine to a pressure of 0.035 bar. Determine : 1. The amount of steam bled for feed heating ; 2. The heat supplied in boiler and reheater ; 3. Output of the turbine ; and 4. Overall thermal efficiency.

   [**Ans.** 0.224 kg/kg of steam ; 2956.6 kJ/kg ; 1303.9 kJ/kg 44.1%]

7. A turbine receives steam at 42 bar and 371°C and exhausts at 0.106 bar. At the actual state of 10.5 bar and 204° C, the steam is withdrawn and part is used for feed water heating, while the remainder passes through a reheater. The reheated steam re-enters the turbine at 9.8 bar and 371° C. A second extraction for feed water

heating occurs at 1.4 bar and 177° C. A third extraction occurs at 0.35 bar and 93° C. The actual exhaust is dry and saturated steam. The actual temperatures of the feed water leaving the heaters are 71° C, 107° C and 167°C respectively. Assuming no losses in and between the heaters, find the actual percentage extracted at each point and the thermal efficiency. Represent the processes on *T-s* and *h-s* plane.

[**Ans.** 0.105 kg, 0.0533 kg, 0.0347 kg ; 30%]

8. A steam power plant using regenerative feed heating generates 27 MW. The steam at 60 bar and 450° C is supplied to the steam turbine. The condenser pressure is 0.07 bar. The steam is bled from the steam turbine at 3 bar. The heating of feed water is done in the direct contact heater and the condensate temperature is raised to 110° C.

The pumps absorb 9 percent of the alternator output, the boiler efficiency is 87 percent, the efficiency ratio of each section of turbine is 85 percent and the alternator efficiency is 97 percent. Ignoring all other losses, calculate : 1. the mass tapped to the feed heater per kg of steam to the condenser ; 2. the steam to be generated per hour, and 3. the overall thermal efficiency of the plant, neglecting the boiler feed pump work in calculating input to the boiler. [**Ans.** 0.13 kg ; 100 630 kg/h ; 26.94%]

QUESTIONS

1. Explain the process and purpose of reheating steam in steam turbine application.
2. What is reheat factor ? Explain it with the *h-s* diagram.
3. Describe regenerative feed heating as used in thermal power plants. List its advantages.
4. Explain the purpose of bleeding steam turbines, in detail.
5. Describe, with the help of diagram the binary vapour cycle of a thermal power plant ? What are its advantages.
6. Explain with suitable schematic diagram, the following :
   (*a*) Pass out turbine ; (*b*) Back pressure turbine ; and (*c*) Exhaust turbine.

OBJECTIVE TYPE QUESTIONS

1. The efficiency of steam turbines may be improved by
   (*a*) reheating of steam (*b*) regenerative feed heating
   (*c*) binary vapour plant (*d*) any one of these
2. The reheating of steam in a turbine
   (*a*) increases the workdone through the turbine
   (*b*) increases the efficiency of the turbine
   (*c*) reduces wear on the blades
   (*d*) all of the above
3. The ratio of isentropic heat drop to the heat supplied is called
   (*a*) Rankine efficiency (*b*) stage efficiency
   (*c*) reheat factor (*d*) internal efficiency
4. The ratio of total useful heat drop to the total isentropic heat drop is called
   (*a*) internal efficiency (*b*) efficiency ratio
   (*c*) Rankine efficiency (*d*) stage efficiency
   (*e*) both (*a*) and (*b*) (*f*) both (*c*) and (*d*)
5. The reheat factor is the ratio of
   (*a*) cumulative heat drop to the isentropic heat drop
   (*b*) isentropic heat drop to the heat supplied
   (*c*) total useful heat drop to the total isentropic heat drop
   (*d*) none of the above

6. The reheat factor depends upon
   (*a*) initial pressure and superheat
   (*b*) exit pressure
   (*c*) turbine stage efficiency
   (*d*) all of these
7. The value of reheat factor varies from
   (*a*) 1.02 to 1.06 (*b*) 1.08 to 1.10 (*c*) 1.2 to 1.6 (*d*) 1.6 to 2.0

8. The process of draining steam from the turbine, at certain points during its expansion and using this steam for heating the feed water in feed water heaters and then supplying it to the boiler, is known as
   (*a*) regenerative feed heating
   (*b*) reheating of steam
   (*c*) bleeding
   (*d*) none of these
9. The effect of bleeding is that
   (*a*) it increases the thermodynamic efficiency of the turbine
   (*b*) boiler is supplied with hot water
   (*c*) it decreases the power developed by the turbine
   (*d*) all of the above
10. A binary vapour plant consists of
   (*a*) steam turbine
   (*b*) steam condenser
   (*c*) mercury boiler
   (*d*) economiser
   (*e*) superheater
   (*f*) all of these

ANSWERS

1. (*d*)	2. (*d*)	3. (*a*)	4. (*e*)	5. (*a*)
6. (*d*)	7. (*a*)	8. (*c*)	9. (*d*)	10. (*f*)

26

Internal Combustion Engines

1. Introduction. 2. Comparison of Steam Engines and Internal Combustion Engines. 3. Classification of I.C. Engines. 4. Main Components of I.C. Engines. 5. Sequence of Operations in a Cycle. 6. Two-stroke and Four-stroke Cycle Engines. 7. Advantages and Disadvantages of Two-stroke over Four-stroke Cycle Engines. 8. Valve Timing Diagrams. 9. Four-stroke Cycle Petrol Engine. 10. Actual Indicator Diagram for a Four-stroke Cycle Petrol Engine. 11. Valve Timing Diagram for a Four-stroke Cycle Petrol Engine. 12. Four-stroke Cycle Diesel Engine. 13. Actual Indicator Diagram for a Four-stroke Cycle Diesel Engine. 14. Valve Timing Diagram for a Four-stroke Cycle Diesel Engine. 15. Four-stroke Cycle Gas Engines. 16. Comparison of Petrol and Diesel Engines. 17. Two-stroke Cycle Petrol Engine. 18. Actual Indicator Diagram for a Two-stroke Cycle Petrol Engine. 19. Valve Timing Diagram for a Two-stroke Cycle Petrol Engine. 20. Two-stroke Cycle Diesel Engine. 21. Actual Indicator Diagram for a Two-stroke Cycle Diesel Engine. 22. Valve Timing Diagram for a Two-stroke Cycle Diesel Engine. 23. Scavenging. 24. Types of Scavenging. 25. Detonation in I.C. Engines. 26. Rating of S.I. Engine Fuels-Octane Number. 27. Rating of C.I. Engine Fuels-Cetane Number. 28. Ignition Systems for Petrol Engines. 29. Coil Ignition System. 30. Magneto Ignition System. 31. Fuel Injection System for Diesel Engines. 32. Cooling of I.C. Engines. 33. Cooling Systems for I.C. Engines. 34. Comparison of Air Cooling and Water Cooling Systems. 35. Supercharging of I.C. Engines. 36. Methods of Supercharging. 37. Lubrication of I.C. Engines. 38. Lubrication Systems for I.C. Engines. 39. Governing of I.C. Engines. 40. Methods of Governing I.C. Engines. 41. Carburettor. 42. Spark Plug. 43. Fuel Pump. 44. Injector or Atomiser.

26.1. Introduction

As the name implies or suggests, the internal combustion engines (briefly written as I.C. 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.

26.2. Comparison of Steam Engines and Internal Combustion Engines

Following points are important for the comparison of steam engines and internal combustion engines.

S. No	Steam Engines	I.C. Engines
1.	The combustion of fuel takes place outside the engine cylinder (*i.e.* in a boiler)	The combustion of fuel takes place inside the engine cylinder.
2.	Since combustion of fuel takes place outside the engine cylinder, therefore these engines are smooth and silent running.	Since combustion of fuel takes place inside the engine cylinder, these engines are very noisy.
3.	The working pressure and temperature inside the engine cylinder is low.	The working pressure and temperature inside the cylinder is very high.
4.	Because of low pressure and temperature, ordinary alloys are used for the manufacture of engine cylinder and its parts.	Because of very high pressure and temperature, special alloys are used for the manufacture of engine cylinder and its parts.
5.	A steam engine requires a boiler and other components to transfer energy. Thus, it is heavy and cumbersome.	An I.C. engine does not require a boiler or other components. Thus, it is light and compact.
6.	The steam engines have efficiency about 15-20 percent.	The I.C. engines have efficiency about 35-40 percent.
7.	It can not be started instantaneously.	It can be started instantaneously.

26.3. Classification of I.C. 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.I. 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, and (*c*) High speed engines.

6. *According to the cooling system*

   (*a*) Air-cooled engines, (*b*) Water-cooled engines, and (*c*) Evaporative cooling engines.

7. *According to the method of fuel injection*

   (*a*) Carburettor engines, (*b*) Air injection engines, and (*c*) Airless or solid injection engines.

8. *According to the number of cylinders*

   (*a*) Single cylinder engines, and (*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, (*f*) Opposite-cylinder engines, and (*g*) Opposite-piston engines.

10. *According to the valve mechanism*

    (*a*) Overhead valve engines, and (*b*) Side valve engines.

11. *According to the method of governing*

    (*a*) Hit and miss governed engines, (*b*) Quantitatively governed engines, and (*c*) Qualitatively governed engines.

26.4 Main Components of I.C. Engines

As a matter of fact, an I.C. 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 in Fig. 26.1 and are discussed below :

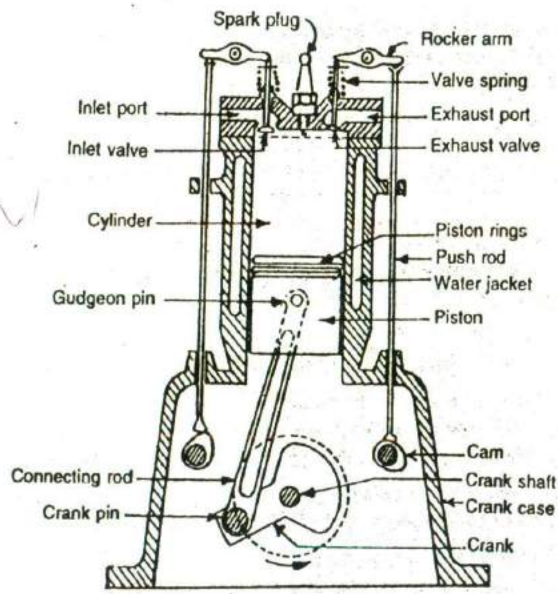


Fig. 26.1. Main components of I.C. engines.

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 I.C. engine, whose main function is to transmit the force exerted 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 I.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.

26.5. 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.

1. *Suction stroke.* In this stroke, the fuel vapour in correct proportion, is supplied to the engine cylinder.

2. *Compression stroke.* In this stroke, the fuel vapour is compressed in the engine cylinder.

3. *Expansion or working stroke.* In this stroke, the fuel vapour 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 vapour.

Note : The above mentioned strokes are meant for gas and petrol engines. But in case of diesel engines, pure air is sucked in suction stroke which is compressed during the compression stroke. The diesel oil is admitted into the engine cylinder (just before the beginning of the expansion stroke) and it is ignited by the hot air present in the cylinder. The expansion and exhaust strokes are similar to the gas and petrol engines.

26.6. Two-stroke and Four-stroke Cycle Engines

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.

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

Following are the advantages and disadvantages of two-stroke cycle engines 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 the theoretical 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. Inspite 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 are 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 creates noise, because of short time available for their exhaust.

26.8. Valve Timing Diagrams

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.

1. *Theoretical valve timing diagram for four stroke cycle engine.* The theoretical valve timing diagram for a four-stroke cycle engine is shown in Fig. 26.2. In this diagram, the inlet valve opens at *A* and the suction takes place from *A* to *B*. The crankshaft revolves through 180° and the piston moves from *T.D.C.* to *B.D.C.* At *B*, the inlet valve closes and the compression takes place from *B* to *C*. The crankshaft revolves through 180° and the piston moves from *B.D.C.* to *T.D.C.* At *C*, the fuel is fired and the expansion takes place from *C* to *D*. The crankshaft revolves through 180° 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° and the piston moves back to *T.D.C.*

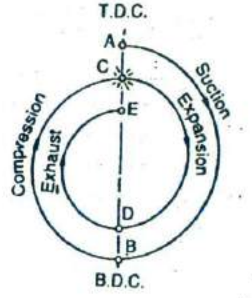


Fig. 26.2. Four-stroke cycle engine.

Note : In four-stroke cycle, the crank revolves through two revolutions.

2. *Theoretical valve timing diagram for two-stroke cycle engine.* The theoretical valve timing diagram for a two-stroke cycle engine is shown in Fig. 26.3. 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° and the piston moves from *T.D.C.* towards *B.D.C.* At *B*, both the valves open and suction as well as exhaust take place from *B* to *C*. The crankshaft revolves through approximately 120° and the piston moves first to *BDC* and then little upwards. At *C*, both the valves close and compression takes place from *C* to *A*. The crankshaft revolves through approximately 120° and the piston moves to *T.D.C.*

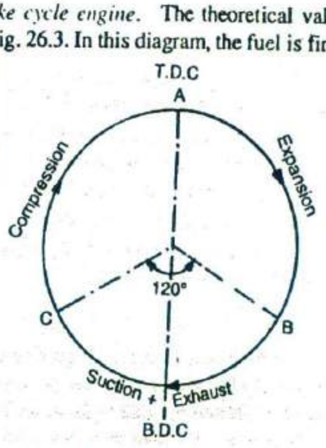


Fig. 26.3. Two-stroke cycle engine.

Notes : 1. In a two stroke cycle, the crank revolves through one revolution.

2. The readers will find valve timing diagram, drawn in other books having slight difference in the angles. This is due to the different engine speed and the manufacturer's design. However, in this book, the authors have given the diagrams which are widely used.

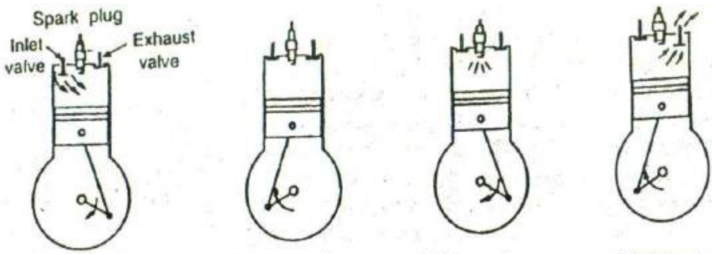
26.9. Four-stroke Cycle Petrol Engine

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 carburettor 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 Fig. 26.4 (*a*).

* In 1978, Doctor Otto, a German engineer devised an engine working on this cycle. The determination of air standard efficiency of Otto cycle has been discussed in Chapter 6.

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 *T.D.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 crankshaft. The compression stroke is shown in Fig. 26.4 (*b*).



(*a*) Suction or charging stroke. (*b*) Compression stroke. (*c*) Expansion or working stroke. (*d*) Exhaust stroke.

Fig. 26.4. Four-stroke cycle 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 Fig. 26.4 (*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 Fig. 26.4 (*d*). This completes the cycle, and the engine cylinder is ready to suck the charge again.

Note : The four stroke cycle petrol engine are usually employed in light vehicles such as cars, jeeps and aeroplanes.

26.10. Actual Indicator Diagram for a Four-stroke Cycle Petrol Engine

The actual indicator diagram for a four stroke cycle petrol engine is shown in Fig. 26.5. 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 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 valve closes (*IVC*) 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 (*IGN*) with the help of spark plug as shown in the figure. The sparking suddenly increases pressure and temperature of the

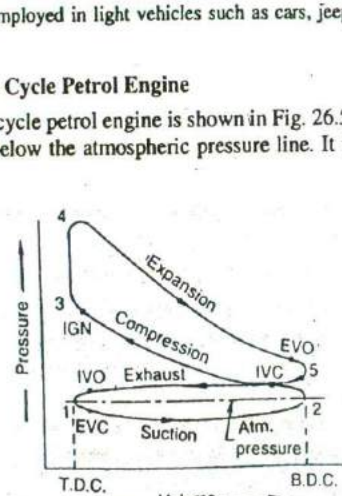
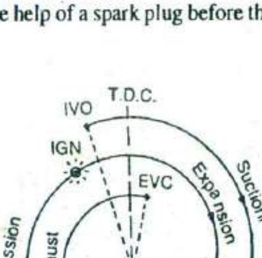


Fig. 26.5. Actual indicator diagram for a four stroke cycle petrol engine.

products of combustion. But 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 can not 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.

26.11. Valve Timing Diagram for a Four-stroke Cycle Petrol Engine

In the valve timing diagram, as shown in Fig. 26.6, 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 ignited with the help of a spark plug before the end of compression stroke. This is done as the charge



TDC : Top dead centre

BDC : Bottom dead centre

IVO : Inlet valve opens (10^o-20^o before *TDC*)

IVC : Inlet valve closes (30^o- 40^o after *BDC*)

IGN : Ignition (20^o-30^o before *TDC*)

EVO : Exit valve opens (30^o-50^o before *BDC*)

EVC : Exit valve closes (10^o-15^o after *TDC*)

Fig. 26.6. Valve timing diagram for a four stroke cycle petrol engine.

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 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.

26.12. Four-stroke Cycle Diesel Engine

It is also known as *compression ignition engine* because the ignition takes place 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 in Fig. 26.7 (*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 Fig. 26.7 (*b*).

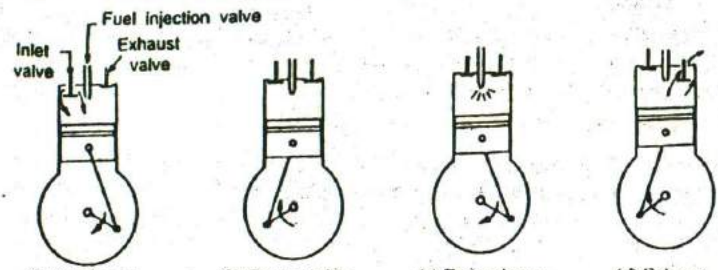


Fig. 26.7. Four stroke cycle diesel engine.

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.

Note : The four stroke cycle diesel engines are generally employed in heavy vehicles such as buses, trucks, tractors, pumping sets, diesel locomotives and in earth moving machinery.

26.13. Actual Indicator Diagram for a Four Stroke Cycle Diesel Engine

The actual indicator diagram for a four-stroke cycle diesel engine is shown in Fig. 26.8. 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 (*IVC*) 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 by high temperature of the compressed air. The ignition

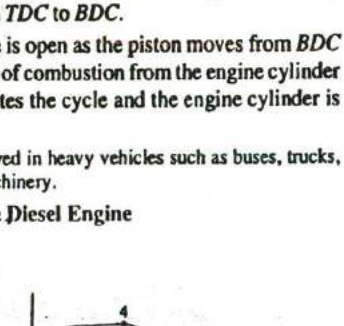
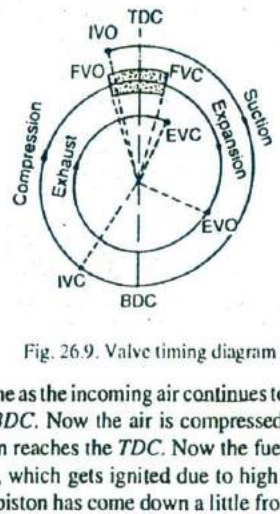


Fig. 26.8. Actual indicator diagram for a four stroke cycle diesel engine.

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 can not 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.

26.14. Valve Timing Diagram for a Four-stroke Cycle Diesel Engine

In the valve timing diagram as shown in Fig. 26.9, 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



TDC : Top dead centre

BDC : Bottom dead centre

IVO : Inlet valve opens (10° - 20° before *TDC*)

IVC : Inlet valve closes (25° - 40° after *BDC*)

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

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

EVO : Exhaust valve opens (39° - 50° before *BDC*)

EVC : Exhaust valve closes (10° - 15° after *TDC*)

Fig. 26.9. Valve timing diagram for a four stroke cycle diesel engine.

BDC. 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.

26.15. Four-stroke Cycle Gas Engines

A four stroke cycle gas engine, as the name indicates, uses natural or manufactured gas as the working fuel and works on Otto cycle. All the mechanical features of a gas engine are the same as those of a petrol engine. The ignition system, which usually consists of spark plug, is the same in both the gas and petrol engines. The working of a gas engine on *p-v* diagram is exactly similar to that of the petrol engine. The valve timing diagram for a four stroke cycle engine is also exactly similar to that of a petrol engine.

The only difference between a gas engine and a petrol engine is fuel and the way it is supplied. In a petrol engine, a mixture of petrol and air is sucked inside the engine cylinder during the suction stroke. But in a gas engine, gas and air are supplied to the engine. In most of the cases, the density and calorific value of gas is considerably less than that of the petrol. As a result of this, the cylinder of a gas engine is made larger than that of the petrol engine.

26.16. Comparison of Petrol and Diesel Engines

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

S. No	Petrol Engines	*Diesel Engines*
1.	A petrol engine draws a mixture of petrol and air during suction stroke.	A diesel engine draws only air during suction stroke.
2.	The carburettor is employed to mix air and petrol in the required proportion and to supply it to the engine during suction stroke.	The injector or atomiser is employed to inject the fuel at the end of compression stroke.
3.	Pressure at the end of compression is about 10 bar.	Pressure at the end of compression is about 35 bar.
4.	The charge (*i.e.* petrol and air mixture) is ignited with the help of spark plug.	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.
5.	The combustion of fuel takes place approximately at constant volume. In other words, it works on Otto cycle.	The combustion of fuel takes place approximately at constant pressure. In other words, it works on Diesel cycle.
6.	A petrol engine has compression ratio approximately from 6 to 10.	A diesel engine has compression ratio approximately from 15 to 25.
7.	The starting is easy due to low compression ratio.	The starting is little difficult due to high compression ratio.
8.	As the compression ratio is low, the petrol engines are lighter and cheaper.	As the compression ratio is high, the diesel engines are heavier and costlier.
9.	The running cost of a petrol engine is high because of the higher cost of petrol.	The running cost of diesel engine is low because of the lower cost of diesel.
10.	The maintenance cost is less.	The maintenance cost is more.
11.	The thermal efficiency is upto about 26%.	The thermal efficiency is upto about 40%.
12.	Oveheating trouble is more due to low thermal efficiency.	Oveheating trouble is less due to high thermal efficiency.
13.	These are high speed engines.	These are relatively low speed engines.
14.	The petrol engines are generally employed in light duty vehicles such as scooters, motorcycles, cars. These are also used in aeroplanes.	The diesel engines are generally employed in heavy duty vehicles like buses, trucks, and earth moving machines etc.

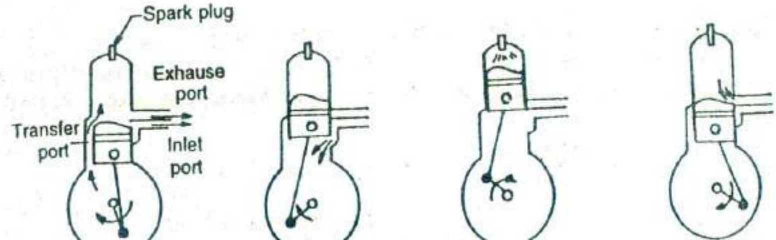
26.17. Two-stroke Cycle Petrol Engine

A two-stroke cycle-petrol engine was devised by Duglad Clerk in 1880. 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 in Fig. 26.10 (*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 in Fig. 26.10 (*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 Fig. 26.10 (*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.



(*a*) Suction. (*b*) Compression. (*c*) Expansion. (*d*) Exhaust.

Fig. 26.10. Two-stroke cycle 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 in Fig. 26.10 (*d*). This completes the cycle and the engine cylinder is ready to suck the charge again.

Note : The two stroke petrol engines are generally employed in very light vehicles such as scooters, motor cycles, three wheelers and sprayers.

26.18. Actual Indicator Diagram for a Two Stroke Cycle Petrol Engine

The actual indicator diagram for a two-stroke cycle petrol engine is shown in Fig 26.11. The 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 1 to 2 (*i.e. BDC*). In the second half of the 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. 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 as shown in the figure. 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

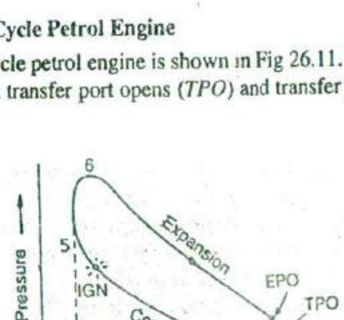


Fig. 26.11. Actual indicator diagram for a two stroke cycle petrol engine.

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.

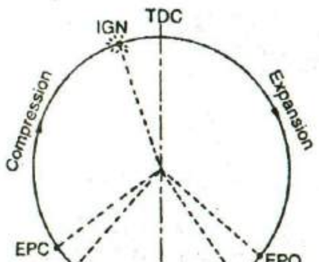
Notes : 1. The piston top is shaped in such a way that the fresh charge will move up towards spark plug, and it will also push out the remaining burnt gases through the exhaust port.

2. As soon as the exhaust port is uncovered by the piston (at this moment the transfer port is still covered), the burnt gases will be exhausted with a great force till the pressure in the cylinder is reduced, approximately to that of atmosphere.

3. Due to downward movement of the piston, the charge below the piston will be compressed. As the transfer port is uncovered, it will flow into cylinder due to pressure difference between the spaces below and above the piston.

26.19. Valve Timing Diagram for a Two-stroke Cycle Petrol Engine

In the valve timing diagram, as shown in Fig. 26.12, 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



TDC : Top dead centre

BDC : Bottom dead centre

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 opens (35° - 50° after *BDC*)

IGN : Ignition (15° - 20° before *TDC*)

Fig. 26.12. Valve timing diagram for a two-stroke cycle 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.

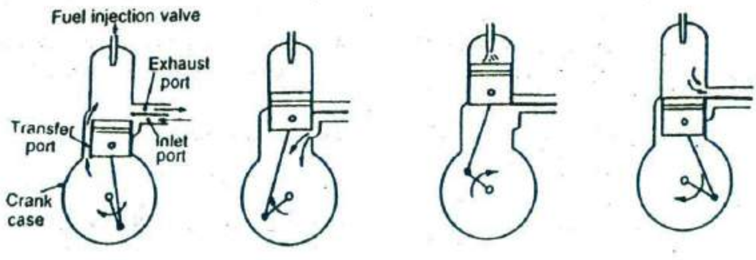
26.20. Two-stroke Cycle Diesel Engine

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 the transfer port and the exhaust port. The fresh air flows into the engine cylinder from the crank case, as shown in Fig. 26.13 (*a*).

2. *Compression stage.* In this stage, the piston while moving up, first covers the transfer port and then exhaust port. After that the air is compressed as the piston moves upwards as shown in Fig. 26.13 (*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 fig. 26.13 (*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.



(*a*) Suction. (*b*) Compression. (*c*) Expansion. (*d*) Exhaust.

Fig. 26.13. Two-stroke cycle 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 Fig. 26.13 (*d*). This completes the cycle, and the engine cylinder is ready to suck the air again.

Note : The two stroke diesel engines are mainly used in marine propulsion where space and lightness are the main considerations.

26.21. Actual Indicator Diagram for a Two Stroke Cycle Diesel Engine

The actual indicator diagram for a two-stroke cycle diesel engine is shown in Fig. 26.14. The 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 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. 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 expansion is shown by the line 6-7, Now the exhaust port opens (*EPO*) at 7 and the

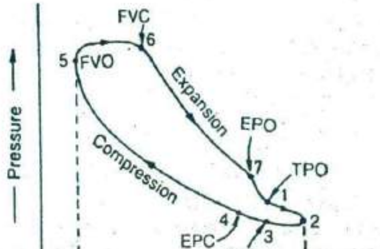


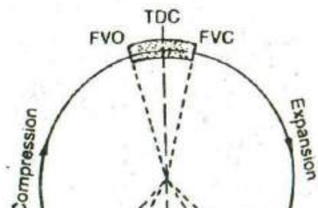
Fig. 26.14. Actual indicator diagram for a two stroke cycle diesel engine.

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.

Note : All the notes given at the end of two-stroke cycle petrol engine are valid for this cycle also.

26.22. Valve Timing Diagram for a Two-stroke Cycle Diesel Engine

In the valve timing diagram, as shown in Fig. 26.15, 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. 26.15. Valve timing diagram for a two stroke cycle 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.

26.23. Scavenging of I.C. Engines

We have already discussed in Art. 26.5, the sequence of operations in a cycle of an I.C. engine. The last stroke of an I.C. 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 I.C. 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 as 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 engines.* 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 are 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.

26.24. Types of Scavenging

Though there are many types of scavenging, yet the following are important from the subject point of view :

1. *Crossflow 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 Fig. 26.16 (*a*).

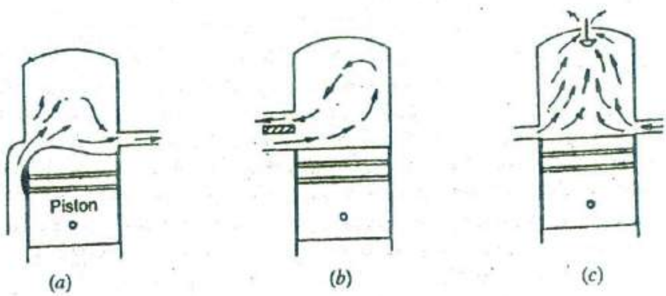


Fig. 26.16. Types of scavenging.

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 Fig. 26.16 (*b*).

3. *Uniflow scavenging.* In this method, the fresh charge, while entering from one side (or sometimes 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 Fig. 26.16 (*c*).

26.25. Detonation in I.C. 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 :

1. The shape of the combustion chamber,
2. The relative position of the sparking plugs in case of petrol engines,
3. The chemical nature of the fuel,
4. The initial temperature and pressure of the fuel, and

5. 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 unburnt 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 :

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

26.26. Rating of S.I. 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 and 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.I. 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 a 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.

26.27. Rating of C.I. Engine Fuels--Cetane Number

The knocking tendency is also found in compression ignition (C.I.) engines with an effect similar to that of S.I. 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.

Note : The knocking in C.I. engines may be controlled by decreasing ignition lag. The shorter the ignition lag, the less is the tendency to knock.

26.28. 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, and 2. Magneto ignition system.

These ignition systems are discussed, in detail, in the following pages :

26.29. 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 in Fig. 26.17. 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.

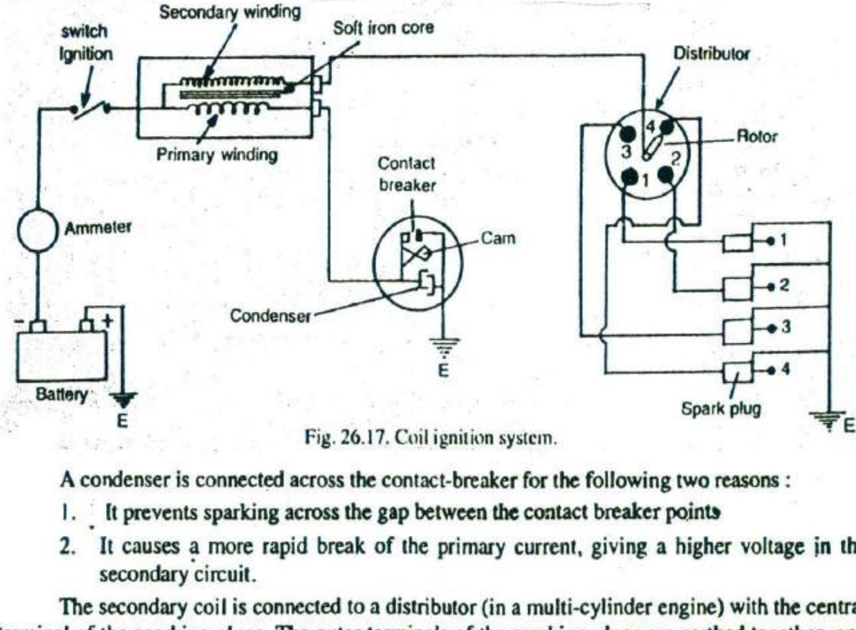


Fig. 26.17. Coil ignition system.

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

1. It prevents sparking across the gap between the contact breaker points
2. 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 connect 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 magnetic field 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

* A four lobed cam for four cylinder engine is an essential component of the make and break arrangement. It is rotated at half the engine speed.

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.

26.30. Magneto Ignition System

The magneto ignition system, as shown in Fig. 26.18, has the same principle of working as that of coil ignition system, except that no battery is required, as the magneto acts as its own generator.

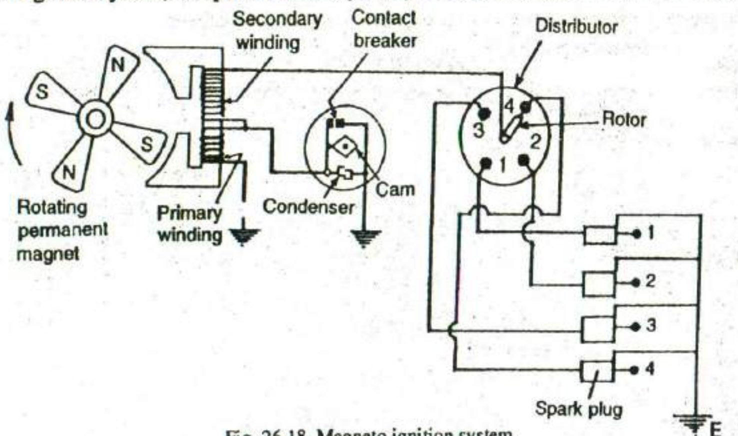


Fig. 26.18. Magneto ignition system.

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.

26.31. Fuel Injection System for Diesel Engines

The following 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.

(*a*) Common rail system, and

(*b*) Individual pump system.

These systems are discussed, in detail, as below :

(*a*) *Common rail system.* In the common rail system, as shown in Fig. 26.19, a multi-cylinder high pressure pump is used to supply the fuel at a high pressure to a common rail or header.

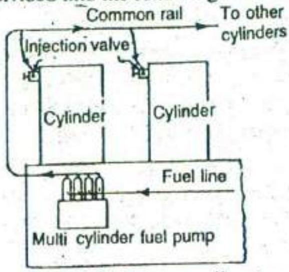


Fig. 26.19. Common rail system.

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.

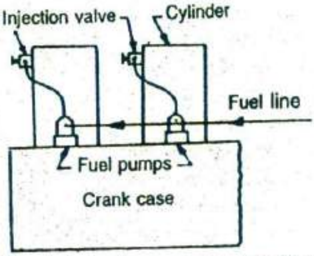


Fig. 26.20. Individual pump system of fuel injection.

(*b*) *Individual pump system.* In the individual pump system, as shown in Fig. 26.20, 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.

26.32. 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 :

1. The overheating causes thermal stresses in the engine parts, which may lead to their distortion.
2. The overheating reduces strength of the piston. The overheating may cause even seizure of the piston.
3. The overheating causes decomposition of the lubricating oil, which may cause carbon deposit on the engine and piston head.
4. The overheating causes burning of valves and valve seats.
5. The overheating reduces volumetric efficiency of the engine.
6. The overheating increases tendency of the detonation.

In order 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 :

1. It should be capable of removing about 30% of the 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.
2. 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.

26.33. 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 in Fig. 26.21, 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.

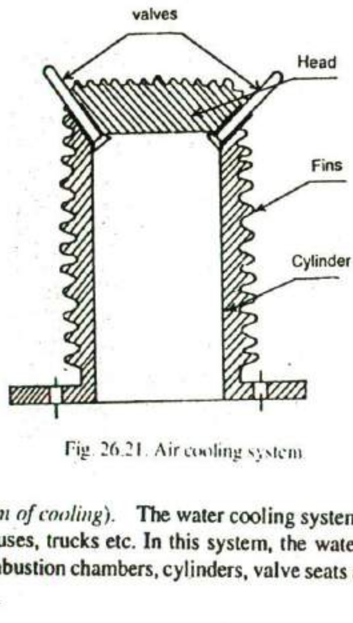


Fig. 26.21. Air cooling system.

2. *Water cooling system (Thermosyphon system of cooling).* The water cooling system as shown in Fig. 26.22, 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

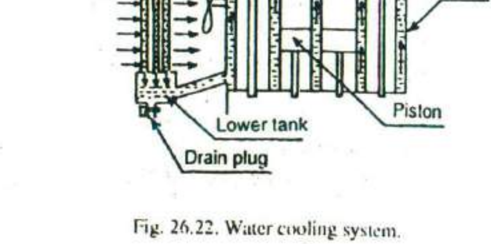


Fig. 26.22. Water cooling system.

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.

26.34. Comparison of Air Cooling and Water Cooling Systems

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

S.No.	*Air cooling system*	*Water cooling system*
1.	The design of this system is simple and less costly.	The design of this system is complicated and more costly.
2.	The mass of cooling system (per b.p. of the engine) is very less.	The mass of cooling system (per b.p. of the engine) is much more.
3.	The fuel consumption (per b.p. of the engine) is more.	The fuel consumption (per b.p. of the engine) is less.
4.	Its installation and maintenance is very easy and less costly.	Its installation and maintenance is difficult and more costly.
5.	There is no danger of leakage or freezing of the coolant.	There is a danger of leakage or freezing of the coolant.
6.	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.

26.35. Supercharging of I.C. 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).

26.36. 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 methods 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 in its outward appearance, but differs in action. There are many types of rotary pumps, but gear type, lobe type and vane type are commonly used.

26.37. 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.

26.38. Lubrication System for I.C. 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 in Fig. 26.23. A small hole is 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.

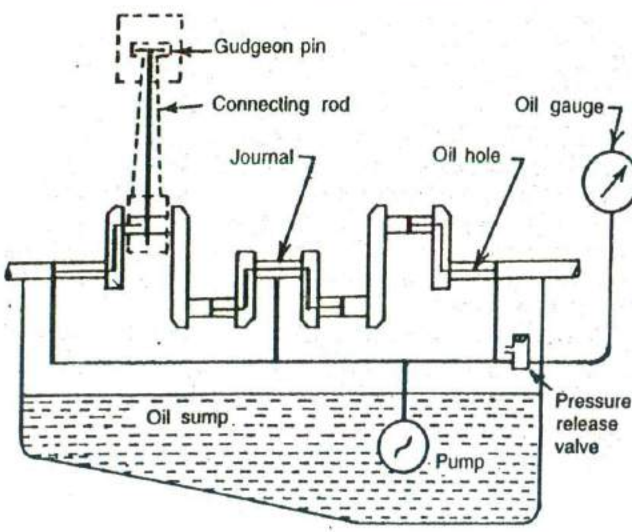


Fig.26.23. Splash lubrication.

The surplus oil is thrown out, in the form of a spray, from the bearings by centrifugal action. The 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 and is 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, as shown in Fig. 26.24.

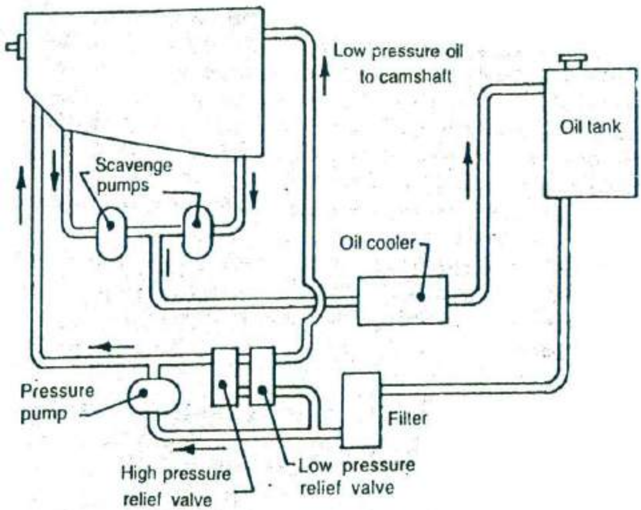


Fig. 26.24. Forced lubrication.

26.39. 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*.

26.40. 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 :

1. *Hit and miss governing.* This method of governing is widely used for I.C. engines of smaller capacity or gas engines. This method is most suitable for engines, which are frequently subjected to reduced loads and as 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 explosions are omitted or missed. This is done with help of centrifugal governor (Art. 24.11) 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 fuel delivery 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 the engine 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 (due 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.

26.41. Carburettor

The carburettor is a device for *atomising and **vaporising 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.*

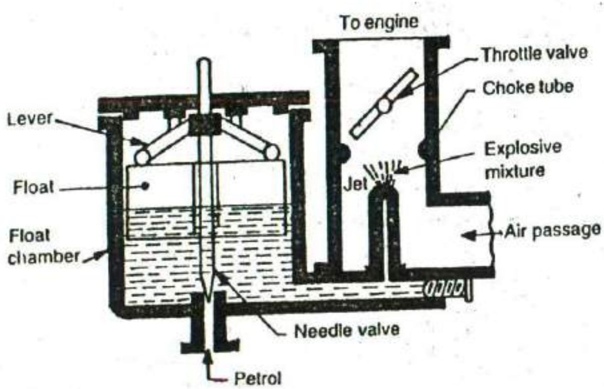


Fig. 26.25. Carburettor.

There are many types of the carburettors in use, but the simplest form of the carburettor is shown in Fig. 26.25. 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 the choke tube. The reduced diameter of the choke tube increases the velocity of air and reduces the pressure. The high velocity and low

* Atomisation is the mechanical breaking up of the liquid fuel into small particles so that every minute particle of the fuel is surrounded by air.

** Vaporisation is a change of state of fuel from a liquid to vapour.

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.

26.42. 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 10 000 to 30 000 volts.

A spark plug consists of central porcelain insulator, containing an axial electrode length wise and ground electrode welded to it. The central electrode has 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 in Fig. 26.26. 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.

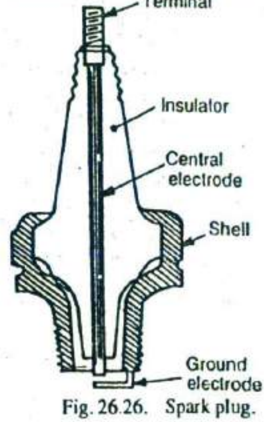
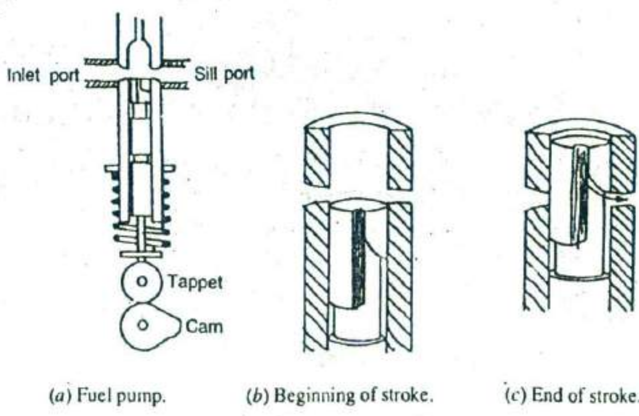


Fig. 26.26. Spark plug.

Note : The spark plug gap is kept from 0.3 mm to 0.7 mm. The experiments have shown that efficiency of the ignition system is greatly reduced if the gap is too large or too small. Sometimes, foreign materials (such as carbon) gets deposited in the spark gap. It is a source of nuisance, as it permits some of the high voltage current to bypass the gap and reduce the intensity of spark as well as engine efficiency.

26.43. 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.



(a) Fuel pump. (b) Beginning of stroke. (c) End of stroke.

Fig. 26.27

The simplified sketch of a fuel pump is shown in Fig. 26.27 (a). It consists 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. Fig. 26.27 (*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 in Fig 26.27 (*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 Fig. 26.27 (*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.

26.44. 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.

Fig. 26.28 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.

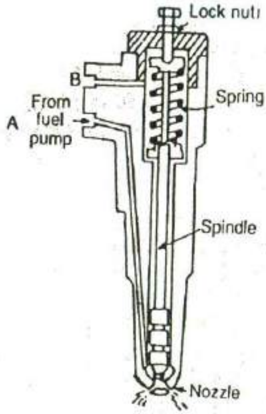


Fig. 26.28. Injector or Atomiser.

QUESTIONS

1. What is the difference between internal combustion and external combustion engines ?
2. How the internal combustion engines are classified ?
3. List the advantages and disadvantages of a two stroke cycle engine over a four stroke one.
4. Describe briefly and with appropriate sketches, the actual sequence of events in the cylinder of a petrol engine working on the four stroke cycle.
5. Discuss the working of a two stroke cycle petrol engine with the help of neat sketches.
6. Differentiate between petrol and diesel engine.
7. Explain with the help of suitable sketches, the working of a four stroke cycle and a two stroke cycle diesel engine.
8. What do you understand by air injection and solid injection system generally employed with the diesel engines ?
9. Describe the phenomenon of detonation in I.C. engines. On what factors does detonation depend ?
10. Explain what do you understand by octane and cetane number rating of a fuel.
11. Draw the electrical circuit used for battery ignition in a four stroke four cylinder engines. Explain the function of each component.

12. Discuss the cooling requirement of an I.C. engine. Describe the different methods of cooling and give specific examples where each method is employed.

13. Explain the need and methods of supercharging in I.C. Engines.

14. Discuss the lubrication system in I.C. engines.

15. What is the function of a carburettor in an S.I. engine ? Briefly explain with a neat sketch the operation of a simple float type carburettor.

16. Write short notes on the following :

(*a*) Scavenging ; (*b*) Fuel pump ; (*c*) Atomiser ; and (*d*) Spark plug.

OBJECTIVE TYPE QUESTIONS

1. In a four stroke engine, the working cycle is completed in

(*a*) one revolution of the crankshaft (*b*) two revolutions of the crankshaft

(*c*) three revolutions of the crankshaft (*d*) four revolutions of the crankshaft

2. A two stroke cycle engine gives...........the number of power strokes as compared to the four stroke cycle engine, at the same engine speed.

(*a*) half (*b*) same (*c*) double (*d*) four times

3. The thermal efficiency of a two stroke cycle engine is..........a four stroke cycle engine.

(*a*) equal to (*b*) less than (*c*) greater than

4. The theoretically correct mixture of air and petrol is

(*a*) 10 : 1 (*b*) 15 : 1 (*c*) 20 : 1 (*d*) 25 : 1

5. The thermodynamic cycle on which the petrol engine works, is

(*a*) Otto cycle (*b*) Joule cycle (*c*) Rankine cycle (*d*) Stirling cycle

6. A diesel engine has

(a) one valve (b) two valves (c) three valves (d) four valves

7. If petrol is used in a diesel engine, then

(*a*) low power will be produced (*b*) efficiency will be low

(*c*) higher knocking will occur (*d*) black smoke will be produced

8. A petrol engine has compression ratio from

(*a*) 6 to 10 (*b*) 10 to 15 (*c*) 15 to 25 (*d*) 25 to 40

9. The function of a distributor in a coil ignition system of I.C. engines is

(*a*) to distribute the spark (*b*) to distribute the power

(*c*) to distribute the current (*d*) to time the spark

10. Supercharging................the power developed by the engine.

(*a*) has no effect on (*b*) increases (*c*) decreases

11. A carburettor is used to supply

(*a*) petrol, air and lubricating oil (*b*) air and diesel

(*c*) petrol and lubricating oil (*d*) petrol and air

12. A spark plug gap is kept from

(*a*) 0.3 to 0.7 mm (*b*) 0.2 to 0.8 mm

(*c*) 0.4 to 0.9 mm (*d*) 0.6 to 1.0 mm

13. The knocking tendency in spark ignition engines may be decreased by

(*a*) controlling the air fuel mixture (*b*) controlling the ignition timing

(*c*) reducing the compression ratio (*d*) all of these

14. The violent sound pulsations within the cylinder of an I.C. engine are due to
    (*a*) detonation (*b*) turbulence (*c*) pre-ignition (*d*) none of these
15. Which of the following does not relate to a spark ignition engine ?
    (*a*) Ignition coil (*b*) Spark plug (*c*) Distributor (*d*) Fuel injector

ANSWERS

1. (*b*)	2. (*c*)	3. (*b*)	4. (*b*)	5. (*a*)
6. (*c*)	7. (*c*)	8. (*a*)	9. (*d*)	10. (*b*)
11. (*d*)	12. (*a*)	13. (*c*)	14. (*a*)	15. (*d*)

27

Testing of Internal Combustion Engines

1. Introduction. 2. Thermodynamic Tests for I.C. Engines. 3. Indicated Mean Effective Pressure. 4. Indicated Power. 5. Morse Test. 6. Brake Power. 7. Efficiencies of an I.C. Engine. 8. Air Consumption 9. Heat Balance Sheet.

27.1. Introduction

In the last chapter, we have discussed the working *i.e.* operation of the I.C. engines. As a matter of fact, when an I.C. engine is designed and manufactured, then it is tested in a laboratory. The purpose of testing are :

1. To determine the information, which can not be obtained by calculations.
2. To confirm the data used in design, the validity of which may be doubtful.
3. To satisfy the customer regarding the performance of the engine.

Note : By performance, we mean the operation of all variables relating to the working of the engine. These variables are power, fuel consumption etc.

27.2. Thermodynamic Tests for I.C. Engines

An internal combustion engine is put to the thermodynamic tests, so as to determine the following quantities :

1. Indicated mean effective pressure ; 2. Indicated power ; 3.Speed of the engine ; 4.Brake torque ; 5. Brake Power ; 6. Mechanical Losses (Motoring test) ; 7. Mechanical efficiency ; 8. Fuel consumption ; 9. Thermal efficiency ; 10. Air consumption ; 11. Volumetric efficiency ; 12. Various temperatures ; and 13. Heat balance sheet.

It may be noted that these quantities are measured after the engine has reached the steady conditions.

27.3. Indicated Mean Effective Pressure

The indicated mean effective pressure of an engine is obtained from the indicator diagram drawn with the help of an engine indicator, by any one of the following methods :

1. By drawing the diagram on a squared paper and then finding its area by counting the number of squares.
2. By finding the area of the diagram with the help of a planimeter.
3. By mid-ordinates taken from one end to another.

In all these methods, the aim is to determine the height of a rectangle of an area equal to the area of the indicator diagram. The height of this rectangle gives the mean effective pressure.

Fig. 27.1 (*a*) shows the indicator diagram and the equivalent rectangle, *i.e.* a rectangle having the same area as that of the indicator diagram, whose length is equal to the length of the indicator diagram or card.

Let s = Scale of the pressure, *i.e.* scale of indicator spring* in bar per mm,

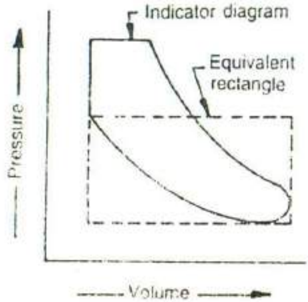
a = Area of the diagram or rectangle in mm², and

l = Length of the diagram in mm.

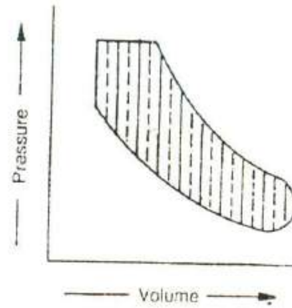
∴ Mean effective pressure (in bar)

$$= \frac{\text{Area of indicator card} \times \text{Scale of indicator spring}}{\text{Length of indicator card}} = \frac{a \times s}{l}$$

This relation is helpful for finding out the mean effective pressure by using the first two methods.



(*a*) Equivalent rectangle method.



(*b*) Mid-ordinate method.

Fig. 27.1

In case of mid-ordinate method, the indicator diagram is divided into strips of equal width as shown in Fig. 27.1 (*b*). At the centre of each strip, mid-ordinate (shown dotted) are drawn. All these mid-ordinates are added and the total is divided by number of ordinates to get the mean height of the diagram.

Let m = Mean height of the diagram, and

s = Scale of the indicator spring.

∴ Mean effective pressure

$$= m.s$$

Note : The mean effective pressure calculated on the basis of the theoretical indicator diagram, is known as *theoretical mean effective pressure*. If it is based on the actual indicator diagram, then it is called *actual mean effective pressure.*

27.4. Indicated Power

The indicated power (briefly written as I.P.) is the power actually developed by the engine cylinder. It is based on the information obtained from the indicator diagram of the engine.

Let p_m = Actual mean effective pressure as obtained from the indicator diagram in bar ;

L = Length of stroke in metres ;

A = Area of the piston in m² ;

N = Speed of the engine in r.p.m., and

* An engine indicator is, usually, provided with a set of accurately calibrated springs each of which is stamped with a number. This number indicates the pressure (in bar) required to produce a deflection of 1 mm. The number is sometimes known as spring number.

$$n = \text{Number of working strokes per minute}$$

$$= N \qquad \text{... (For two stroke cycle engine)}$$

$$= N/2 \qquad \text{... (For four stroke cycle engine)}$$

$$\therefore \text{ Indicated power, I.P.} = \frac{p_m \times 10^5 \times L \times A \times n}{60} \text{ watts}$$

$$= \frac{100\, p_m\, L\, A\, n}{60} \text{ kW} \qquad \text{... (For single cylinder engine)}$$

$$= \frac{100\, K\, p_m\, L\, A\, n}{60} \text{ kW} \qquad \text{... (For multi-cylinder engine)}$$

where K = Number of cylinders.

27.5. Morse Test

The Morse test is adopted to find the indicated power of each cylinder of a high speed I.C. engine, without using indicator diagram. The test is carried out as follows :

Consider a four cylinder engine. First of all, the brake power of the engine, when all the cylinders are in operation, is measured accurately (by means of a brake dynamometer) at a constant speed and load. Now, one of the cylinders (say cylinder 1) is cut-off so that it does not develop any power. This is done by short circuiting the spark plug of the cylinder in petrol engines and cutting-off individual fuel supply in diesel engines. The speed of the engine decreases and in order to bring the speed back to the original speed, the load on the engine is reduced. The brake power is now measured in this new condition which gives the brake power of the remaining three cylinders.

In the similar way, each cylinder is cut-off one by one and the brake power of the remaining three cylinders is determined by correcting the engine speed, if necessary.

Let I_1, I_2, I_3 and I_4 = Indicated power of each individual cylinder.

F_1, F_2, F_3 and F_4 = Frictional power of each individual cylinder.

We know that total brake power of the engine when all the cylinders are working is given by

$$B = \text{Total indicated power} - \text{Total frictional power}$$

$$= (I_1 + I_2 + I_3 + I_4) - (F_1 + F_2 + F_3 + F_4) \qquad \text{...}(i)$$

When the cylinder No. 1 is cut-off, then $I_1 = 0$, but the frictional losses of the cylinder remain the same.

$\therefore$ Brake power of the remaining three cylinders,

$$B_1 = (0 + I_2 + I_3 + I_4) - (F_1 + F_2 + F_3 + F_4) \qquad \text{...}(ii)$$

Subtracting equation (*ii*) from equation (*i*),

$$B - B_1 = I_1$$

or Indicated power of the first cylinder,

$$I_1 = B - B_1$$

Similarly, indicated power of the second cylinder,

$$I_2 = B - B_2$$

Indicated power of the third cylinder,

$$I_3 = B - B_3$$

Indicated power of the fourth cylinder,

$$I_4 = B - B_4$$

and total indicated power of the engine,

$$I = I_1 + I_2 + I_3 + I_4$$

Example 27.1. *A single cylinder, two stroke petrol engine develops 4 kW indicated power. Find the average speed of the piston, if the mean effective pressure is 6.5 bar and piston diameter is 100 mm.*

Solution. Given : I.P. = 4 kW ; $p_m = 6.5$ bar ; $D_p = 100$ mm = 0.1 m

Let L = Length of stroke in metres, and

N = Speed of the engine in r.p.m.

∴ Number of working strokes per minute,

$$n = N \quad \ldots (\because \text{ Engine works on two stroke cycle})$$

We know that area of piston,

$$A = \frac{\pi}{4}(0.1)^2 = 7.855 \times 10^{-3} \text{ m}^2$$

∴ Indicated power (I.P.),

$$4 = \frac{100\, p_m L A n}{60} = \frac{100 \times 6.5 \times L \times 7.855 \times 10^{-3} \times N}{60} = 0.0851\, L N$$

or $$L N = 47$$

∴ Average speed of the piston

$$= 2 L N = 2 \times 47 = 94 \text{ m/s } \textbf{Ans.}$$

Example 27.2. *In a laboratory experiment, the following observations were noted during the test of a four stroke Diesel engine :*

Area of indicator diagram = 420 mm² ; Length of indicator diagram = 62 mm ; Spring number = 1.1 bar/mm ; Diameter of piston = 100 mm ; Length of stroke = 150 mm ; Engine speed = 450 r.p.m. Determine : 1. Indicated mean effective pressure, and 2. Indicated power.

Solution. Given : $a = 420$ mm² ; $l = 62$ mm ; $s = 1.1$ bar/mm ; $D_p = 100$ mm = 0.1 m ; $L = 150$ mm = 0.15 m ; $N = 450$ r.p.m.

We know that area of piston,

$$A = \frac{\pi}{4}(0.1)^2 = 7.855 \times 10^{-3} \text{ m}^2$$

and number of working strokes per minute,

$$n = N/2 = 450/2 = 225 \quad \ldots (\because \text{ Engine works on four stroke cycle})$$

1. *Indicated mean effective pressure*

We know that indicated mean effective pressure,

$$p_m = \frac{a.s}{l} = \frac{420 \times 1.1}{62} = 7.45 \text{ bar } \textbf{Ans.}$$

2. *Indicated power*

We know that indicated power,

$$\text{I.P.} = \frac{100\, p_m L A n}{60} = \frac{100 \times 7.45 \times 0.15 \times 7.855 \times 10^{-3} \times 225}{60} \text{ kW}$$

$$= 3.29 \text{ kW } \textbf{Ans.}$$

27.6. Brake Power

The brake power (briefly written as B.P.) is the power available at the crank shaft. The brake power of an I.C. engine is, usually, measured by means of brake mechanism (prony brake or rope brake).

In case of prony brake, let

W = Brake load in newtons,

l = Length of arm in metres, and

N = Speed of the engine in r.p.m.

$\therefore$ Brake power of the engine,

$$\text{B.P.} = \frac{\text{Torque in N-m} \times \text{Angle turned in radians through 1 revolution}}{60} \times \text{R.P.M. watts}$$

$$= \frac{T \times 2\pi N}{60} = \frac{W l \times 2\pi N}{60} \text{ watts}$$

In case of rope brake, let

W = Dead load in newtons,

S = Spring balance reading in newtons,

D = Diameter of brake drum in metres,

d = Diameter of the rope in metres, and

N = Speed of the engine in r.p.m.

$\therefore$ Brake power of the engine,

$$\text{B.P.} = \frac{(W - S)\,\pi\, DN}{60} \text{ watts} \qquad \text{... [without considering diameter } (d) \text{ of the rope]}$$

$$= \frac{(W - S)\,\pi\,(D + d)\,N}{60} \text{ watts} \qquad \text{... [Considering diameter (d) of the rope]}$$

Example 27.3. *The following data were recorded during a test on an oil engine :*

Speed of the engine = 1000 r.p.m. ; Load on the brake = 1000 N ; Length of the brake = 750 mm.

Determine : 1. Brake torque ; and 2. Brake power of the engine.

Solution. Given : $N = 1000$ r.p.m. ; $W = 1000$ N ; $l = 750$ mm $= 0.75$ m

1. *Brake torque*

We know that brake torque,

$$T = W l = 1000 \times 0.75 = 750 \text{ N-m } \textbf{Ans.}$$

2. *Brake power of the engine*

We know that brake power of the engine,

$$\text{B.P.} = \frac{T \times 2\,\pi N}{60} = \frac{750 \times 2\,\pi \times 1000}{60} = 78\,550 \text{ W} = 78.55 \text{ kW } \textbf{Ans.}$$

Example 27.4. *A rope brake has brake wheel diameter of 600 mm and the diameter of rope is 5 mm. The dead load on the brake is 210 N and spring balance reads 30 N. If the engine makes 450 r.p.m., find the brake power developed.*

Solution. Given : $D = 600$ mm $= 0.6$ m ; $d = 5$ mm $= 0.005$ m ; $W = 210$ N ; $S = 30$ N ; $N = 450$ r.p.m.

We know that brake power developed,

$$\text{B.P.} = \frac{(W - S)\,\pi\,(D + d)\,N}{60} = \frac{(210 - 30)\,\pi\,(0.6 + 0.005)\,450}{60}\ \text{W}$$

$$= 2570\ \text{W} = 2.57\ \text{kW}\ \textbf{Ans.}$$

27.7. Efficiencies of I.C. Engine

The efficiency of an engine is defined as the ratio of work done to the energy supplied to an engine. The following efficiencies of an I.C. engine are important from the subject point of view :

1. *Mechanical efficiency.* It is the ratio of brake power (B.P.) to the indicated power (I.P.). Mathematically, mechanical efficiency,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}}$$

It may be observed that the mechanical efficiency is always less than unity (*i.e.* 100 %) because some power is lost in overcoming the engine friction. In other words, the indicated power is always greater than brake power. The power, which is lost in overcoming the engine friction, is known as ***frictional power***. Therefore frictional power,

$$\text{F.P.} = \text{I.P.} - \text{B.P.}$$

2. *Overall efficiency.* It is the ratio of work obtained at the crankshaft in a given time to the energy supplied by the fuel during the same time.

Let m_f = Mass of fuel consumed in kg per hour, and

C = Calorific value of fuel in kJ/kg of fuel.

$\therefore$ Energy supplied by the fuel per minute

$$= \frac{m_f \times C}{60}\ \text{kJ}$$

and work obtained at the crankshaft per minute

$$= \text{B.P.} \times 60\ \text{kJ} \qquad \ldots (\because \text{B.P. is in kW and } 1\ \text{kW} = 1\ \text{kJ/s})$$

$$\therefore \text{Overall efficiency,}\quad \eta_O = \frac{\text{B.P.} \times 60 \times 60}{m_f \times C} = \frac{\text{B.P.} \times 3600}{m_f \times C}$$

3. *Indicated thermal efficiency.* It is the ratio of the heat equivalent to one kW hour to the heat in the fuel per I.P. hour. Mathematically, thermal efficiency,

$$\eta_t = \frac{\text{Heat equivalent to one kW hour}}{\text{Heat in fuel per I.P. hour}}$$

$$= \frac{3600}{\dfrac{m_f \times C}{\text{I.P.}}} = \frac{\text{I.P.} \times 3600}{m_f \times C}$$

Note : The ratio $\frac{m_f}{\text{I.P.}}$ is known as specific fuel consumption per I.P. per hour.

4. *Brake thermal efficiency.* It is the ratio of the heat equivalent to one kW hour to the heat in fuel per B. P. hour. It is also known as *overall thermal efficiency* of the engine. Mathematically, brake thermal efficiency,

$$\eta_b = \frac{\text{Heat equivalent to one kW hour}}{\text{Heat in fuel per B.P. hour}}$$

$$= \frac{3600}{\frac{m_f \times C}{\text{B.P.}}} = \frac{\text{B.P.} \times 3600}{m_f \times C}$$

Note : The ratio $\frac{m_f}{\text{B.P.}}$ is known as specific fuel consumption per B.P. per hour.

5. *Air standard efficiency.* The air standard efficiency of an I.C. engine may also be obtained mathematically from the general expression for the air standard efficiency *i.e.*

$$\eta_{ase} = 1 - \frac{1}{r^{\gamma-1}} \qquad \text{... (For petrol engines)}$$

$$= 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}\right] \qquad \text{... (For Diesel engines)}$$

where r = Compression ratio,

γ = Ratio of specific heats, and

ρ = Cut-off ratio.

6. *Relative efficiency.* It is also known as efficiency ratio. The relative efficiency of an I.C. engine is the ratio of indicated thermal efficiency to the air standard efficiency. Mathematically,

$$\text{Relative efficiency} = \frac{\text{Indicated thermal efficiency}}{\text{Air standard efficiency}}$$

7. *Volumetric efficiency.* It is the ratio of actual volume of charge admitted during the suction stroke at N.T.P. to the swept volume of the piston. Mathematically, volumetric efficiency,

$$\eta_v = \frac{\text{Volume of charge admitted at N.T.P.}}{\text{Swept volume of the piston}} = \frac{v_a}{v_s}$$

Note : The volumetric efficiency may also be defined as the ratio of the mass of actual charge admitted to the swept mass of the charge at N.T.P.

Example 27.5. *A gas engine has piston diameter of 150 mm, length of stroke 400 mm and mean effective pressure 5.5 bar. The engine makes 120 explosions per minute. Determine the mechanical efficiency of the engine, if its B.P. is 5 kW.*

Solution. Given : D_p = 150 mm = 0.15 m ; L = 400 mm = 0.4 m ; p_m = 5.5 bar ; n = 120 ; B.P. = 5 kW

We know that area of the piston,

$$A = \frac{\pi}{4}(0.15)^2 = 0.0177 \text{ m}^2$$

and indicated power,

$$\text{I.P.} = \frac{100\, p_m L A n}{60} = \frac{100 \times 5.5 \times 0.4 \times 0.0177 \times 120}{60} = 7.79 \text{ kW}$$

We know that mechanical efficiency of the engine,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{5}{7.79} = 0.642 \text{ or } 64.2\% \textbf{ Ans.}$$

Example 27.6. *A four cylinder two stroke cycle petrol engine develops 23.5 kW brake power at 2500 r.p.m. The mean effective pressure on each piston is 8.5 bar and the mechanical efficiency is 85%. Calculate the diameter and stroke of each cylinder, assuming the length of stroke equal to 1.5 times the diameter of cylinder.*

Solution. Given : $K = 4$; B.P. = 23.5 kW ; $N = 2500$ r.p.m. ; $p_m = 8.5$ bar ; $\eta_m = 85\% = 0.85$

Let D_c = Diameter of cylinder, and

L = Length of stroke = $1.5\, D_c$ . . . (Given)

We know that area of the cylinder,

$$A = \frac{\pi}{4}(D_c)^2 = 0.7855 (D_c)^2$$

and number of working strokes per minute,

$$n = N = 2500 \qquad \ldots (\because \text{Engine works on two stroke cycle})$$

We know that indicated power,

$$\text{I.P.} = \frac{\text{B.P.}}{\eta_m} = \frac{23.5}{0.85} = 27.65 \text{ kW}$$

We also know that indicated power (I.P.),

$$27.65 = \frac{100\, K\, p_m\, L\, A\, n}{60}$$

$$= \frac{100 \times 4 \times 8.5 \times 1.5\, D_c \times 0.7855\, (D_c)^2\, 2500}{60} = 166\,920\,(D_c)^3$$

$\therefore$ $(D_c)^3 = 0.000\,165$ or $D_c = 0.055 \text{ m} = 55 \text{ mm}$ **Ans.**

and $L = 1.5 \times 55 = 82.5$ mm **Ans.**

Example 27.7. *During the test on single cylinder oil engine, working on the four stroke cycle and fitted with a rope brake, the following readings are taken :*

Effective diameter of brake wheel = 630 mm ; Dead load on brake = 200 N ; Spring balance reading = 30 N ; Speed = 450 r.p.m. ; Area of indicator diagram = 420 mm² ; Length of indicator diagram = 60 mm ; Spring scale = 1.1 bar per mm ; Diameter of cylinder = 100 mm ; Stroke = 150 mm ; Quantity of oil used = 0.815 kg/h ; Calorific value of oil = 42 000 kJ/kg.

Calculate brake power, indicated power, mechanical efficiency, brake thermal efficiency and brake specific fuel consumption.

Solution. Given : $K = 1$; $D = 630$ mm = 0.63 m ; $W = 200$ N ; $S = 30$ N ; $N = 450$ r.p.m. ; $a = 420 \text{ mm}^2$; $l = 60$ mm ; $s = 1.1$ bar/mm ; $D_c = 100$ mm = 0.1 m ; $L = 150$ mm = 0.15 m ; $m_f = 0.815$ kg/h ; $C = 42\,000$ kJ/kg

Brake power

We know that brake power,

$$\text{B.P.} = \frac{(W - S)\,\pi D N}{60} = \frac{(200 - 30)\,\pi \times 0.63 \times 450}{60} = 2520 \text{ W}$$

$$= 2.52 \text{ kW } \textbf{Ans.}$$

Indicated power

We know that indicated mean effective pressure,

$$p_m = \frac{a \times s}{l} = \frac{420 \times 1.1}{60} = 7.7 \text{ bar}$$

Area of the cylinder, $A = \frac{\pi}{4}(D_c)^2 = \frac{\pi}{4}(0.1)^2 = 7.855 \times 10^{-3} \text{ m}^2$

and number of working strokes per min,

$$n = N/2 = 450/2 = 225 \quad \ldots (\because \text{Engine works on four stroke cycle})$$

We know that indicated power,

$$\text{I.P.} = \frac{100\, K\, p_m\, L\, A\, n}{60} = \frac{100 \times 1 \times 7.7 \times 0.15 \times 7.855 \times 10^{-3} \times 225}{60}\ \text{kW}$$

$$= 3.4\ \text{kW}\ \textbf{Ans.}$$

Mechanical efficiency

We know that mechanical efficiency,

$$\eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{2.52}{3.4} = 0.7418 \text{ or } 74.18\%\ \textbf{Ans.}$$

Brake thermal efficiency

We know that brake thermal efficiency,

$$\eta_b = \frac{\text{B.P.} \times 3600}{m_f \times C} = \frac{2.52 \times 3600}{0.815 \times 42\,000} = 0.265 \text{ or } 26.5\%\ \textbf{Ans.}$$

Brake specific fuel consumption

We know that brake specific fuel consumption

$$= \frac{m_f}{\text{B.P.}} = \frac{0.815}{2.52} = 0.323\ \text{kg / B.P. / h}\ \textbf{Ans.}$$

Example 27.8. *An engine uses 6.5 kg of oil per hour of calorific value 30 000 kJ/kg. If the B.P. of the engine is 22 kW and mechanical efficiency 85%, calculate : 1. Indicated thermal efficiency ; 2. Brake thermal efficiency ; and 3. Specific fuel consumption in kg/B.P./h.*

Solution. Given : $m_f = 6.5$ kg/h ; $C = 30\,000$ kJ/kg ; B.P. = 22 kW ; $\eta_m = 85\% = 0.85$

1. *Indicated thermal efficiency*

We know that indicated power,

$$\text{I.P.} = \frac{\text{B.P.}}{\eta_m} = \frac{22}{0.85} = 25.88\ \text{kW}$$

$\therefore$ Indicated thermal efficiency,

$$\eta_i = \frac{\text{I.P.} \times 3600}{m_f \times C} = \frac{25.88 \times 3600}{6.5 \times 30\,000} = 0.48 \text{ or } 48\%\ \textbf{Ans.}$$

2. *Brake thermal efficiency*

We know that brake thermal efficiency,

$$\eta_b = \frac{\text{B.P.} \times 3600}{m_f \times C} = \frac{22 \times 3600}{6.5 \times 30\,000} = 0.406 \text{ or } 40.6\%\ \textbf{Ans.}$$

3. *Specific fuel consumption*

We know that specific fuel consumption

$$= \frac{m_f}{\text{B.P.}} = \frac{6.5}{22} = 0.295\ \text{kg / B.P. / h}\ \textbf{Ans.}$$

Example 27.9. *A four cylinder engine running at 1200 r.p.m. gave 18.6 kW brake power. The average torque when one cylinder was cut out was 105 N-m. Determine the indicated thermal* *efficiency if the calorific value of the fuel is 42 000 kJ/kg and the engine uses 0.34 kg of petrol per brake power hour.*

Solution. Given : $K = 4$; $N = 1200$ r.p.m. ; B.P. = 18.6 kW ; $T = 105$ N-m ; $C = 42\,000$ kJ/kg ; $m_f = 0.34 \text{ kg / B.P. / h} = 0.34 \times 18.6 = 6.324 \text{ kg / h}$

We know that brake power per cylinder

$$= \frac{18.6}{4} = 4.65 \text{ kW}$$

$\therefore$ Brake power for three cylinders (*i.e.* when one cylinder is cut-out)

$$= 4.65 \times 3 = 13.95 \text{ kW}$$

Since the average torque (T) for three cylinders (*i.e.* when one cylinder is cut-out) is given as $T = 105$ N-m, therefore

Brake power for the three cylinders,

$$= \frac{T \times 2\pi N}{60} = \frac{105 \times 2\pi \times 1200}{60} = 13\,200 \text{ W} = 13.2 \text{ kW}$$

and frictional power per cylinder

$$= 13.95 - 13.2 = 0.75 \text{ kW}$$

$\therefore$ Total frictional power for four cylinders,

$$\text{F.P.} = 0.75 \times 4 = 3 \text{ kW}$$

We know that indicated power,

$$\text{I.P.} = \text{B.P.} + \text{F.P.} = 18.6 + 3 = 21.6 \text{ kW}$$

$\therefore$ Indicated thermal efficiency,

$$\eta_i = \frac{\text{I.P.} \times 3600}{m_f \times C} = \frac{21.6 \times 3600}{6.324 \times 42\,000} = 0.293 \quad \text{or} \quad 29.3\% \textbf{ Ans.}$$

Example 27.10. *A four stroke, six cylinder gas engine with a stroke volume of 1.75 litres develops 25 kW at 480 r.p.m.. The mean effective pressure is 6 bar. Find the average number of times each cylinder misfired in one minute.*

Solution. Given $K = 6$; $v_s = LA = 1.75 \text{ litres} = 1.75 \times 10^{-3} \text{ m}^3$; I.P = 25 kW ; $N = 480$ r.p.m ; $p_m = 6$ bar

Let $\quad n_1 =$ Number of working strokes produced per minute.

We know that indicated power (I.P.),

$$25 = \frac{100\, K\, p_m\, L\, A\, n_1}{60} = \frac{100 \times 6 \times 6 \times 1.75 \times 10^{-3} \times n_1}{60} = 0.105\, n_1$$

$$\therefore \quad n_1 = 238$$

But the number of working strokes per minute are given as

$$n = N/2 = 480/2 = 240 \qquad \ldots (\because \text{ Engine works on four stroke cycle})$$

$\therefore$ Number of times each cylinder misfired in one minute

$$= n - n_1 = 240 - 238 = 2 \textbf{ Ans.}$$

Example 27.11. *The diameter and stroke length of a single cylinder two stroke gas engine, working on the constant volume cycle, are 200 mm and 300 mm respectively with clearance volume 2.78 litres.*

When the engine is running at 135 r.p.m., the indicated mean effective pressure was 5.2 bar and the gas consumption 8.8 m³/hour. If the calorific value of the gas used is 16 350 kJ/m³, find 1. air standard efficiency ; 2. indicated power developed by the engine ; and 3. indicated thermal efficiency of the engine.

Solution. Given : $D_c = 200$ mm $= 0.2$ m ; $L = 300$ mm $= 0.3$ m ; $v_c = 2.78$ litres $= 0.002\ 78$ m^3 ; $N = 135$ r.p.m.; $p_m = 5.2$ bar ; $m_f = 8.8$ m^3/h ; $C = 16\ 350$ kJ/m^3

1. *Air standard efficiency*

We know that area of the cylinder,

$$A = \frac{\pi}{4}(0.2)^2 = 0.031\ 42 \text{ m}^2$$

$\therefore$ Stroke volume, $v_s = A\,L = 0.031\ 42 \times 0.3 = 0.009\ 426 \text{ m}^3$

and compression ratio,

$$r = \frac{\text{Total volume of cylinder}}{\text{Clearance volume}} = \frac{v_c + v_s}{v_c}$$

$$= \frac{0.002\ 78 + 0.009\ 426}{0.002\ 78} = 4.4$$

We know that air standard efficiency,

$$\eta_{ase} = 1 - \frac{1}{r^{\gamma - 1}} = 1 - \frac{1}{(4.4)^{1.4-1}} = 1 - 0.553 = 0.447 \text{ or } 44.7\% \textbf{ Ans.}$$

2. *Indicated power developed by the engine*

We know that indicated power developed by the engine,

$$\text{I.P.} = \frac{100\,p_m\,L\,A\,n}{60} = \frac{100 \times 5.2 \times 0.3 \times 0.031\ 42 \times 135}{60} \text{ kW}$$

$\ldots (\because n = N$, for two stroke cycle engine)

$$= 11.03 \text{ kW } \textbf{Ans.}$$

3. *Indicated thermal efficiency*

We know that indicated thermal efficiency,

$$\eta_i = \frac{\text{I.P.} \times 3600}{m_f \times C} = \frac{11.03 \times 3600}{8.8 \times 16\ 350} = 0.276 \text{ or } 27.6\% \textbf{ Ans.}$$

Example 27.12. *A four stroke petrol engine 80 mm bore, 100 mm stroke, is tested at full throttle at constant speed. The fuel supply is fixed at 0.068 kg/min and the plugs of the four cylinders are successively short circuited without change of speed, brake torque being correspondingly adjusted. The brake power measurements are the following :*

With all cylinders firing	=	*12.5 kW*
With cylinder No. 1 cut off	=	*9 kW*
With cylinder No. 2 cut off	=	*9.15 kW*
With cylinder No. 3 cut off	=	*9.2 kW*
With cylinder No. 4 cut off	=	*9.1 kW*

Determine I.P. of the engine under these conditions. Also determine the indicated thermal efficiency. Calorific value of the fuel is 44 100 kJ/kg. Compare this efficiency with the air standard value. Clearance volume of one cylinder is 70×10^3 mm^3.

Solution. Given : D_c = 80 mm = 0.08 m ; L = 100 mm = 0.1 m ; m_f = 0.068 kg/min = 4.08 kg/h ; B = 12.5 kW ; B_1 = 9 kW ; B_2 = 9.15 kW ; B_3 = 9.2 kW ; B_4 = 9.1 kW ; C = 44100 kJ/kg ; $v_c = 70 \times 10^3$ mm^3

I.P. of the engine

We know that indicated power produced in cylinder 1,

$$I_1 = B - B_1 = 12.5 - 9 = 3.5 \text{ kW}$$

Indicated power produced in cylinder 2,

$$I_2 = B - B_2 = 12.5 - 9.15 = 3.35 \text{ kW}$$

Indicated power produced in cylinder 3,

$$I_3 = B - B_3 = 12.5 - 9.2 = 3.3 \text{ kW}$$

and indicated power produced in cylinder 4,

$$I_4 = B - B_4 = 12.5 - 9.1 = 3.4 \text{ kW}$$

∴ Indicated power of the engine,

$$\text{I.P.} = I_1 + I_2 + I_3 + I_4 = 3.5 + 3.35 + 3.3 + 3.4 = 13.55 \text{ kW} \quad \textbf{Ans.}$$

Indicated thermal efficiency

We know that indicated thermal efficiency,

$$\eta_{I} = \frac{\text{I.P.} \times 3600}{m_f \times C} = \frac{13.55 \times 3600}{4.08 \times 44\,100} = 0.271 \text{ or } 27.1\% \text{ Ans.}$$

Air standard efficiency

We know that swept volume

$$v_s = \frac{\pi}{4}(D_c)^2 L = \frac{\pi}{4}(80)^2\,100 = 503 \times 10^3 \text{ mm}^3$$

∴ Compression ratio,

$$r = \frac{\text{Total cylinder volume}}{\text{Clearance volume}} = \frac{v_c + v_s}{v_c}$$

$$= \frac{70 \times 10^3 + 503 \times 10^3}{70 \times 10^3} = 8.18$$

We know that air standard efficiency,

$$\eta_{ase} = 1 - \frac{1}{r^{\gamma - 1}} = 1 - \frac{1}{(8.18)^{1.4-1}} \qquad \text{...(Taking } \gamma = 1.4 \text{ for air)}$$

$$= 1 - 0.431 = 0.569 \text{ or } 56.9\% \textbf{ Ans.}$$

Ratio of air standard efficiency to indicated thermal efficiency

$$= \frac{0.569}{0.271} = 2.1$$

Thus air standard efficiency is 2.1 times the indicated thermal efficiency. **Ans.**

Example 27.13. *A four stroke diesel engine has a cylinder bore of 150 mm and a stroke of 250 mm. The crankshaft speed is 300 r.p.m. and fuel consumption is 1.2 kg/h, having a calorific value of 39 900 kJ/kg. The indicated mean effective pressure is 5.5 bar. If the compression ratio is 15 and cut-off ratio is 1.8, calculate the relative efficiency, taking* $\gamma = 1.4$.

Solution. Given : $D_c = 150$ mm = 0.15 m; $L = 250$ mm = 0.25 m ; $N = 300$ r.p.m. ; $m_f = 1.2$ kg/h ; $C = 39\,900$ kJ/kg ; $p_m = 5.5$ bar ; $r = 15$; $\rho = 1.8$; $\gamma = 1.4$

We know that area of the cylinder,

$$A = \frac{\pi}{4}(0.15)^2 = 0.0177 \text{ m}^2$$

and number of working stroke per minute,

$$n = N/2 = 300/2 = 150 \qquad \ldots (\because \text{ Engine works on four stroke cycle})$$

$\therefore$ Indicated power,

$$\text{I.P.} = \frac{100\, p_m\, L\, A\, n}{60} = \frac{100 \times 5.5 \times 0.25 \times 0.0177 \times 150}{60} = 6.1 \text{ kW}$$

and indicated thermal efficiency,

$$\eta_i = \frac{\text{I.P.} \times 3600}{m_f \times C} = \frac{6.1 \times 3600}{1.2 \times 39\,900} = 0.4586 \text{ or } 45.86\%$$

We know that air standard efficiency for diesel engine,

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

$$= 1 - \frac{1}{(15)^{1.4-1}}\left[\frac{(1.8)^{1.4}-1}{1.4\,(1.8-1)}\right] = 1 - 0.386 = 0.614 \text{ or } 61.4\%$$

$\therefore$ Relative efficiency,

$$\eta_r = \frac{\text{Indicated thermal efficiency}}{\text{Air standard efficiency}} = \frac{0.4586}{0.614}$$

$$= 0.747 \text{ or } 74.7\% \textbf{ Ans.}$$

Example 27.14. *A petrol engine has a cylinder diameter of 60 mm and stroke 100 mm. If the mass of the charge admitted per cycle is 0.0002 kg, find the volumetric efficiency of the engine. Assume characteristic constant for the charge as 287 J/kg K.*

Solution. Given : $D_c = 60$ mm = 0.06 m ; $L = 100$ mm = 0.1 m ; $m = 0.0002$ kg

We know that swept volume of the piston,

$$v_s = \frac{\pi}{4}(0.06)^2\, 0.1 = 0.283 \times 10^{-3} \text{ m}^3$$

Let $\qquad v_a$ = Volume of charge admitted at *N.T.P.

We know that according to characteristic gas equations,

$$v_a = \frac{m\,R\,T}{p} = \frac{0.0002 \times 287 \times 273}{1.013 \times 10^5} \text{ m}^3 \qquad \ldots (\because p\,v = m\,R\,T)$$

$$= 0.155 \times 10^{-3} \text{ m}^3$$

---

* N.T.P means normal temperature and pressure, *i.e.* temperature (T) of 0°C or 273 K and pressure 1.013 bar or 1.013×10^5 N/m^2.

$\therefore$ Volumetric efficiency,

$$\eta_v = \frac{v_a}{v_s} = \frac{0.155 \times 10^{-3}}{0.283 \times 10^{-3}} = 0.548 \text{ or } 54.8\% \textbf{ Ans.}$$

Example 27.15. *Find the engine dimensions of a two cylinder, two stroke I. C. engine from the following data :*

Engine speed = 4000 r.p.m. ; Volumetric efficiency = 0.77 ; Mechanical efficiency = 0.75 ; Fuel consumption = 10 litres/h (specific gravity = 0.73) ; Air-fuel ratio = 18:1 ; Piston speed = 600 m/min ; Indicated mean effective pressure = 5 bar.

Find also the brake power. Take R for gas mixture as 281 J/kg K at S.T.P.

Solution. Given : $K = 2$; $N = 4000$ r.p.m. ; $\eta_v = 0.77$; $\eta_m = 0.75$; $m_f = 10 \times 0.73 = 7.3$ kg / h ; $m_a / m_f = 18$; $2LN = 600$ m/min ; $p_m = 5$ bar ; $R = 281$ J/kg K

Engine dimensions

Let D_c = Diameter of the cylinder, and

L = Length of the stroke.

We know that piston speed,

$$2LN = 600$$

or

$$L = 600 / 2N = 600 / 2 \times 4000 = 0.075 \text{ m or } 75 \text{ mm } \textbf{Ans.}$$

Mass of air required $m_a = m_f \times 18 = 7.3 \times 18 = 131.4$ kg / h ... ($\because m_a/m_f = 18$)

and corresponding volume of air required at *S.T.P.,

$$v_a = \frac{m_a . R T}{p} = \frac{131.4 \times 281 \times 288}{1.013 \times 10^5} = 105 \text{ m}^3\text{/h} = 1.75 \text{ m}^3\text{/min}$$

We know that swept volume of the piston per minute,

$$v_s = \frac{\pi}{4}(D_c)^2 L \times n \times K$$

$$= \frac{\pi}{4}(D_c)^2\, 0.075 \times 4000 \times 2 = 471.3\,(D_c)^2 \text{ m}^3\text{/min}$$

... ($\because n = N$, for two stroke cycle engine)

and volumetric efficiency (η_v),

$$0.77 = \frac{v_a}{v_s} = \frac{1.75}{4.71.3\,(D_c)^2}$$

$$\therefore \quad (D_c)^2 = 4.82 \times 10^{-3} \text{ or } D_c = 0.0694 \text{ m or } 69.4 \text{ mm } \textbf{Ans.}$$

Brake power

We know that area of the cylinder,

$$A = \frac{\pi}{4}(D_c)^2 = \frac{\pi}{4}(0.0694)^2 = 0.003\,78 \text{ m}^2$$

* S.T.P. means standard temperature and pressure, *i.e.* temperature (T) of 15° C or 288 K and pressure 1.013 bar or 1.013×10^5 N/m^2.

and indicated power, $$\text{I.P.} = \frac{100\,K\,p_m\,L\,A\,n}{60} = \frac{100 \times 2 \times 5 \times 0.075 \times 0.003\,78 \times 4000}{60} \text{ kW}$$

$$= 18.9 \text{ kW}$$

$\therefore$ Brake power, $\quad$ B.P. $= \text{I.P.} \times \eta_m = 18.9 \times 0.75 = 14.17$ kW **Ans.**

Example 27.16. *A four stroke petrol engine with a compression ratio of 6.5 to 1 and total piston displacement of 5.2×10^{-3} m^3 develops 100 kW brake power and consumes 33 kg of petrol per hour of calorific value 44 300 kJ/kg at 3000 r.p.m.*

Find : 1. Brake mean effective pressure ; 2. Brake thermal efficiency ; 3. Air standard efficiency ($\gamma = 1.4$) ; and 4. Air-fuel ratio by mass.

Assume a volumetric efficiency of 80 %. One kg of petrol vapour occupies 0.26 m^3 at 1.013 bar and 15° C. Take R for air 287 J/kg K.

Solution. Given : $r = 6.5$; $v_s = LA = 5.2 \times 10^{-3}$ m^3 ; B.P. = 100 kW ; $m_f = 33$ kg/h ; $C = 44\,300$ kJ/kg ; $N = 3000$ r.p.m. ; $\gamma = 1.4$; $\eta_v = 80\% = 0.8$

1. *Brake mean effective pressure*

Let $\quad p_{mb}$ = Brake mean effective pressure in bar.

We know that brake power (B.P.),

$$100 = \frac{100\,p_{mb}\,L\,A\,n}{60} = \frac{100 \times p_{mb} \times 5.2 \times 10^{-3} \times 1500}{60} = 13\,p_{mb}$$

... ($\because$ $n = N/2$, for four stroke engine)

$\therefore \quad p_{mb} = 7.7$ bar **Ans.**

2. *Brake thermal efficiency*

We know that brake thermal efficiency,

$$\eta_b = \frac{\text{B.P.} \times 3600}{m_f \times C} = \frac{100 \times 3600}{33 \times 44\,300} = 0.246 \quad \text{or} \quad 24.6\% \textbf{ Ans.}$$

3. *Air standard efficiency*

We know that air standard efficiency,

$$\eta_{ase} = 1 - \frac{1}{r^{\gamma - 1}} = 1 - \frac{1}{(6.5)^{1.4-1}} = 1 - 0.473$$

$$= 0.527 \quad \text{or} \quad 52.7\% \textbf{ Ans.}$$

4. *Air-fuel ratio by mass*

We know that actual volume of charge admitted during the suction stroke at N. T.P.,

$$v_a = v_s \times \eta_v = 5.2 \times 10^{-3} \times 0.8 = 4.16 \times 10^{-3} \text{ m}^3 \qquad \dots (i)$$

Let $\quad v_1$ = Specific volume of petrol (*i.e.* volume of 1 kg of petrol) at N.T.P. conditions, *i.e.* at $T_1 = 0°$ C or 273 K and $p_1 = 1.013$ bar, and

v_2 = Specific volume of petrol (*i.e.* volume of 1 kg of petrol) at T_2 = 15° C or 288 K and $p_2 = 1.013$ bar

$= 0.26$ m^3 ... (Given)

We know that $\quad \dfrac{p_1 v_1}{T_1} = \dfrac{p_2 v_2}{T_2}$

$$\therefore \qquad v_1 = \frac{p_2 v_2 T_1}{p_1 T_2} = \frac{1.013 \times 0.26 \times 273}{1.013 \times 288} = 0.246 \text{ m}^3/\text{ kg of petrol}$$

Mass of petrol consumed per cycle,

$$m_1 = \frac{\text{Mass of petrol consumed per minute}}{\text{Number of working cycles per minute}}$$

$$= \frac{33/60}{3000/2} = 0.367 \times 10^{-3} \text{kg}$$

$\therefore$ Total volume of petrol at N.T.P.

$$= 0.246 \times 0.367 \times 10^{-3} = 0.0903 \times 10^{-3} \text{ m}^3 \qquad \ldots (ii)$$

We know that specific volume of air (*i.e.* volume of 1 kg of air) at N.T.P. (*i.e.* at $T = 0°$ C or 273 K and $p = 1.013$ bar or 1.013×10^5 N/m^2),

$$= \frac{m R T}{p} = \frac{1 \times 287 \times 273}{1.013 \times 10^5} = 0.773 \text{ m}^3/\text{kg of air}$$

Let m kg of air is admitted per cycle, then total volume of air admitted,

$$v = 0.773 \times m \text{ m}^3 \qquad \ldots (iii)$$

We know that volume of charge admitted per cycle at N.T.P.,

v_a = Volume of petrol per cycle at N.T.P. + Volume of air per cycle at N.T.P.

$$4.16 \times 10^{-3} = 0.0903 \times 10^{-3} + 0.773 \times m$$

or

$$m = 5.265 \times 10^{-3} \text{ kg}$$

$\therefore$ Air-fuel ratio,

$$= \frac{m}{m_1} = \frac{5.265 \times 10^{-3}}{0.367 \times 10^{-3}} = 14.35 \textbf{ Ans.}$$

27.8. Air Consumption

The supply of air to an I.C. engine may be measured experimentally by passing the air through a sharp edged orifice into a large tank (the volume being 500 times the swept volume of the engine). The air is then passed to the engine. It is a cheap and simple method of estimating the air supply to an engine.

The air is drawn into a large tank through an orifice whose diameter and coefficient of discharge are known. The engine now draws air from this tank as shown in Fig. 27.2. The pressure of air in the tank is less than the atmospheric pressure due to the powerful engine suction. Since the tank is relatively large, the air pressure may be assumed to remain constant.

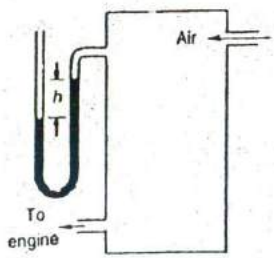


Fig. 27.2. Air consumption.

The outside air is assumed to flow continuously, through the orifice, with a constant velocity. This velocity depends upon the difference of pressure between the air in the tank and the atmospheric air. This pressure difference is measured by a U-tube containing water, whose one limb is connected to the inside of the chamber while the other end is open to the atmosphere. The temperature of atmosphere and barometer reading is also taken.

Let a = Area of orifice in m^2,

C_d = Coefficient of discharge of orifice,

H = Head causing flow of air through orifice in metres of air,

ρ_a = Density of air under atmospheric conditions in kg/m^3,

ρ_w = Density of water in kg/m^3 = 1000 kg/m^3,

h = Pressure difference measured in U-tube in metres of water.

Now head causing flow of air through orifice,

$$H = h \times \frac{\rho_w}{\rho_a} = \frac{1000\,h}{\rho_a} \text{ metres of air} \qquad \ldots (i)$$

We know that velocity of air,

$$V = \sqrt{2gH} \text{ m/s}$$

$\therefore$ Quantity or volume of air passing through the orifice,

$$Q = C_d\, a\, V = C_d\, a\sqrt{2gH} \text{ m}^3\text{/s} \qquad \ldots (ii)$$

We know that mass of air supplied,

$$m_a = \text{Volume} \times \text{Density} = v_a\,\rho_a = Q\,\rho_a$$

$$= C_d\, a\sqrt{2gH} \times \rho_a$$

$$= C_d\, a\sqrt{2 \times 9.81 \times \frac{1000\,h}{\rho_a}} \times \rho_a$$

$$= 140\, C_d\, a\sqrt{h\,\rho_a} \text{ kg/s}$$

Note : The density of air (ρ_a) at pressure p bar and absolute temperature T K may be obtained by applying the characteristic equation of a gas, *i.e.*

$$p = \rho_a RT \quad \text{or} \quad \rho_a = p/RT$$

Example 27.17. *Following readings were obtained during the test on a single cylinder, 4-stroke I.C. engine :*

Engine speed = 300 r.p.m. ; Diameter of orifice of the air tank = 20 mm ; Pressure causing air flow through the orifice = 100 mm of water.

Find the quantity of air consumed per second, if its density under atmospheric conditions is 1.15 kg/m^3. Take coefficient of discharge for the orifice as 0.7.

Solution. Given : *N = 300 r.p.m. ; d = 20 mm = 0.02 m ; h = 100 mm of water = 0.1 m of water ; ρ_a = 1.15 kg/m^3 ; C_d = 0.7

We know that area of orifice,

$$a = \frac{\pi}{4}(0.02)^2 = 0.3142 \times 10^{-3} \text{ m}^2$$

and head causing flow of air through orifice,

$$H = h \times \frac{\rho_w}{\rho_a} = 0.1 \times \frac{1000}{1.15} = 86.96 \text{ m of air} \quad \ldots (\because \rho_w = 1000 \text{ kg/m}^3)$$

$\therefore$ Quantity of air flow,

$$Q = C_d\, a\sqrt{2gH} = 0.7 \times 0.3142 \times 10^{-3}\sqrt{2 \times 9.81 \times 86.96} \text{ m}^3\text{/s}$$

$$= 0.0091 \text{ m}^3\text{/s} \quad \textbf{Ans.}$$

* Superfluous data

27.9. Heat Balance Sheet

The complete record of heat supplied and heat rejected during a certain time (say one minute) by an I.C. engine is entered in a tabulated form known as heat balance sheet. The following values are required to complete the heat balance sheet of an I.C. engine :

1. *Heat supplied by the fuel*

Let m_f = Mass of fuel supplied in kg/min, and

C = Lower calorific value of the fuel in kJ/kg.

We know that the heat in fuel supplied

$$= m_f \times C \text{ kJ/min} \qquad \ldots (i)$$

Note : In case of a gas engine, the volume of the gas supplied is first converted to N.T.P. conditions. It is then multiplied by its lower calorific value to get the heat supplied by the fuel.

Let v_n = Volume of gas supplied in m^3/min at N.T.P., and

C = Lower calorific value of the gas in kJ/m^3 at N.T.P.

∴ Heat supplied by fuel $= v_n \times C$ kJ/min

2. *Heat absorbed in I.P. produced*

We know that the indicated power produced by I.C. engine,

$$\text{I.P.} = \frac{100\, p_m\, L\, A\, n}{60} \text{ kW} \qquad \ldots \text{(For single cylinder engine)}$$

$$\therefore \text{Heat absorbed in I.P./min} = 100\, p_m\, L\, A\, n \text{ kJ/min} \qquad \ldots (\because 1 \text{ kW} = 1 \text{ kJ/s}) \quad \ldots (ii)$$

3. *Heat rejected to the cooling water*

The mass of cooling water, circulating through the cylinder jackets, as well as its inlet and outlet temperatures are measured in order to determine the heat rejected to the cooling water.

Let m_w = Mass of cooling water supplied in kg/min,

c_w = Specific heat of water which may be taken as 4.2 kJ/kg K,

t_1 = Inlet temperature, and

t_2 = Outlet temperature.

Then heat rejected to cooling water

$$= m_w\, c_w\, (t_1 - t_2) \text{ kJ/min} \qquad \ldots (iii)$$

4. *Heat carried away by exhaust gases*

The mass of exhaust gases may be obtained by adding together the mass of fuel supplied and the mass of air supplied. The mass of air supplied may be measured by an orifice or it may be calculated from the analysis of the exhaust gases. The temperature of the exhaust gases is also measured.

Let m_g = Mass of exhaust gases produced in kg/min,

c_g = Specific heat of exhaust gases, and

t = Rise in temperature.

∴ Heat carried away by exhaust gases

$$= m_g\, c_g\, t \text{ kJ/min} \qquad \ldots (iv)$$

5. *Unaccounted heat*

There is always some loss of heat due to friction, leakage, radiation etc. which cannot be determined experimentally. In order to complete the heat balance sheet, this loss is obtained by the difference of heat supplied by the fuel and heat absorbed in I.P., cooling water and exhaust gases.

Finally, the heat balance sheet is prepared as given below :

S.No.	*Particulars*	*Heat in*	
		kJ	*%*
	Total heat supplied	...	100
1.	Heat absorbed in I.P.	...	...
2.	Heat rejected to the cooling water	...	...
3.	Heat carried away by exhaust gases	...	...
4.	Unaccounted heat	...	...

Example 27.18. *An I.C. engine uses 6 kg of fuel having calorific value 44 000 kJ/kg in one hour. The I.P. developed is 18 kW. The temperature of 11.5 kg of cooling water was found to rise through 25° C per minute. The temperature of 4.2 kg of exhaust gas with specific heat 1 kJ/kg K was found to rise through 220° C. Draw the heat balance sheet for the engine.*

Solution. Given : $m_f = 6$ kg/h $= 0.1$ kg/min ; $C = 44\,000$ kJ/kg ; I.P. = 18 kW ; $m_w = 1.5$ kg/min ; $t_2 - t_1 = 25°$ C ; $m_g = 4.2$ kg ; $c_g = 1$ kJ/kg K ; $t = 220°$ C

We know that heat supplied by the fuel

$$= m_f \times C = 0.1 \times 44\,000 = 4400 \text{ kJ/min}$$

Heat absorbed in I.P. produced $= 18 \text{ kW} = 18 \text{ kJ/s} = 1080 \text{ kJ/min}$

Heat rejected to cooling water $= m_w c_w (t_2 - t_1) = 11.5 \times 4.2 \times 25 = 1207.5 \text{ kJ/min}$

Heat lost to exhaust gases $= m_g c_g t = 4.2 \times 1 \times 220 = 924 \text{ kJ/min}$

and unaccounted heat $= 4400 - (1080 + 1207.5 + 924) = 1188.5 \text{ kJ/min}$

Now prepare the heat balance sheet as given below :

S.No.	*Particulars*	*Heat in*	
		kJ	*%*
	Total heat supplied	4400	100
1.	Heat absorbed in I.P.	1080	24.55
2.	Heat rejected to cooling water	1207.5	27.44
3.	Heat carried away by exhaust gases	924	21.00
4.	Unaccounted heat	1188.5	27.01
	Total	4400	100

Example 27.19. *A gas engine, working on four stroke constant volume cycle, gave the following results when loaded by friction brake during a test of an hour's duration :*

Cylinder diameter 240 mm ; Stroke length 480 mm ; Clearance volume 4450×10^{-6} m³ ; Effective circumference of the brake wheel 3.86 m ; Net load on brake 1260 N at overall speed of 226.7 r.p.m. ; Average explosions/min 77 ; m.e.p. of indicator card 7.5 bar ; Gas used 13 m³/h at 15° C and 771 mm of Hg ; Lower calorific value of gas 49 350 kJ/m³ at N.T.P. ; Cooling jacket water

660 kg raised to 34.2° C ; Heat lost to exhaust gases 8%. Calculate : 1. I.P. ; 2. B.P. ; 3. Indicated thermal efficiency ; and 4. Efficiency ratio. Also draw a heat balance sheet for the engine.

Solution. Given : $D_c = 240$ mm = 0.24 m ; $L = 480$ mm = 0.48 m ; $v_c = 4450 \times 10^{-6}$ m^3; $\pi D = 3.86$ m ; $(W - S) = 1260$ N ; $N = 226.7$ r.p.m. ; $n = 77$; $p_m = 7.5$ bar ; $v_1 = 13$ m^3/h ; $T_1 = 15°$ C = 288 K ; $p_1 = 771$ mm of Hg ; $C = 49\ 350$ kJ/m^3 ; $m_w = 660$ kg/h = 11 kg/min ; $t_2 - t_1 = 34.2°$ C ; Heat lost to exhaust gases = 8%

1. *I.P.*

We know that area of the cylinder,

$$A = \frac{\pi}{4}(D_c)^2 = \frac{\pi}{4}(0.24)^2 = 0.045 \text{ m}^2$$

$$\therefore \qquad \text{I.P.} = \frac{100\, p_m L A n}{60} = \frac{100 \times 7.5 \times 0.48 \times 0.045 \times 77}{60} = 20.8 \text{ kW} \quad \textbf{Ans.}$$

2. *B.P.*

$$\text{We know that B.P.} = \frac{(W-S)\,\pi DN}{60} = \frac{1260 \times 3.86 \times 226.7}{60} = 18\ 400 \text{ W}$$

$$= 18.4 \text{ kW} \quad \textbf{Ans.}$$

3. *Indicated thermal efficiency*

First of all, let us find the volume of gas at N.T.P. (*i.e.* temperature 0° C and pressure 760 mm of Hg).

Let v_0 = Volume of gas at N.T.P.

T_0 = Absolute temperature of gas = 273 K

p_0 = Pressure at N.T.P. = 760 mm of Hg

According to the gas equation,

$$\frac{p_0 v_0}{T_0} = \frac{p_1 v_1}{T_1}$$

or

$$v_0 = \frac{p_1 v_1}{T_1} \times \frac{T_0}{p_0} = \frac{771 \times 13}{288} \times \frac{273}{760} = 12.5 \text{ m}^3$$

$\therefore$ Indicated thermal efficiency,

$$\eta_i = \frac{\text{I.P.} \times 3600}{v_0 \times C} = \frac{20.8 \times 3600}{12.5 \times 49\ 350} = 0.121 \quad \text{or} \quad 12.1\% \quad \textbf{Ans.}$$

4. *Efficiency ratio**

First of all, let us find out the air standard efficiency (η_{ase}). We know that swept volume,

$$v_s = \frac{\pi}{4}(D_c)^2 L = \frac{\pi}{4}(0.24)^2\, 0.48 = 0.0217 \text{ m}^3$$

and clearance volume, $v_c = 4450 \times 10^{-6}$ m^3

$\therefore$ Total volume $= v_s + v_c = 0.0217 + 4450 \times 10^{-6} = 0.026\ 15$ m^3

* It is also known as relative efficiency.

and compression ratio, $r = \dfrac{\text{Total volume}}{\text{Clearance volume}} = \dfrac{0.026\,15}{4450 \times 10^{-6}} = 5.876$

We know that air standard efficiency,

$$\eta_{ase} = 1 - \frac{1}{r^{\gamma - 1}} = 1 - \frac{1}{(5.876)^{1.4-1}} = 1 - 0.492 = 0.518$$

... (For air, $\gamma = 1.4$)

$\therefore$ Efficiency ratio $= \dfrac{\text{Indicated thermal efficiency}}{\text{Air standard efficiency}} = \dfrac{0.121}{0.518} = 0.234$ or 23.4% **Ans.**

Heat balance sheet

We know that heat supplied by the fuel

$= \dot{v}_0 \times C$ ... ($\dot{v}_0$ = Vol. of gas used for one hour at N.T.P.)

$$= \frac{12.5 \times 49\,350}{60} = 10\,280 \text{ kJ/min}$$

Heat absorbed in I.P. produced

$= 20.8 \text{ kW} = 20.8 \text{ kJ/s} = 1248 \text{ kJ/min}$

Heat rejected to cooling water

$= m_w\, c_w\, (t_2 - t_1) = 11 \times 4.2 \times 34.2 = 1580 \text{ kJ/min}$

Heat lost to exhaust gases (8% given)

$= 0.08 \times 10\,280 = 822.4 \text{ kJ/min}$

and unaccounted heat $= 10\,280 - (1248 + 1580 + 822.4) = 6629.6 \text{ kJ/min}$

Now prepare the heat balance sheet as given below :

S.No.	Particulars	Heat in	
		kJ	%
	Total heat supplied	10 280	100
1.	Heat absorbed in I.P. produced	1248	12.14
2.	Heat rejected to cooling water	1580	15.36
3.	Heat lost to exhaust gases	822.4	8.00
4.	Unaccounted heat	6629.6	64.50
	Total	10 280	100

Example 27.20. *A six cylinder, four stroke diesel engine has bore 360 mm and stroke 500 mm. A trial on the engine provided the following data :*

Mean area of indicator diagram = 780 mm^2 ; Length of the indicator diagram = 75 mm ; Spring number = 0.7 bar per mm of compression ; Brake torque = 14 000 N-m ; Speed = 500 r.p.m. ; Fuel consumption = 240 kg/h ; Calorific value of fuel oil = 44 000 kJ/kg ; Jacket cooling water = 320 kg/min ; Rise in temperature of cooling water = 40° C ; Piston cooling oil = 140 kg/min ; Specific heat = 2.1 kJ/kg K ; Temperature rise of oil = 28° C ; Circulating water in gas calorimeter = 300 kg/min ; Rise in temperature of this water = 42° C.

All heat of the exhaust gases is absorbed in the calorimeter. Estimate the specific fuel consumption and mechanical efficiency of the engine. Draw up a heat balance sheet of the engine on 1 kg of fuel oil basis.

Solution. Given : $K = 6$; $D_c = 360$ mm $= 0.36$ m ; $L = 500$ mm $= 0.5$ m ; $a = 780$ mm^2 ; $l = 75$ mm ; $s = 0.7$ bar/mm ; $T = 14\ 000$ N-m ; $N = 500$ r.p.m. ; $m_f = 240$ kg / h $= 4$ kg / min ; $C = 44\ 000$ kJ/kg ; $m_w = 320$ kg/min ; $t_2 - t_1 = 40°$ C ; $m_a = 140$ kg/min ; $c_a = 2.1$ kJ/kg K ; $t_a = 28°$ C ; $m_e = 300$ kg/min ; $t_e = 42°$ C

Specific fuel consumption

We know that indicated mean effective pressure,

$$p_m = \frac{a \times s}{l} = \frac{780 \times 0.7}{75} = 7.28 \text{ bar}$$

Area of cylinder, $$A = \frac{\pi}{4}(D_c)^2 = \frac{\pi}{4}(0.36)^2 = 0.102 \text{ m}^2$$

and number of working strokes per minute,

$$n = N/2 = 500/2 = 250 \quad \ldots (\because \text{Engine works on four stroke cycle})$$

We know that indicated power,

$$\text{I.P.} = \frac{100\, K\, p_m\, L\, A\, n}{60} = \frac{100 \times 6 \times 7.28 \times 0.5 \times 0.102 \times 250}{60} \text{ kW}$$

$$= 928 \text{ kW}$$

$\therefore$ Specific fuel consumption

$$= \frac{m_f}{\text{I.P.}} = \frac{240}{928} = 0.258 \text{ kg / kWh } \textbf{Ans.}$$

Mechanical efficiency

We know that brake power,

$$\text{B.P.} = \frac{T \times 2\pi N}{60} = \frac{14\ 000 \times 2\pi \times 500}{60} = 733\ 000 \text{ W} = 733 \text{ kW}$$

and mechanical efficiency, $$\eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{733}{928} = 0.79 \text{ or } 79\% \textbf{ Ans.}$$

Heat balance sheet for 1 kg of fuel oil

We know that heat supplied by the fuel

$$= 44\ 000 \text{ kJ/kg of fuel}$$

Since the fuel consumption is 240 kg / h, therefore time for 1 kg of fuel consumption

$$= \frac{1}{240}\text{ h} = \frac{3600}{240} = 15 \text{ s}$$

Heat absorbed in I.P. produced

$$= 928 \text{ kW} = 928 \text{ kJ/s}$$

$$= 928 \times 15 = 13\ 920 \text{ kJ/kg of fuel}$$

$$\ldots (\because 1 \text{ kg of fuel consumption takes } 15 \text{ s})$$

Heat rejected to cooling water

$$= m_w \times c_w (t_2 - t_1) = 320 \times 4.2 \times 40 = 53\ 760 \text{ kJ/min} = 896 \text{ kJ/s}$$

$$= 896 \times 15 = 13\ 440 \text{ kJ/kg of fuel}$$

Heat lost to piston cooling oil

$$= m_o c_o t_o = 140 \times 2.1 \times 28 = 8232 \text{ kJ/min} = 137.2 \text{ kJ/s}$$

$$= 137.2 \times 15 = 2058 \text{ kJ/kg of fuel}$$

Heat lost to water in calorimeter

$$= m_c c_c t_c = 300 \times 4.2 \times 42 = 52\,920 \text{ kJ/min} = 882 \text{ kJ/s}$$

$$= 882 \times 15 = 13\,230 \text{ kJ/kg of fuel}$$

Unaccounted heat $= 44\,000 - (13\,920 + 13\,440 + 2058 + 13\,230) = 1352$ kJ/kg of fuel

Now prepare the heat balance sheel on 1 kg of fuel basis, as given below :

S.No.	*Particulars*	*Heat in*	
		kJ	*%*
	Total heat supplied	44 000	100
1.	Heat absorbed in I.P. produced	13 920	31.64
2.	Heat rejected to cooling water	13 440	30.54
3.	Heat lost to piston cooling oil	2058	4.68
4.	Heat lost to water in calorimeter	13 230	30.07
5.	Unaccounted heat (by difference)	1352	3.07
	Total	44 000	100

EXERCISES

1. The following data were recorded during testing of a four stroke cycle gas engine :

Area of indicator diagram = 900 mm^2 ; Length of indicator diagram = 70 mm ; Spring scale = 0.3 bar/mm ; Diameter of piston = 200 mm ; Length of stroke = 250 mm ; Speed = 300 r.p.m. Determine :

1. Indicated mean effective pressure ; and 2. Indicated power. **[Ans. 3.86 bar ; 7.58 kW]**

2. A two stroke cycle internal combustion engine has a mean effective pressure of 6 bar. The speed of the engine is 1000 r.p.m, If the diameter of piston and stroke are 110 mm and 140 mm respectively, find the indicated power. **[Ans. 13.3 kW]**

3. A gas engine has a piston diameter of 150 mm and stroke 250 mm. The speed of the engine is 250 r.p.m. and the average number of explosions are 90 per minute. The mean effective pressure is 7 bar. If the average torque on the brake is 140 N-m, find indicated power, brake power and mechanical efficiency. **[Ans. 4.65 kW ; 3.67 kW ; 78.8%]**

4. During a trial of a single cylinder four stroke I.C. engine, the following observations were recorded:

Mean effective pressure = 4 bar ; Speed = 200 r.p.m. ; Brake power = 7.5 kW ; Length of stroke = 1.5 times diameter of piston.

If the mechanical efficiency is 70% ; find the dimensions of the engine. **[Ans. 240 mm ; 360 mm]**

5. A constant speed four stroke cycle compression-ignition engine has a bore of 100 mm, stroke 150 mm and runs at 450 r.p.m. The following data refer to a test on this engine:

Brake wheel diameter = 600 mm ; Band thickness = 5 mm ; Load on band = 210 N ; Spring balance reading = 30 N ; Area of indicator diagram = 415 mm^2 ; Length of indicator diagram = 62.5 mm ; Spring scale = 1.1 bar per mm ; Specific fuel consumption. = 0.3 kg /b.p./h ; Calorific value of fuel = 42 000 kJ/kg.

Calculate : 1. the mechanical efficiency, and 2. the indicated thermal efficiency. **[Ans. 80.3% ; 35.6%]**

6. The following data refer to a test on a petrol engine:

Indicated power = 30 kW ; Brake power = 26 kW ; Engine speed = 1800 r.p.m. ; Fuel per brake power hour = 0.35 kg ; Calorific value of the fuel used = 44 100 kJ/kg.

Calculate : 1. the mechanical efficiency ; 2. the indicated thermal efficiency ; and 3. the brake thermal efficiency. [Ans. 26.9% ; 23.3%]

7. Following observations were taken during the trial of a single cylinder, four stroke, oil engine, running at full load :

Area of indicator diagram = 300 mm^2 ; Length of the diagram = 40 mm ; Spring stiffness = 1 bar/mm ; Speed of the engine = 400 r.p.m ; Brake load = 400 N ; Spring balance reading = 50 N ; Diameter of the brake drum = 1.2 m ; Fuel consumption per hour = 3 kg ; Calorific value of fuel = 42 000 kJ/kg ; Cylinder diameter = 160 mm ; Stroke = 200 mm.

Find the indicated power, brake power, mechanical efficiency and brake thermal efficiency.

[Ans. 10 kW ; 8.8 kW ; 88% ; 25.14%]

8. The output of an I.C engine is measured by a rope brake dynamometer. The diameter of the brake pulley is 750 mm and rope diameter is 50 mm. The dead load on the tight side of the rope is 410 N and the spring balance reading is 50 N. The engine consumes 4 kg/h of fuel at rated speed of 1000 r.p.m. The calorific value of fuel is 44 100 kJ/kg. Calculate brake specific fuel consumption and the brake thermal efficiency.

[Ans. 0.265 kg/B.P./h ; 30.8%]

9. A compression ignition engine at rated condition develops 7.5 kW brake power. The mechanical losses are 1.5 kW. If the indicated thermal efficiency is 42% ; air fuel ratio 22 and calorific value of fuel 43 260 kJ/kg, determine : 1. fuel consumption is kg/h ; 2 air intake in kg/h ; and 3. brake thermal efficiency.

[Ans. 1.783 kg/h ; 39.23 kg/h ; 35%]

10. A four cylinder, two stroke cycle petrol engine develops 30 kW brake power at 2500 r.p.m. The mean effective pressure on each piston is 8 bar and the mechanical efficiency is 80%. Calculate the diameter and stroke of each cylinder if the stroke to bore ratio is 1.5. Also calculate the brake specific fuel consumption of the engine, if brake thermal efficiency is 28%. The calorific value of the fuel is 44 100 kJ/kg.

[Ans. 62 mm ; 93 mm ; 0.29 kg/B.P./h]

11. An engine is used on a job requiring 110 kW B.P., the mechanical efficiency of the engine is 80 percent and the engine uses 50 kg fuel per hour under the conditions of operation. A design improvement is made which reduces the engine friction by 5 kW. Assuming the indicated thermal efficiency remains the same, how many kg of fuel per hour will be saved ? [Ans. 1.8 kg/h]

12. The following data relates to a four cylinder four stroke petrol engine :

Diameter of the piston = 80 mm ; Length of the stroke = 120 mm ; Clearance volume = 100×10^3 mm^3 ; Fuel supply = 4.8 kg/h ; Calorific value = 44 100 kJ/kg.

When the Morse test was performed on the engine, the following data were obtained :

B.P. with all the cylinders working	= 14.5 kW
B.P. with cylinder 1 cut-off	= 9.8 kW
B.P. with cylinder 2 cut-off	= 10.3 kW
B.P. with cylinder 3 cut-off	= 10.14 kW
B.P. with cylinder 4 cut-off	= 10 kW

Find I.P. of the engine and also calculate indicated thermal efficiency, brake thermal efficiency and relative efficiency. [Ans. 17.76 kW ; 30.2% ; 24.66% ; 55.7%]

13. A petrol engine uses per brake power hour 0.36 kg of fuel of calorific value 44 100 kJ/kg. The mechanical efficiency is 78 percent and compression ratio is 5.6. Calculate : 1. Brake thermal efficiency ; 2. Indicated thermal efficiency ; and 3. Ideal air standard efficiency. Take $\gamma = 1.4$. [Ans. 22.7% ; 29.1% ; 49.8%]

14. A four cylinder four-stroke petrol engine produces 56 kW indicated power when running at 4400 r.p.m. with a volumetric efficiency of 81.5%. The air-fuel ratio is 16 : 1 and the thermal efficiency is 35%. The fuel used has a calorific value of 44 100 kJ/kg. If the bore to stroke ratio is 1 : 1.04, calculate the cylinder dimensions. Assume the charge to have the density of air equal to 1.293 kg/m^3 at N.T.P.

[Ans. 77.2 mm ; 80.3 mm]

15. A six cylinder, four stroke S.I. engine, having a piston displacement of 700×10^{-6} m^3 per cylinder developed 78 kW at 3200 r.p.m. and consumed 27 kg of petrol per hour. The calorific value of petrol is 44 MJ/kg.

Estimate 1. the volumetric efficiency of the engine if the air fuel ratio is 12 and the intake air is at 0.9 bar, and 32° C ; 2. the brake thermal efficiency ; and 3. the brake torque. For air, $R = 0.287$ kJ/kg K.

[Ans. 78.15% ; 23.64% ; 232.7 N-m]

16. Calculate the bore and stroke of a four stroke single cylinder oil engine designed to the following particulars.

Brake power 18 kW at 250 r.p.m. when running on oil having composition by mass C 85%, H 15% and a lower calorific value of 42 000 kJ/kg. The oil is burnt with 25% excess air. The volumetric efficiency reckoned on atmospheric conditions of 1.013 bar and 10° C is 0.8. The mechanical efficiency is 0.9 and indicated thermal efficiency is 0.35. Take $R = 0.287$ kJ/kg K and bore-stroke ratio as 1 : 1.2. **[Ans. 236 mm ; 283.2 mm]**

17. In a test of one hour duration on a single cylinder oil engine performing on a four stroke cycle, 8.08 kg of oil of calorific value 42 000 kJ/kg were used. The jacket water was 658 kg and its temperature rise is 22° C. The average speed was 200 r.p.m. and the m.e.p. in the cylinder is 5.95 bar. The cylinder diameter is 300 mm ; stroke 450 mm, and brake friction load 1900 N applied at the periphery of a flywheel of 1.2 m diameter. Show by heat balance chart, how the heat supplied is apportioned between the several items concerned and estimate the brake thermal efficiency and mechanical efficiency of the engine. **[Ans. 25.3% ; 75.8%]**

18. Calculate the brake specific fuel consumption, indicated thermal efficiency and obtain a heat balance sheet on minute basis from the following test data obtained in a four stroke two cylinder diesel engine :

Duration of test = 1 hour ; Brake power = 15 kW ; Total indicated power = 17.8 kW ; Fuel consumption = 4.24 litres of specific gravity 0.875 ; Lower calorific value of fuel = 43 340 kJ/kg ; Jacket cooling water circulated = 215 kg ; Inlet and outlet cooling water temperature = 30° C and 80° C.

The heat in exhaust gases is measured by an exhaust gas calorimeter as 808 kJ/minute.

[Ans. 0.247 kg/B.P./h ; 39.8%]

19. Draw a heat balance sheet for a two stroke diesel engine run for 20 minutes at full load, from the data given below :

R.P.M. = 350 ; M.E.P. = 3 bar ; Net brake load = 650 N ; Fuel consumption = 1.5 kg ; Cooling water = 160 kg ; Water inlet temperature = 35° C ; Water outlet temperature = 60° C ; Air used per kg of fuel = 30 kg ; Room temperature = 20° C ; Exhaust temperature = 300° C ; Cylinder bore = 200 mm ; Cylinder stroke = 280 mm ; Brake diameter = 1 m ; Calorific value of fuel = 44 100 kJ/kg ; Steam formed per kg of fuel in the exhaust = 1.35 kg ; Specific heat of steam in exhaust = 2.1 kJ/kg K ; Specific heat of dry exhaust gas = 1.008 kJ/kg K.

20. The following particulars refer to the full load test of a single cylinder, petrol engine working on the four stroke cycle :

Speed = 2500 r.p.m. ; Brake power = 118 kW ; Cylinder bore = 110 mm ; Cylinder diameter = 120 mm ; Lower calorific value of fuel = 41150 kJ/kg ; Petrol consumption = 40 kg/h ; Jacket water rate = 2800 kg/h ; Jacket water inlet temperature = 20° C ; Jacket water outlet temperature = 65.5° C ; Fuel-air ratio = 1 : 16 ; Room temperature = 29° C ; Exhaust temperature = 399° C ; Hydrogen in fuel = 15% by mass ; Sp. heat of dry exhaust gases = 0.9945 kJ/kg K ; Sp. heat of water vapour = 1.838 kJ/kg K.

Draw up a heat balance sheet. Calculate the brake thermal efficiency and volumetric efficiency of the engine. **[Ans. 25.8% ; 73.7%]**

QUESTIONS

1. What is the purpose of engine testing ?
2. Name the various measurements which are to be taken in a test of an I.C. engine ?
3. Describe the method of measuring mean effective pressure of an I.C. engine.
4. Explain the method for determining the indicated power of a multi-cylinder engine without using an indicator.
5. Define volumetric efficiency for an I.C. engine. What is the effect of volumetric efficiency on (*i*) engine power ; and (*ii*) specific fuel consumption.
6. What is the use of heat balance sheet of an engine ? Mention the various items to be determined to complete the heat balance sheet.

OBJECTIVE TYPE QUESTIONS

1. The power actually developed by the engine cylinder of an I.C. engine is known as

(*a*) brake power (*b*) indicated power (*c*) actual power

2. The number of working strokes per minute for a four stroke cycle engine are....the speed of the engine in r.p.m.

(*a*) equal to (*b*) one-half (*c*) twice (*d*) four times

3. If the speed of the engine is increased, the indicated power will

(*a*) increase (*b*) decrease (*c*) remain same

4. The brake power of the engine is the power available

(*a*) at the crank pin (*b*) in the engine cylinder

(*c*) at the crankshaft (*d*) none of these

5. The brake power of an engine is always.....the indicated power.

(*a*) equal to (*b*) less than (*c*) greater than

6. The ratio of the indicated thermal efficiency to the air standard efficiency is called

(*a*) mechanical efficiency (*b*) overall efficiency

(*c*) volumetric efficiency (*d*) relative efficiency

7. The ratio of the volume of charge admitted at N.T.P. to the swept volume of the piston is called

(*a*) mechanical efficiency (*b*) overall efficiency

(*c*) volumetric efficiency (*d*) relative efficiency

8. The thermal efficiency of petrol engines is about

(*a*) 15% (*b*) 30% (*c*) 50% (*d*) 70%

9. The volumetric efficiency of a well designed engine may be

(*a*) 30 to 40% (*b*) 40 to 60% (*c*) 60 to 70% (*d*) 75 to 90%

10. The Morse test is used to find the indicated power of a

(*a*) single cylinder petrol engine (*b*) single cylinder diesel engine

(*c*) multi-cylinder engine (*d*) none of these

ANSWERS

1. (*b*)	2. (*b*)	3. (*a*)	4. (*c*)	5. (*b*)
6. (*d*)	7. (*c*)	8. (*b*)	9. (*d*)	10. (*c*)

28

Reciprocating Air Compressors

28.1. Introduction

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

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

28.2. Classification of Air Compressors

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

1. *According to working*

(*a*) **Reciprocating compressors, and** (*b*) **Rotary compressors.**

2. *According to action*

(*a*) **Single acting compressors, and** (*b*) **Double acting compressors.**

3. *According to number of stages*

(*a*) **Single stage compressors, and** (*b*) **Multi-stage compressors.**

28.3. Technical Terms

The following technical terms, which will be frequently used in this chapter, should be clearly understood at this stage :

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

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

3. *Compression ratio* (**or pressure ratio**). It is the ratio of discharge pressure to the inlet pressure. Since the discharge pressure is always more than the inlet pressure, therefore the value of compression ratio is more than unity.

4. *Compressor capacity.* It is the volume of air delivered by the compressor, and is expressed in m^3/min or m^3/s.

5. *Free air delivery.* It is the actual volume delivered by a compressor when reduced to the normal temperature and pressure condition. The capacity of a compressor is generally given in terms of free air delivery.

6. *Swept volume.* It is the volume of air sucked by the compressor during its suction stroke. Mathematically, the swept volume or displacement of a single acting air compressor is given by

$$v_s = \frac{\pi}{4} \times D^2 \times L$$

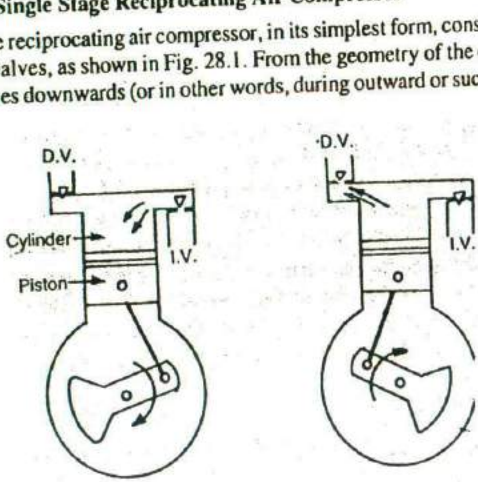
where D = Diameter of cylinder bore, and

L = Length of piston stroke.

7. *Mean effective pressure.* As a matter of fact, air pressure on the compressor piston keeps on changing with the movement of the piston in the cylinder. The mean effective pressure of the compressor is found out mathematically by dividing the work done per cycle to the stroke volume.

28.4. Working of Single Stage Reciprocating Air Compressor

A single stage reciprocating air compressor, in its simplest form, consists of a cylinder, piston, inlet and discharge valves, as shown in Fig. 28.1. From the geometry of the compressor, we find that when the piston moves downwards (or in other words, during outward or suction stroke), the pressure



(*a*) Suction stroke. (*b*) Delivery stroke.

Fig. 28.1. Single stage reciprocating air compressor.

inside the cylinder falls below the atmospheric pressure. Due to this pressure difference, the inlet valve (I.V.) gets opened and air is sucked into the cylinder, at inlet pressure until the piston completes the outward stroke. Now when the piston moves upwards (or in other words, during inward or delivery stroke), the pressure inside the cylinder goes on increasing till it reaches the discharge pressure. At this stage, the discharge valve (D.V.) gets opened and air is delivered to the container. At the end of delivery stroke, a small quantity of air, at high pressure, is left in the clearance space. As the piston starts its suction stroke, the air contained in the clearance space expands till its pressure falls below the atmospheric pressure. At this stage, the inlet valve gets opened as a result of which fresh air is sucked into the cylinder, and the cycle is repeated.

It may be noted that in a single acting reciprocating air compressor, the suction, compression and delivery of air takes place in two strokes of the piston or one revolution of the crankshaft.

Note : In a double acting reciprocating compressor, the suction, compression and delivery of air takes place on both sides of the piston. It is thus obvious, that such a compressor will supply double the volume of air than a single acting reciprocating compressor (neglecting volume of piston rod).

28.5. Workdone by a Single Stage Reciprocating Air Compressor

We have already discussed that in a reciprocating air compressor, the air is first sucked, compressed and then delivered. So there are three different operations of the compressor. Thus we see that work is done on the piston during the suction of the air. Similarly, work is done by the piston during compression as well as delivery of the air. A little consideration will show, that the work done by a reciprocating air compressor is mathematically equal to the work done by the compressor during suction. Here we shall discuss the following two important cases of work done :

1. when there is no clearance volume in the cylinder, and

2. when there is some clearance volume.

28.6. Workdone by a Single Stage Reciprocating Air Compressor without Clearance Volume

Consider a single stage reciprocating air compressor without clearance volume delivering air from one side of the piston only.

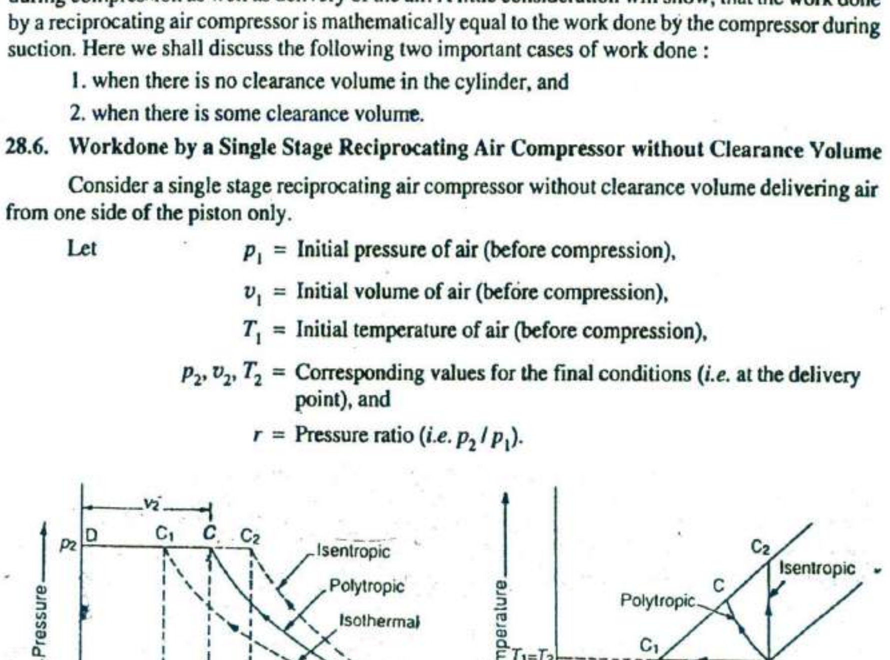
Let p_1 = Initial pressure of air (before compression),

v_1 = Initial volume of air (before compression),

T_1 = Initial temperature of air (before compression),

p_2, v_2, T_2 = Corresponding values for the final conditions (*i.e.* at the delivery point), and

r = Pressure ratio (*i.e.* p_2 / p_1).



(*a*) *p-v* diagram without clearance volume. (*b*) *T-s* diagram.

Fig. 28.2. *p-v* and *T-s* diagrams for a single stage reciprocating air compressor.

The *p-v* and *T-s* diagrams of a single acting single stage reciprocating air compressor without clearance volume is shown in Fig. 28.2. We know that during return stroke, the air is compressed by its major part (*i.e.* compression stroke *BC*) at constant temperature. The compression continues till the pressure (p_2) in the cylinder is sufficient to force open the delivery valve at *C*. After that no more compression takes place with the inward movement of the piston. Now during the remaining part of compression stroke, the compressed air is delivered till the piston head reaches the cylinder end. After that, the air is sucked from the atmosphere during the suction stroke AB at pressure p_1.

As a matter fact, the compression of air may be isothermal, polytropic or isentropic (reversible adiabatic). Now, we shall find out the amount of work done in compressing the air in all the above mentioned three cases.

1. *Work done during isothermal compression*

The isothermal compression and delivery of air is shown by the graphs BC_1 and C_1D respectively. Now C_1D represents the volume of air delivered. We know that work done by the compressor per cycle,

$$W = \text{Area } ABC_1D$$

$$= \text{Area } A'DC_1C_1' + \text{Area } C_1BB'C_1' - \text{Area } A'ABB'$$

$$= p_2v_2 + 2.3\, p_2v_2 \log\left(\frac{v_1}{v_2}\right) - p_1v_1$$

$$= 2.3\, p_1v_1 \log\left(\frac{v_1}{v_2}\right) = 2.3\, p_1v_1 \log\left(\frac{p_2}{p_1}\right) \qquad \ldots (\because p_1v_1 = p_2v_2)$$

$$= 2.3\, p_1v_1 \log r = 2.3\, m\,R\,T_1 \log r \qquad \ldots (\because p_1v_1 = m\,R\,T_1)$$

2. *Work done during polytropic compression (pv^n = Constant)*

The polytropic compression is shown by the line BC in Fig. 28.2. Now CD represents the volume of air delivered, *i.e.* v_2. We know that work done on the air per cycle,

$$W = \text{Area } ABCD$$

$$= \text{Area } A'DCC' + \text{Area } CBB'C' - \text{Area } A'ABB'$$

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

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

$$= \frac{n}{n-1}\,(p_2v_2 - p_1v_1) \qquad \ldots (i)$$

$$= \frac{n}{n-1} \times p_1v_1\left(\frac{p_2v_2}{p_1\,v_1} - 1\right) \qquad \ldots (ii)$$

We also know that for polytropic compression,

$$p_1v_1^n = p_2v_2^n$$

$$\therefore \qquad \frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} \text{ or } \frac{v_1}{v_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

Substituting the value of v_2 / v_1 in equation (*ii*),

$$W = \frac{n}{n-1} \times p_1v_1\left[\frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} - 1\right]$$

$$= \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{n}{n-1} \times m\,R\,T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \qquad \ldots (iii)$$

The equation (i) may also be written as :

$$W = \frac{n}{n-1} \times p_2 v_2 \left(1 - \frac{p_1 v_1}{p_2 v_2} \right)$$

$$= \frac{n}{n-1} \times p_2 v_2 \left[1 - \frac{p_1}{p_2} \left(\frac{p_2}{p_1} \right)^{\frac{1}{n}} \right]$$

$$= \frac{n}{n-1} \times p_2 v_2 \left[1 - \left(\frac{p_1}{p_2} \right)^{\frac{n-1}{n}} \right]$$

$$= \frac{n}{n-1} \times m\,R\,T_2 \left(1 - \frac{T_1}{T_2} \right) = \frac{n}{n-1} \times m\,R\,(T_2 - T_1) \qquad \ldots (iv)$$

3. *Work done during isentropic compression*

The isentropic compression is shown by the curve BC_2 in Fig. 28.2. In this case, the volume of air delivered v_2 is represented by the line C_2D.

The work done on the air per cycle during isentropic compression may be worked out in the similar way as polytropic compression. The polytropic index n is changed to isentropic index γ in the previous results.

$\therefore$ Work done on the air per cycle,

$$W = \frac{\gamma}{\gamma - 1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$= \frac{\gamma}{\gamma - 1} \times m\,R\,T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

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

We know that the ratio of specific heats,

$$\frac{c_p}{c_v} = \gamma; \text{ and } c_p - c_v = R$$

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

Now work done, $$W = \frac{\gamma}{\gamma - 1} \times m\,R\,(T_2 - T_1)$$

$$= \frac{\gamma}{\gamma - 1} \times m\,c_p \left(\frac{\gamma - 1}{\gamma} \right) (T_2 - T_1) = m\,c_p\,(T_2 - T_1)$$

We see that the work done on the air during isentropic compression is equal to the heat required to raise the temperature of air from T_1 to T_2 at a constant pressure.

Note : The work done on the air is minimum when the compression is isothermal (*i.e.* when $n = 1$) and it is maximum when the compression is isentropic (*i.e.* when $n = \gamma$) because isothermal line has less slope than isentropic line. It may be noted that in order to perform isothermal process, the compression should be very slow so that the temperature is maintained constant, which is not possible in actual practice. However, the isothermal compression may be approached, if

1. the air or water cooling is done during the compression,
2. the cold water is sprayed (injected) in the cylinder during the compression, and
3. in multi-stage compressors, intercooling is done.

28.7. Power Required to Drive a Single-stage Reciprocating Air Compressor

We have already obtained in the last article the expressions for the work done (W) per cycle during isothermal, polytropic and isentropic compression. The power required to drive the compressor may be obtained from the usual relation,

$$P = \frac{W N_w}{60} \text{ watts}$$

If N is the speed of the compressor in r.p.m., then number of working strokes per minute,

$$N_w = N \quad \text{... (For single acting compressor)}$$

$$= 2N \quad \text{... (For double acting compressor)}$$

Note : Since the compression takes in three different ways, therefore power obtained from different works done will be different. In general, following are the three values of power obtained :

1. Isothermal power $= \dfrac{W \text{ (in isothermal compression) } N_w}{60}$ watts
2. Isentropic power $= \dfrac{W \text{ (in isentropic compression) } N_w}{60}$ watts
3. Indicated power $= \dfrac{W \text{ (in polytropic compression) } N_w}{60}$ watts

The indicated power is also known as *air power* of the compressor.

Example 28.1. *A single stage reciprocating air compressor is required to compress 1 kg of air from 1 bar to 4 bar. The initial temperature is 27° C. Compare the work requirement in the following cases :*

1. Isothermal compression ; 2. Compression with $p v^{1.2}$ = constant ; and 3. Isentropic compression.

Solution. Given : $m = 1$ kg ; $p_1 = 1$ bar ; $p_2 = 4$ bar ; $T_1 = 27°\text{C} = 27 + 273 = 300$ K ; $n = 1.2$

1. *Work required for isothermal compression*

We know that work required by the compressor,

$$W = 2.3\, p_1 v_1 \log\left(\frac{p_2}{p_1}\right) = 2.3\, m R T_1 \log\left(\frac{p_2}{p_1}\right) \quad \ldots (\because p_1 v_1 = m R T_1)$$

$$= 2.3 \times 1 \times 287 \times 300 \log\left(\frac{4}{1}\right) = 119\,230 \text{ J} = 119.23 \text{ kJ} \quad \textbf{Ans.}$$

$$\ldots (\because R \text{ for air} = 287 \text{ J/kg K})$$

2. *Work required for polytropic compression (i.e. $pv^{1.2}$ = constant)*

We know that work required by the compressor,

$$W = \frac{n}{n-1} \times m\,R\,T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.2}{1.2-1} \times 1 \times 287 \times 300 \left[\left(\frac{4}{1} \right)^{\frac{1.2-1}{1.2}} - 1 \right] = 134\,320 \text{ J}$$

$$= 134.32 \text{ kJ } \textbf{Ans.}$$

3. *Work required for isentropic compression*

We know that work required by the compressor,

$$W = \frac{\gamma}{\gamma-1} \times m\,R\,T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$= \frac{1.4}{1.4-1} \times 1 \times 287 \times 300 \left[\left(\frac{4}{1} \right)^{\frac{1.4-1}{1.4}} - 1 \right] = 146\,630 \text{ J}$$

$$= 146.63 \text{ kJ } \textbf{Ans.}$$

Example 28.2. *Determine the size of the cylinder for a double acting air compressor of 40 kW indicated power, in which air is drawn in at 1 bar and 15° C and compressed according to the law $p\,v^{1.2}$ = constant, to 6 bar. The compressor runs at 100 r.p.m. with average piston speed of 152.5 m/min. Neglect clearance.*

Solution. Given : I.P. = 40 kW = 40×10^3 W ; p_1 = 1 bar = 1×10^5 N/m² ; T_1 = 15° C = 15 + 273 = 288 K ; n = 1.2 ; p_2 = 6 bar ; N = 100 r.p.m. ; Average piston speed = 152.5 m/min

Let D = Diameter of the cylinder in metres, and

L = Length of the stroke in metres.

We know that average piston speed,

$$2\,LN = 152.5$$

$$\therefore \quad L = 152.5 / 2N = 152.5 / 2 \times 100 = 0.7625 \text{ m } \textbf{Ans.}$$

Volume of air before compression,

$$v_1 = \frac{\pi}{4} \times D^2 L = \frac{\pi}{4} \times D^2 \times 0.7625 = 0.6\,D^2 \text{ m}^3$$

and workdone by the compressor,

$$W = \frac{n}{n-1} \times p_1\,v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.2}{1.2-1} \times 1 \times 10^5 \times 0.6\,D^2 \left[\left(\frac{6}{1} \right)^{\frac{1.2-1}{1.2}} - 1 \right] \text{N-m}$$

$$= 125\,310\,D^2 \text{ N-m}$$

Since the compressor is double acting, therefore number of working strokes per minute,

$$N_w = 2\,N = 2 \times 100 = 200$$

We know that indicated power (I.P.),

$$40 \times 10^3 = \frac{W \times N_w}{60} = \frac{125\,310\,D^2 \times 200}{60} = 417.7 \times 10^3\,D^2$$

$$\therefore \quad D^2 = 0.096 \text{ or } D = 0.31 \text{ m or } 310 \text{ mm} \quad \textbf{Ans.}$$

Example 28.3. *A single acting reciprocating air compressor has cylinder diameter and stroke of 200 mm and 300 mm respectively. The compressor sucks air at 1 bar and 27° C and delivers at 8 bar while running at 100 r.p.m. Find : 1. Indicated power of the compressor ; 2. Mass of air delivered by the compressor per minute ; and 3. Temperature of the air delivered by the compressor. The compression follows the law $pv^{1.25} = C$. Take R as 287 J/kg K.*

Solution. Given : $D = 200$ mm $= 0.2$ m ; $L = 300$ mm $= 0.3$ m ; $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2 ; $T_1 = 27°$ C $= 27 + 273 = 300$ K ; $p_2 = 8$ bar ; $N = 100$ r.p.m. ; $n = 1.25$; $R = 287$ J/kg K

We know that volume of air before compression,

$$v_1 = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4}(0.2)^2\,0.3 = 0.0094 \text{ m}^3$$

1. *Indicated power of the compressor*

We know that workdone by the compressor for polytropic compression of air,

$$W = \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.25}{1.25 - 1} \times 1 \times 10^5 \times 0.0094 \left[\left(\frac{8}{1} \right)^{\frac{1.25-1}{1.25}} - 1 \right] \text{N-m}$$

$$= 4700\,(1.516 - 1) = 2425 \text{ N-m}$$

Since the compressor is single acting, therefore number of working strokes per minute,

$$N_w = N = 100$$

$\therefore$ Indicated power of the compressor

$$= \frac{W \times N_w}{60} = \frac{2425 \times 100}{60} = 4042 \text{ W} = 4.042 \text{ kW} \quad \textbf{Ans.}$$

2. *Mass of air delivered by the compressor per minute*

Let m = Mass of air delivered by the compressor per stroke.

We know that $p_1 v_1 = m\,R\,T_1$

$$\therefore \quad m = \frac{p_1 v_1}{R\,T_1} = \frac{1 \times 10^5 \times 0.0094}{287 \times 300} = 0.0109 \text{ kg per stroke}$$

and mass delivered minute $= m \times N_w = 0.0109 \times 100 = 1.09$ kg **Ans.**

3. *Temperature of air delivered by the compressor*

Let T_2 = Temperature of air delivered by the compressor.

We know that

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} = \left(\frac{8}{1} \right)^{\frac{1.25-1}{1.25}} = 8^{0.2} = 1.516$$

$$\therefore \quad T_2 = 1.516 \times T_1 = 1.516 \times 300 = 454.8 \text{ K} = 181.8°\text{C} \quad \textbf{Ans.}$$

28.8. Workdone by Reciprocating Air Compressor with Clearance Volume

In the previous articles, we have assumed that there is no clearance volume in the compressor cylinder. In other words, the entire volume of air, in the compressor cylinder, is compressed by the inward stroke of the piston. But in actual practice, it is not possible to reduce the clearance volume to zero, for mechanical reasons. Moreover, it is not desirable to allow the piston head to come in contact with the cylinder head. In addition to this, the passage leading to the inlet and outlet valves always contribute to clearance volume. In general, the clearance volume is expressed as some percentage of the piston displacement.

Now consider a reciprocating air compressor with clearance volume, as shown in Fig. 28.3.

Let p_1 = Initial pressure of air (before compression),

v_1 = Initial volume of air (before compression),

T_1 = Initial temperature of air (before compression),

p_2, v_2, T_2 = Corresponding values for the final conditions (*i.e.* at the delivery points),

r = Pressure ratio (*i.e.* p_2/p_1),

v_c = Clearance volume (*i.e.*, volume at point 3),

v_s = Stroke volume $= v_1 - v_c$, and

n = Polytropic index for compression and expansion.

The *p-v* diagram of a single stage single acting reciprocating air compressor with clearance volume (v_c) is shown in Fig. 28.3. We know that during return stroke, the air is compressed by its major part *i.e.* compression stroke 1-2. This compression continues, till the pressure p_2 in the cylinder is sufficient to force open the delivery valve at 2. After that, no more compression takes place with the inward movement of the piston. Now during the remaining part of compression stroke, compressed air is delivered till the piston reaches at 3. At this stage, there will be some air (equal to clearance volume) left in the clearance space of the cylinder at pressure p_2. After that air in the clearance space will expand during some part of outward stroke of the piston *i.e.* expansion stroke 3-4. This expansion continues till the pressure p_1 in the cylinder is sufficient to force open the inlet valve at 4. After that the air is sucked from the atmosphere during the suction stoke 4-1 at pressure p_1.

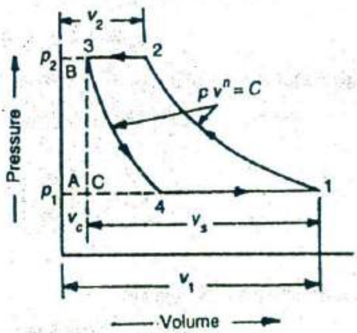


Fig. 28.3. *p-v* diagram with clearance volume.

Though the compression and expansion of air may be isothermal, isentropic or polytropic, yet for all calculation purposes, it is assumed to be polytropic. We know that work done by the compressor per cycle,

$$W = \text{Area 1-2-3-4} = \text{Area A-1-2-B} - \text{Area A-4-3-B}$$

$$= \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} p_1 v_4 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{n}{n-1} \times p_1 (v_1 - v_4) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{n}{n-1} \times m R T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

where $(v_1 - v_4)$ and m is equal to the actual volume and mass of air sucked by the piston per cycle respectively.

We see that the clearance volume does not effect the work done on the air and the power required for compressing the air. This is due to the reascn that the work required to compress the clearance volume air is theoretically regained during its expansion from 3 to 4.

Note : The terms v_4 and $(v_1 - v_4)$ are known as expanded clearance volume and effective swept volume respectively.

Example 28.4. *A single stage, single acting reciprocating air compressor has a bore of 200 mm and a stroke of 300 mm. It receives air at 1 bar and 20° C and delivers it at 5.5 bar. If the compression follows the law $pv^{1.3} = C$ and clearance volume is 5 percent of the stroke volume, determine : 1. the mean effective pressure ; and 2. the power required to drive the compressor, if it runs at 500 r.p.m.*

Solution. Given : $D = 200$ mm $= 0.2$ m ; $L = 300$ mm $= 0.3$ m ; $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2 ; $T_1 = 20°$ C $= 20 + 273 = 293$ K ; $p_2 = 5.5$ bar ; $n = 1.3$; $v_c = 5\%\ v_s$; $N = 500$ r.p.m

We know that stroke volume,

$$v_s = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4} (0.2)^2\, 0.3 = 0.009 42 \text{ m}^3$$

$\therefore$ Clearance volume,

$$v_c = 5\%\, v_s = 0.05 \times 0.009 42 = 0.000 47 \text{ m}^3$$

and initial volume of air, $v_1 = v_c + v_s = 0.000 47 + 0.009 42 = 0.009 89 \text{ m}^3$

We know that expanded clearance volume,

$$v_4 = v_c \left(\frac{p_2}{p_1} \right)^{\frac{1}{n}} = 0.000 47 \left(\frac{5.5}{1} \right)^{\frac{1}{1.3}} = 0.001 74 \text{ m}^3$$

$$\ldots (\because p_1 v_4^n = p_2 v_c^n)$$

and effective swept volume,

$$v_1 - v_4 = 0.009 89 - 0.001 74 = 0.008 15 \text{ m}^3$$

We know that work done by the compressor per cycle,

$$W = \frac{n}{n-1} \times p_1 (v_1 - v_4) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.3}{1.3 - 1} \times 1 \times 10^5 \times 0.008 15 \left[\left(\frac{5.5}{1} \right)^{\frac{1.3-1}{1.3}} - 1 \right] = 1702 \text{ N-m}$$

1. *Mean effective pressure*

We know that mean effective pressure,

$$p_m = \frac{\text{Work done}}{\text{Stroke volume}} = \frac{1702}{0.009 42} = 180 700 \text{ N/m}^2$$

$$= 1.807 \text{ bar } \textbf{Ans.}$$

2. *Power required to drive the compressor*

Since the compressor is single acting, therefore number of working strokes per minute,

$$N_w = N = 500$$

$\therefore$ Power required to drive the compressor,

$$P = \frac{W \times N_w}{60} = \frac{1702 \times 500}{60} = 14\,183 \text{ W} = 14183 \text{ kW} \textbf{ Ans.}$$

28.9. Multistage Compression

In the previous articles, we have been taking into consideration the compression of air in single stage. In other words, air is sucked, compressed in the cylinder and then delivered at a higher pressure. But sometimes, the air is required at a high pressure. In such cases, either we employ a large pressure ratio (in single cylinder) or compress the air in two or more cylinders in series. It has been experienced that if we employ single stage compression for producing high pressure air (say 8 to 10 bar), it suffers the following drawbacks :

1. The size of the cylinder will be too large.
2. Due to compression, there is a rise in temperature of the air. It is difficult to reject heat from the air in the small time available during compression.
3. Sometimes, the temperature of air, at the end of compression, is too high. It may heat up the cylinder head or burn the lubricating oil.

In order to overcome the above mentioned difficulties, two or more cylinders are provided in series with intercooling arrangement between them. Such an arrangement is known as *multistage compression.*

28.10. Advantages of Multistage Compression

Following are the main advantages of multistage compression over single stage compression :

1. The work done per kg of air is reduced in multistage compression with intercooler as compared to single stage compression for the same delivery pressure.
2. It improves the volumetric efficiency for the given pressure ratio.
3. The sizes of the two cylinders (*i.e.* high pressure and low pressure) may be adjusted to suit the volume and pressure of the air.
4. It reduces the leakage loss considerably.
5. It gives more uniform torque, and hence a smaller size flywheel is required.
6. It provides effective lubrication because of lower temperature range.
7. It reduces the cost of compressor.

28.11. Two-stage Reciprocating Air Compressor with Intercooler

A schematic arrangement for a two-stage reciprocating air compressor with water cooled intercooler is shown in Fig. 28.4.

First of all, the fresh air is sucked from the atmosphere in the low pressure (L.P.) cylinder during its suction stroke at intake pressure p_1 and temperature T_1. The air, after compression in the L.P. cylinder (*i.e.* first stage) from 1 to 2, is delivered to the intercooler at pressure p_2 and temperature T_2. Now the air is cooled in the intercooler from 2 to 3 at constant pressure p_2 and from temperature T_2 to T_3. After that, the air is sucked in the high pressure (H.P.) cylinder during its suction stroke.

Finally, the air, after further compression in the H.P. cylinder (*i.e.* second stage) from 3 to 4, is delivered by the compressor at pressure p_3 and temperature T_4.

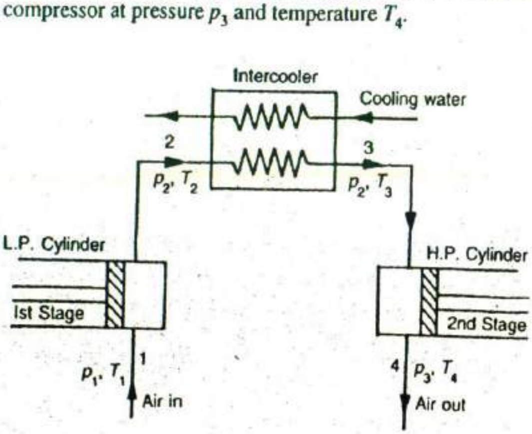


Fig. 28.4. Two-stage reciprocating air compressor with intercooler.

28.12. Assumptions in Two-stage Compression with Intercooler

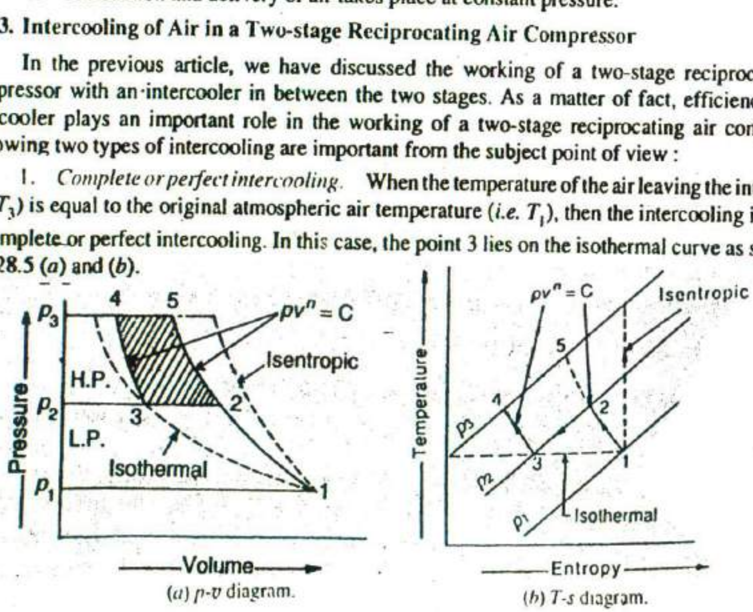
The following simplifying assumptions are made in case of two stage compression with intercooler :

1. The effect of clearance is neglected.
2. There is no pressure drop in the intercooler.
3. The compression in both the cylinders (*i.e.* L.P. and H.P.) is polytropic (*i.e.* $p\,v^n = C$).
4. The suction and delivery of air takes place at constant pressure.

28.13. Intercooling of Air in a Two-stage Reciprocating Air Compressor

In the previous article, we have discussed the working of a two-stage reciprocating air compressor with an intercooler in between the two stages. As a matter of fact, efficiency of the intercooler plays an important role in the working of a two-stage reciprocating air compressor. Following two types of intercooling are important from the subject point of view :

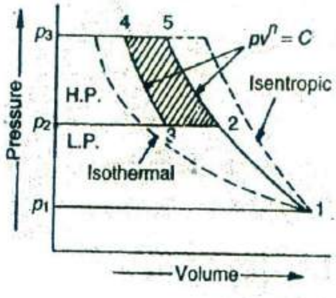
1. *Complete or perfect intercooling.* When the temperature of the air leaving the intercooler (*i.e.* T_3) is equal to the original atmospheric air temperature (*i.e.* T_1), then the intercooling is known as complete or perfect intercooling. In this case, the point 3 lies on the isothermal curve as shown in Fig. 28.5 (*a*) and (*b*).



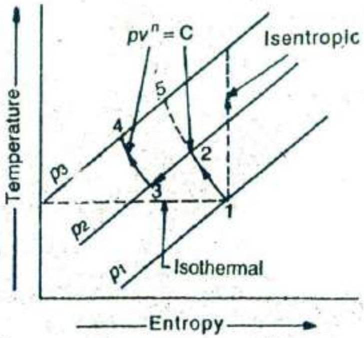
(*a*) *p-v* diagram. (*b*) *T-s* diagram.

Fig. 28.5. Complete intercooling of air.

2. *Incomplete or imperfect intercooling.* When the temperature of the air leaving the intercooler (*i.e.* T_3) is more than the original atmospheric air temperature (*i.e.* T_1), then the intercooling is known as incomplete or imperfect intercooling. In this case, the point 3 lies on the right side of the isothermal curve as shown in Fig. 28.6 (*a*) and (*b*).



(*a*) *p-v* diagram.



(*b*) *T-s* diagram.

Fig. 28.6. Incomplete intercooling of air.

Note : The amount of work saved due to intercooling is shown by the shaded area 2-3-4-5 in both the cases, to some scale. The amount of work saved with incomplete intercooling is less than that in case of complete intercooling.

28.14. Workdone by a Two-stage Reciprocating Air Compressor with Intercooler

Consider a two-stage reciprocating air compressor with intercooler compressing air in its L.P. and H.P. cylinders.

Let p_1 = Pressure of air entering the L.P. cylinder,

v_1 = Volume of the L.P. cylinder,

p_2 = Pressure of air leaving the L.P. cylinder or entering the H.P. cylinder,

v_2 = Volume of H.P. cylinder,

p_3 = Pressure of air leaving the H.P. cylinder, and

n = Polytropic index for both the cylinders.

Now we shall consider both the cases of incomplete intercooling as well as complete intercooling one by one.

1. *When the intercooling is incomplete*

We know that work done per cycle in L.P. cylinder,

$$W_1 = \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad \ldots (i)$$

Similarly, work done per cycle in compressing air in H.P. cylinder,

$$W_2 = \frac{n}{n-1} \times p_2 v_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 1 \right] \quad \ldots (ii)$$

$\therefore$ Total work done per cycle,

$$W = W_1 + W_2$$

$$= \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} \times p_2 v_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{n}{n-1} \left[p_1 v_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\} + p_2 v_2 \left\{ \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 1 \right\} \right] \quad \ldots (iii)$$

2. *When the intercooling is complete*

In case of complete intercooling, $p_1 v_1 = p_2 v_2$. Therefore substituting this value in equation (*iii*),

$$W = \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right]$$

$$= \frac{n}{n-1} \times m\,R\,T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right]$$

Example 28.5. *Estimate the work done by a two stage reciprocating single acting air compressor to compress 2.8 m³ of air per min at 1.05 bar and 10° C to a final pressure of 35 bar. The intermediate receiver cools the air to 30° C and 5.6 bar pressure. For air, take n = 1.4.*

Solution. Given : $v_1 = 2.8$ m³/min ; $p_1 = 1.05$ bar $= 1.05 \times 10^5$ N/m² ; $T_1 = 10°$ C $= 10 + 273 = 283$ K ; $p_3 = 35$ bar ; $T_3 = 30°$ C $= 30 + 273 = 303$ K ; $p_2 = 5.6$ bar $= 5.6 \times 10^5$ N/m² ; $n = 1.4$

Let v_2 = Volume of the high pressure cylinder.

We know that $\dfrac{p_1 v_1}{T_1} = \dfrac{p_2 v_2}{T_3}$

or $$v_2 = \frac{p_1 v_1 T_3}{p_2 T_1} = \frac{1.05 \times 10^5 \times 2.8 \times 303}{5.6 \times 10^5 \times 283} = 0.562 \text{ m}^3\text{/min}$$

$\therefore$ Work done by the compressor,

$$W = \frac{n}{n-1} \left[p_1 v_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\} + p_2 v_2 \left\{ \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 1 \right\} \right]$$

$$= \frac{1.4}{1.4-1} \left[1.05 \times 10^5 \times 2.8 \left\{ \left(\frac{5.6}{1.05} \right)^{\frac{1.4-1}{1.4}} - 1 \right\} \right.$$

$$\left. + 5.6 \times 10^5 \times 0.562 \times \left\{ \left(\frac{35}{5.6} \right)^{\frac{1.4-1}{1.4}} - 1 \right\} \right]$$

$$= 3.5\,[1.803 \times 10^5 + 2.166 \times 10^5] = 13.9 \times 10^5 \text{ N-m/min} \quad \textbf{Ans.}$$

28.15. Power Required to Drive a Two-stage Reciprocating Air Compressor

We have already obtained in the last article the expressions for the work done (W) per cycle of a two-stage reciprocating air compressor with incomplete and complete intercooling. The power required to drive the compressor may be obtained from the usual relation :

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

where N_w = Number of working strokes per minute.

Example 28.6. *A two- tage single acting reciprocating air compressor draws in air at a pressure of 1 bar and 17° C c .d compresses it to a pressure of 60 b ır. After compression in the L.P. cylinder, the air is cooled at constant pressure of 8 bar to a temperature of 37° C. The low pressure cylinder has a diameter of 150 mm and both the cylinders have 200 mm stroke. If the law of compression is $pv^{1.35} = C$, find the power of the compressor, when it runs at 200 r.p.m. Take $R = 287$ J/kg K.*

Solution. Given : $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2 ; $T_1 = 17°$ C $= 17 + 273 = 290$ K ; $p_3 = 60$ bar ; $p_2 = 8$ bar $= 8 \times 10^5$ N/m^2 ; $T_3 = 37°$ C $= 37 + 273 = 310$ K ; $D = 150$ mm $= 0.15$ m ; $L = 200$ mm $= 0.2$ m ; $n = 1.35$; $N = 200$ r.p.m. ; $R = 287$ J/kg K

We know that volume of L.P. cylinder,

$$v_1 = \frac{\pi}{4} \times D^2 \times L = \frac{\pi}{4} (0.15)^2\, 0.2 = 0.0035 \text{ m}^3$$

Let v_2 = Volume of H.P. cylinder.

We know that $\dfrac{p_1 v_1}{T_1} = \dfrac{p_2 v_2}{T_3}$

or

$$v_2 = \frac{p_1 v_1 T_3}{p_2 T_1} = \frac{1 \times 10^5 \times 0.0035 \times 310}{8 \times 10^5 \times 290} = 0.000\,47 \text{ m}^3$$

$\therefore$ Workdone by the compressor per stroke,

$$*W = \frac{n}{n-1}\left[p_1 v_1 \left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} + p_2 v_2 \left\{\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right\}\right]$$

$$= \frac{1.35}{1.35-1}\left[1 \times 10^5 \times 0.0035 \left\{\left(\frac{8}{1}\right)^{\frac{1.35-1}{1.35}} - 1\right\}\right]$$

$$+ \left[8 \times 10^5 \times 0.000\,47 \left\{\left(\frac{60}{8}\right)^{\frac{1.35-1}{1.35}} - 1\right\}\right] \text{ N-m}$$

$$= 3.86\,(250 + 258) = 1961 \text{ N-m}$$

Since the compressor is single acting, therefore number of working strokes per minute,

$$N_w = N = 200$$

We know that power of the compressor,

$$P = \frac{W \times N_w}{60} = \frac{1961 \times 200}{60} = 6540 \text{ W} = 6.54 \text{ kW} \textbf{ Ans.}$$

* Since T_3 is more than T_1, therefore it a case of incomplete intercooling.

28.16. Minimum Work Required for a Two-stage Reciprocating Air Compressor

We have already discussed in Art. 28.14 that maximum work is saved in a two-stage reciprocating air compressor with complete intercooling. We have also obtained a relation that work required to be done by a two-stage reciprocating air compressor with complete intercooling,

$$W = \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right] \qquad \ldots (i)$$

If the intake pressure p_1 and the delivery pressure p_3 are fixed, then least value of the intermediate or intercooler pressure p_2 may be obtained by differentiating the above equation with respect to p_2. It may be noted that value of p_2 thus obtained denotes the pressure of the intercooler at which the work required to drive the compressor is minimum. Thus work required is minimum, when

$$\frac{dW}{dp_2} = 0$$

$$\frac{d}{dp_2} \left[\frac{n}{n-1} \times p_1 v_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right\} \right] = 0$$

Now substituting $\left(\dfrac{n}{n-1} \right)$ as a (a constant) in the above equation,

$$\frac{d}{dp_2} \left[a\, p_1 v_1 \left\{ \left(\frac{p_2}{p_1} \right)^{a} + \left(\frac{p_3}{p_2} \right)^{a} - 2 \right\} \right] = 0$$

$$a\, p_1 v_1 \left[\left(\frac{1}{p_1} \right)^{a} (a)\, (p_2)^{a-1} + (p_3)^a (-a)\, (p_2)^{-a-1} \right] = 0$$

$$a\, (p_1)^{-a} (p_2)^{a-1} - a\, (p_3)^a (p_2)^{-a-1} = 0$$

$$a\, (p_1)^{-a} (p_2)^{a-1} = a\, (p_3)^a (p_2)^{-a-1}$$

$$\frac{(p_2)^{a-1}}{(p_2)^{-a-1}} = \frac{(p_3)^a}{(p_1)^{-a}}$$

$$(p_2)^{a-1} (p_2)^{a+1} = (p_3)^a (p_1)^a$$

$$(p_2)^{2a} = (p_3 p_1)^a \quad \text{or} \quad p_2^2 = p_3 p_1$$

$$\therefore \qquad p_2 = \sqrt{p_3 p_1} \qquad \ldots (ii)$$

or in other words,

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \left(\frac{p_3}{p_1} \right)^{1/2} \qquad \ldots (iii)$$

Now substituting the value of $\dfrac{p_3}{p_2} = \dfrac{p_2}{p_1}$ in equation (*i*), we have minimum work required for a two stage reciprocating air compressor,

$$W = \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 2 \right]$$

$$= 2 \times \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad \ldots (iv)$$

$= 2 \times$ Work required for each stage

Now substituting $\frac{p_2}{p_1} = \left(\frac{p_3}{p_1} \right)^{1/2}$ in equation (*iv*),

$$W = 2 \times \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_3}{p_1} \right)^{\frac{n-1}{2n}} - 1 \right]$$

Similarly, it can be proved that for a three stage compressor

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \left(\frac{p_4}{p_1} \right)^{1/3}$$

and minimum work required for a three stage compressor,

$$W = 3 \times \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$$

$$= 3 \times \frac{n}{n-1} \times m\, R\, T_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$$

Note: In general, for a compressor having q number of stages,

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \ldots = \frac{p_{q+1}}{p_q} = \left(\frac{p_{q+1}}{p_1} \right)^{1/q}$$

and minimum work required for compression,

$$W = q \times \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_{q+1}}{p_1} \right)^{\frac{n-1}{qn}} - 1 \right]$$

Example 28.7. *Estimate the minimum work required to compress 1 kg of air from 1 bar 27° C to 16 bar in two stages, if the law of compression is $pv^{1.25}$ = constant and the intercooling is perfect. Take $R = 287$ J/kg K.*

Solution. Given : $m = 1$ kg ; $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2 ; $T_1 = 27°$ C $= 27 + 273 = 300$ K ; $p_3 = 16$ bar ; $n = 1.25$; $R = 287$ J/kg K

We know that for perfect intercooling, the intercooler pressure,

$$p_2 = \sqrt{p_1 p_3} = \sqrt{1 \times 16} = 4 \text{ bar}$$

$\therefore$ Minimum work required to compress 1 kg of air,

$$W = 2 \times \frac{n}{n-1} \times mRT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= 2 \times \frac{1.25}{1.25-1} \times 1 \times 287 \times 300 \left[\left(\frac{4}{1} \right)^{\frac{1.25-1}{1.25}} - 1 \right] \text{N-m}$$

$$= 861\,000 \times 0.3195 = 275\,090 \text{ N-m} \quad \textbf{Ans.}$$

Example 28.8. *A two stage compressor takes in 2.82 m^3 of air per minute at a pressure of 1.05 bar and temperature of 22°C. It delivers the air at 8.44 bar. The compression is carried out in each cylinder according to the law $p\,v^{1.2}$ = Constant. The air is cooled to its initial temperature in intercooler. Neglecting clearance, find the minimum power required to drive the compressor.*

Solution. $v_1 = 2.82 \text{ m}^3/\text{min}$; $p_1 = 1.05 \text{ bar} = 1.05 \times 10^5 \text{ N/m}^2$; $T_1 = 22° \text{C} = 22 + 273 = 295 \text{ K}$; $p_3 = 8.44$ bar ; $n = 1.2$

Since the air is cooled to its initial temperature, therefore the intercooling is perfect. We know that for perfect intercooling, the intercooler pressure,

$$p_2 = \sqrt{p_1\,p_3} = \sqrt{1.05 \times 8.44} = 2.977 \text{ bar}$$

and minimum work required to drive the compressor,

$$W = 2 \times \frac{n}{n-1} \times p_1\,v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= 2 \times \frac{1.2}{1.2-1} \times 1.05 \times 10^5 \times 2.82 \left[\left(\frac{2.977}{1.05} \right)^{\frac{1.2-1}{1.2}} - 1 \right]$$

$$= 35.5 \times 10^5 (1.19 - 1) = 674\,500 \text{ N-m/min}$$

$\therefore$ Power required to drive the compressor

$$= \frac{674\,500}{60} = 11\,242 \text{ W} = 11.242 \text{ kW} \quad \textbf{Ans.} \qquad \ldots (\because 1 \text{ N-m/s} = 1 \text{ W})$$

Example 28.9. *A two stage air compressor compresses air from 1 bar and 20° C to 42 bar. If the law of compression is $p\,v^{1.35}$ = constant and the intercooling is complete to 20° C, find per kg of air : 1. The work done is compressing ; and 2. The mass of water necessary for abstracting the heat in the intercooler, if the temperature rise of the cooling water is 25° C.*

Take R = 287 J/kg K and c_p = 1 kJ/kg K.

Solution. Given : $p_1 = 1 \text{ bar} = 1 \times 10^5 \text{ N/m}^2$; $T_1 = 20° \text{C} = 20 + 273 = 293 \text{ K}$; $p_3 = 42$ bar $= 42 \times 10^5 \text{ N/m}^2$; $n = 1.35$; $T_3 = 20° \text{C} = 20 + 273 = 293 \text{ K}$; $m = 1$ kg ; Rise in temperature of cooling water = 25° C ; $R = 287$ J/kg K ; $c_p = 1$ kJ/kg K

We know that for complete intercooling, the intercooler pressure,

$$p_2 = \sqrt{p_1\,p_3} = \sqrt{1 \times 42} = 6.48 \text{ bar}$$

and volume of air admitted for compression,

$$v_1 = \frac{mRT_1}{p_1} = \frac{1 \times 287 \times 293}{1 \times 10^5} = 0.84 \text{ m}^3/\text{kg of air} \qquad \ldots (\because p_1 v_1 = mRT_1)$$

1. *Work done in compressing the air*

We know that work done in compresing the air,

$$W = \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right]$$

$$= \frac{1.35}{1.35-1} \times 1 \times 10^5 \times 0.84 \left[\left(\frac{6.48}{1} \right)^{\frac{1.35-1}{1.35}} + \left(\frac{42}{6.48} \right)^{\frac{1.35-1}{1.35}} - 2 \right] \text{ N-m}$$

$$= 3.24 \times 10^5 (1.62 + 1.62 - 2) = 4.017 \times 10^5 \text{ N-m} \quad \textbf{Ans.}$$

2. *Mass of water necessary for abstracting the heat in the intercooler*

Let m_w = Mass of water necessary/kg of air, and

T_2 = Temperature of the air entering the intercooler.

We know that $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} = \left(\frac{6.48}{1} \right)^{\frac{1.35-1}{1.35}} = 1.622$$

$\therefore$ $$T_2 = T_1 \times 1.622 = 293 \times 1.622 = 475.6 \text{ K}$$

We also know that heat gained by water

= Heat lost by air

$\therefore m_w \times c_w \times$ Rise in temperature

$$= m\, c_p (T_2 - T_3)$$

$$m_w \times 4.2 \times 25 = 1 \times 1\,(475.6 - 293) = 182.6$$

$\therefore$ $$m_w = 1.74 \text{ kg} \quad \textbf{Ans.}$$

Example 28.10. *Find the percentage saving in work by compressing air in two stages from 1 bar to 7 bar instead of in one stage. Assume compression index 1.35 in both the cases and optimum pressure and complete intercooling in two stage compressor.*

Solution. Given : $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2 ; $p_2 = 7$ bar $= 7 \times 10^5$ N/m^2 ; $n = 1.35$

We know that workdone in compresing air in one stage,

$$W_1 = \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.35}{1.35-1} \times 1 \times 10^5 v_1 \left[(7)^{\frac{1.35-1}{1.35}} - 1 \right] = 253\,100\, v_1 \text{ N-m} \quad \ldots (i)$$

and workdone in compressing air in two stages

$$W_2 = 2 \times \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{2n}} - 1 \right]$$

$$= 2 \times \frac{1.35}{1.35-1} \times 1 \times 10^5 v_1 \left[(7)^{\frac{1.35-1}{2 \times 1.35}} - 1 \right] \text{N-m}$$

$$= 221\,300\, v_1 \text{ N-m} \quad \ldots (ii)$$

$$\therefore \text{ Saving in work done } = \frac{253\,100\,v_1 - 221\,300\,v_1}{253\,100\,v_1} = 0.126 \text{ or } 12.6\% \text{ Ans.}$$

28.17. Heat Rejected in a Reciprocating Air Compressor

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

We know that heat rejected during polytropic compression per kg of air,

$$q_1 = \frac{\gamma - n}{\gamma - 1} \times \text{Work done} = \frac{\gamma - n}{\gamma - 1} \times \frac{R(T_2 - T_1)}{(n-1)}$$

$$= \frac{c_v(\gamma - n)(T_2 - T_1)}{n-1} \qquad \ldots [\because R = c_v(\gamma - 1)]$$

and heat rejected in the intercooler per kg of air,

$$q_2 = c_p(T_2 - T_3)$$

$\therefore$ Total heat rejected per kg of air,

$$q = q_1 + q_2 = \frac{c_v(\gamma - n)(T_2 - T_1)}{(n-1)} + c_p(T_2 - T_3)$$

Note : For complete intercooling $T_1 = T_3$. Therefore substituting this value in case of intercooling,

$$q = \frac{c_v(\gamma - n)(T_2 - T_1)}{n-1} + c_p(T_2 - T_1)$$

$$= (T_2 - T_1)\left[\frac{c_v(\gamma - n)}{n-1} + c_p\right]$$

Example 28.11. *A two-stage single acting reciprocating compressor takes in air at the rate of 0.2 m^3/s. The intake pressure and temperature of air are 0.1 MPa and 16° C. The air is compressed to a final pressure of 0.7 MPa. The intermediate pressure is ideal and intercooling is perfect. The compression index in both the stages is 1.25 and the compressor runs at 600 r.p.m. Neglecting clearance, determine 1. the intermediate pressure, 2. the total volume of each cylinder, 3. the power required to drive the compressor, and 4. the rate of heat rejection in the intercooler. Take C_p = 1.005 kJ/kg K and R = 287 J/kg K.*

Solution. Given : $v_1 = 0.2$ m³/s ; $p_1 = 0.1$ MPa $= 0.1 \times 10^6$ N/m² ; $T_1 = 16°$ C $= 16 + 273 = 289$ K ; $p_3 = 0.7$ MPa $= 0.7 \times 10^6$ N/m² ; $n = 1.25$; $N = 600$ r.p.m. ; $c_p = 1.005$ kJ/kg K ; $R = 287$ J/kg K

1. *Intermediate pressure*

We know that for perfect intercooling, the intermediate pressure (*i.e.* intercooler pressure),

$$p_2 = \sqrt{p_1\,p_3} = \sqrt{0.1 \times 0.7} = 0.2646 \text{ MPa Ans.}$$

2. *Total volume of each cylinder*

Let v_L = Volume of L.P. cylinder, and

v_H = Volume of H.P. cylinder.

Since the compressor is a single acting, therefore number of working strokes/min,

$$N_w = N = 600$$

$$\therefore \quad v_L = \frac{v_1 \times 60}{600} = \frac{0.2 \times 60}{600} = 0.02 \text{ m}^3 \text{ Ans.}$$

We know that $p_1 v_L = p_2 v_H$

$$\therefore \quad v_H = v_L \times \frac{p_1}{p_2} = 0.02 \times \frac{0.1}{0.2646} = 0.007\,56 \text{ m}^3 \textbf{ Ans.}$$

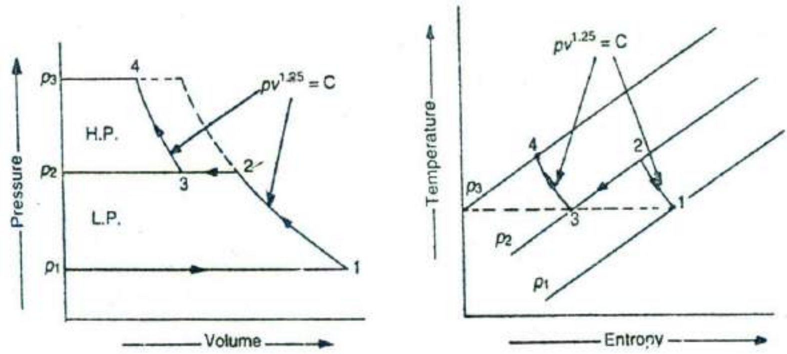


Fig. 28.7

3. *Power required to drive the compressor*

We know that work required to compress the air,

$$W = 2 \times \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= 2 \times \frac{1.25}{1.25-1} \times 0.1 \times 10^6 \times 0.2 \left[\left(\frac{0.2646}{0.1} \right)^{\frac{1.25-1}{1.25}} - 1 \right] \text{ J/s}$$

$$= 42\,970 \text{ J/s} = 42.97 \text{ kJ/s}$$

$\therefore$ Power required to drive the compressor,

$$P = 42.97 \text{ kW } \textbf{Ans.} \qquad \ldots (\because 1 \text{ kW} = 1 \text{ kJ/s})$$

4. *Rate of heat rejection in the intercooler*

Let m = Mass of air admitted for compression, and

T_2 = Temperature of the air entering the intercooler.

We know that $p_1 v_1 = m R T_1$

$$\therefore \quad m = \frac{p_1 v_1}{R\, T_1} = \frac{0.1 \times 10^6 \times 0.2}{287 \times 289} = 0.241 \text{ kg/s}$$

We also know that $\dfrac{T_2}{T_1} = \left(\dfrac{p_2}{p_1} \right)^{\frac{n-1}{n}} = \left(\dfrac{0.2646}{0.1} \right)^{\frac{1.25-1}{1.25}} = 1.215$

$$\therefore \quad T_2 = T_1 \times 1.215 = 289 \times 1.215 = 351.1 \text{ K}$$

We know that heat rejected in the intercooler

$$= m\, c_p\, (T_2 - T_3)$$

$$= 0.241 \times 1.005\, (351.1 - 289) = 15 \text{ kJ/s} \quad \textbf{Ans.} \qquad \ldots (\because T_3 = T_1)$$

28.18. Ratio of Cylinder Diameters

Consider a single acting two stage reciprocating air compressor with complete intercooling, compressing air in its L.P. and H.P. cylinders.

Let p_1 = Pressure of air entering the L.P. cylinder,

v_1 = Volume of the L.P. cylinder,

p_2 = Pressure of air leaving the L.P. cylinder (or intercooler pressure),

v_2 = Volume of H.P. cylinder,

p_3 = Pressure of air leaving the H.P. cylinder,

D_1 = Diameter of L.P. cylinder,

D_2 = Diameter of H.P. cylinder, and

L = Length of the stroke of each cylinder.

We know that for complete intercooling,

$$p_1 v_1 = p_2 v_2$$

$$\frac{p_2}{p_1} = \frac{v_1}{v_2} = \frac{\frac{\pi}{4}(D_1)^2 L}{\frac{\pi}{4}(D_2)^2 L} = \frac{(D_1)^2}{(D_2)^2}.$$

or

$$\frac{D_1}{D_2} = \left(\frac{p_2}{p_1}\right)^{1/2} \qquad \ldots (i)$$

We also know that for two stage compression with complete intercooling,

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

Substituting the value of p_2/p_1 in equation (*i*),

$$\frac{D_1}{D_2} = \left(\frac{p_3}{p_1}\right)^{1/4}$$

Example 28.12. *A two stage air compressor, with complete intercooling, delivers air to the mains at a pressure of 30 bar, the suction conditions being 1 bar and 27° C. If both cylinders have the same stroke, find the ratio of the cylinder diameters, for the efficiency of compression to be maximum. Assume the index of compression to be 1.3.*

Solution. Given : $p_3 = 30$ bar ; $p_1 = 1$ bar, *$T_1 = 27°$ C ; $n = 1.3$

We know that for complete intercooling, the intercooler pressure,

$$p_2 = \sqrt{p_1\, p_3} = \sqrt{1 \times 30} = 5.48 \text{ bar}$$

* Superfluous data

$\therefore$ Ratio of cylinder diameters,

$$\frac{D_1}{D_2} = \left(\frac{p_2}{p_1}\right)^{1/2} = \left(\frac{5.48}{1}\right)^{1/2} = 2.34 \text{ Ans.}$$

Note : The ratio of cylinder diameters is also given by

$$\frac{D_1}{D_2} = \left(\frac{p_3}{p_1}\right)^{1/4} = \left(\frac{30}{1}\right)^{1/4} = 2.34 \text{ Ans.}$$

Example 28.13. *A multi-stage air compressor is to be designed to elevate the pressure from 1 bar to 100 bar such that the stage pressure ratio will not exceed 4. Determine : 1. Number of stages, 2. Exact stage pressure ratio, and 3. Intermediate pressure.*

Solution. Given : $p_1 = 1$ bar ; $p_{q+1} = 100$ bar ; $\dfrac{p_{q+1}}{p_q} = 4$

1. *Number of stages*

Let q = Number of stages.

We know that $\dfrac{p_{q+1}}{p_q} = \left(\dfrac{p_{q+1}}{p_1}\right)^{1/q}$ or $4 = \left(\dfrac{100}{1}\right)^{1/q}$

Taking log on both sides,

$$\log 4 = \frac{1}{q}\log 100 \quad \text{or} \quad 0.6021 = \frac{c}{q} \times 2$$

$\therefore$ $q = 2/0.6021 = 3.32$ say 4 **Ans.**

2. *Exact stage pressure ratio*

We know that $\dfrac{p_{q+1}}{p_q} = \left(\dfrac{p_{q+1}}{p_1}\right)^{1/q} = \left(\dfrac{100}{1}\right)^{1/4} = 3.162$ **Ans.**

3. *Intermediate pressure*

Let $^*p_3$ = Intermediate pressure.

We know that $\dfrac{p_5}{p_4} = 3.162$

$\therefore$ $p_4 = \dfrac{p_5}{3.162} = \dfrac{100}{3.162} = 31.62$ bar

Similarly $\dfrac{p_4}{p_3} = \dfrac{p_5}{p_4} = 3.162$ or $p_3 = \dfrac{p_4}{3.162} = \dfrac{31.62}{3.162} = 10$ bar **Ans.**

Example 28.14. *A three stage compressor compresses air from 1 bar to 35 bar and delivers it at the higher pressure to a receiver. The initial temperature is 17° C. The law of compression is $pv^{1.25}$ = constant, and is the same for each stage. Assuming conditions of minimum work, perfect intercooling and that the effect of cylinder clearance and valve resistance etc., may be neglected, find the power required to deliver 14 m^3/ min air, measured at the suction conditions. Find also the intermediate pressures.*

---

* The intermediate pressure of an air compressor having four stages is p_3.

Solution. Given : $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2 ; $p_4 = 35$ bar ; $T_1 = 17°$ C $= 17 + 273 = 290$ K ; $v_1 = 14$ m^3/min

Power required

We know that minimum work required per min.,

$$W = \frac{3\,n}{n-1} \times p_1 v_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$$

$$= \frac{3 \times 1.25}{1.25 - 1} \times 1 \times 10^5 \times 14 \left[\left(\frac{35}{1} \right)^{\frac{1.25-1}{3 \times 1.25}} - 1 \right] \text{ N-m/min}$$

$$= 5.65 \times 10^6 \text{ N-m/min} = 5650 \text{ kJ/min}$$

$$\therefore \text{ Power required} = \frac{5650}{60} = 94.2 \text{ kW} \quad \textbf{Ans.}$$

Intermediate pressures

Let p_2 and p_3 = Intermediate pressures.

We know that for minimum work,

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \left(\frac{p_4}{p_1} \right)^{1/3}$$

$$\therefore \quad \frac{p_2}{p_1} = \left(\frac{p_4}{p_1} \right)^{1/3} = (35)^{1/3} = 3.27$$

or $$p_2 = p_1 \times 3.27 = 1 \times 3.27 = 3.27 \text{ bar} \quad \textbf{Ans.}$$

Similarly $$\frac{p_3}{p_2} = \left(\frac{p_4}{p_1} \right)^{1/3} = 3.27$$

$$\therefore \quad p_3 = p_2 \times 3.27 = 3.27 \times 3.27 = 10.7 \text{ bar} \quad \textbf{Ans.}$$

Example 28.15. *A three stage compressor delivers air at 70 bar from an atmospheric pressure of 1 bar and 30° C. Assuming the intercooling complete, estimate the amount of minimum work required to deal with 1 kg of air. Also find the amount of heat rejected in each intercooler. The index of compression is 1.2 throughout. Take c_p for air = 1.005 kJ/kg K.*

Solution. Given : $p_4 = 70$ bar ; $p_1 = 1$ bar ; $T_1 = 30°$ C $= 30 + 273 = 303$ K ; $m = 1$ kg ; $n = 1.2$; $c_p = 1.005$ kJ/kg K

Amount of minimum work required

We know that amount of minimum work required,

$$W = \frac{3\,n}{n-1} \times mRT_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$$

$$= \frac{3 \times 1.2}{1.2 - 1} \times 1 \times 287 \times 303 \left[\left(\frac{70}{1} \right)^{\frac{1.2-1}{3 \times 1.2}} - 1 \right] = 417\,070 \text{ J}$$

$$= 417.07 \text{ kJ} \quad \textbf{Ans.}$$

Heat rejected in each intercooler

Let T_2 = Temperature of air at the end of first stage.

Since the air compressor is three stage, therefore

$$\frac{p_2}{p_1} = \left(\frac{p_4}{p_1}\right)^{1/3} = \left(\frac{70}{1}\right)^{1/3} = 4.12$$

We know that $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (4.12)^{\frac{1.2-1}{1.2}} = 1.266$$

$\therefore$ $$T_2 = T_1 \times 1.266 = 303 \times 1.266 = 383.6 \text{ K}$$

Since the pressure ratio and index of compression is same for each stage, and intercooling is complete, therefore temperature of air will be same on entering each intercooler.

$\therefore$ Heat rejected in each intercooler

$$= c_p (T_2 - T_3) = 1.005\,(383.6 - 303) = 81 \text{ kJ} \text{ **Ans.**}$$

Example 28.16. *A three stage single acting reciprocating compressor has perfect intercooling. The pressure and temperature at the end of suction stroke in L.P. cylinder is 1.013 bar and 15° C respectively. If 8.4 m³ of free air is delivered by the compressor at 70 bar per minute and the work done is minimum, calculate :*

1. L.P. and I.P. delivery pressures ; 2. Ratio of cylinder volumes ; and 3. Total indicated power.

Neglect clearance and assume n = 1.2.

Solution. Given : $p_1 = 1.013$ bar $= 1.013 \times 10^5$ N/m² ; $T_1 = 15°$ C $= 15 + 273 = 288$ K ; $v_1 = 8.4$ m³/min ; $p_4 = 70$ bar ; $n = 1.2$

1. *L.P. and I.P. delivery pressures*

Let p_2 = L.P. delivery pressure, and

p_3 = I.P. delivery pressure.

We know that for minimum work,

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \left(\frac{p_4}{p_1}\right)^{1/3}$$

$\therefore$ $$\frac{p_2}{p_1} = \left(\frac{p_4}{p_1}\right)^{1/3} = \left(\frac{70}{1.013}\right)^{1/3} = 4.1$$

or $$p_2 = p_1 \times 4.1 = 1.013 \times 4.1 = 4.153 \text{ bar } \textbf{Ans.}$$

Similarly, $$p_3 = p_2 \times 4.1 = 4.153 \times 4.1 = 17.03 \text{ bar } \textbf{Ans.}$$

2. *Ratio of cylinder volumes*

Let $v_1 : v_2 : v_3$ = Ratio of cylinder volumes.

We know that for minimum work,

$$p_1 v_1 = p_2 v_2 = p_3 v_3$$

or $$\frac{v_1}{v_2} = \frac{p_2}{p_1} = \frac{4.153}{1.013} = 4.1 \qquad \ldots (i)$$

$$\therefore \qquad v_1 = 4.1\, v_2$$

Also $$\frac{v_2}{v_3} = \frac{p_3}{p_2} = \frac{17.03}{4.157} = 4.1$$

$$\therefore \qquad v_2 = 4.1\, v_3 \qquad \ldots (ii)$$

Now taking v_3 as 1 m^3, we find that ratio of cylinder volumes,

$$v_1 : v_2 : v_3 = 16.81 : 4.1 : 1 \textbf{ Ans.}$$

3. *Total indicated power*

We know that minimum work required,

$$W = \frac{3n}{n-1} \times p_1 v_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$$

$$= \frac{3 \times 1.2}{1.2 - 1} \times 1.013 \times 10^5 \times 8.4 \left[\left(\frac{70}{1.013} \right)^{\frac{1.2-1}{3 \times 1.2}} - 1 \right] \text{J/min}$$

$$= 4067 \times 10^3 \text{ J/min} = 4067 \text{ kJ/min}$$

$\therefore$ **Total indicated power,**

$$P = \frac{4067}{60} = 67.8 \text{ kW } \textbf{Ans.}$$

EXERCISES

1. A single stage reciprocating air compressor is required to compress 60 cubic metres of air from 1 bar to 8 bar at 22° C. Find work done by the compressor, if the compression of air is 1. isothermal, 2. isentropic with isentropic index as 1.4 ; and 3. polytropic with polytropic index as 1.25. **[Ans.** 12.5 MJ ; 17 MJ ; 15.5 MJ**]**

2. Find the power required to compress and deliver 2 kg of air per minute from 1 bar and 20° C to a delivery pressure 7 bar, when the compression is carried out in a single stage compressor. The compression of air follows the law $p\,v^{1.4}$ = constant. Neglect clearance. Take R = 287 J/kg K. **[Ans.** 7.3 kW**]**

3. Determine the cylinder dimensions of a double acting, 11 kW indicated power, air compressor which compresses air from 1 bar to 7 bar according to the law $pv^{1.2}$ = constant. The average piston speed is 150 m/s. Assume stroke to diameter ratio of 1.5 and neglect clearance. **[Ans.** 156 mm ; 234 mm**]**

4. A small single acting compressor has a bore and stroke both of 100 mm and is driven at 400 r.p.m. The clearance volume is 80×10^3 mm³ and the index of compression and expansion is 1.2. The suction pressure is 0.95 bar and the delivery pressure is 8 bar. Calculate

1. the volume of free air at 1.013 bar and 20° C dealt with per minute if the temperature at the start of compression is 30° C ; 2. the mean effective pressure of the indicator diagram assuming constant suction and delivery pressure ; and 3. the power required to drive the compressor.

[Ans. 0.1426 m³/min ; 1.22 bar ; 0.637 kW**]**

5. An air compressor takes in air at 0.98 bar and 20° C and compresses it according to the law $pv^{1.2}$ = C. It is then delivered to a receiver at constant pressure of 9.8 bar. Determine : 1. the temperature at the end of compression ; 2. the workdone per kg of air ; 3. the heat transferred during the compression ; and 4. the workdone during delivery. Take R = 287 J/kg K and γ = 1.4. **[Ans.** 150° C ; 236 kJ ; 118 kJ ; 123.7 kJ**]**

6. A two stage air compressor with perfect intercooling takes in air at 1 bar pressure and 27° C. The law of compression in both the stages is $pv^{1.3}$ = constant. The compressed air is delivered at 9 bar from the H.P. cylinder to an air receiver. Calculate per kg of air 1. the minimum workdone, and 2. the heat rejected to intercooler.

[Ans. 215 kJ ; 86.6 kJ**]**

7. A two stage air compressor takes in 22.5 kg of air per minute at 15°C and 1 bar and delivers it at 16.5 bar. At the intermediate pressure, it is cooled to initial temperature. Assuming an ideal diagram with no clearance and compression according to $pv^{1.2}$ = constant, determine the intermediate pressure that gives least work. Also find the heat rejected in the intercooler per minute and minimum power required to run the compressor. Take $c_p = 1$ kJ/kg K; and $R = 287$ J/kg K **[Ans. 4.06 bar ; 1701 kJ/min ; 97.8 kW]**

8. A single acting compressor is required to deliver air at 70 bar from a suction pressure of 1 bar at the rate of 2.3 m³/min. measured at free air conditions of 1.013 bar and 15° C. The temperature at the end of the suction stroke is 32° C. Calculate the indicated power required if the compression is carried out in two stages with an ideal intermediate pressure and complete intercooling. The index of compression and expansion for both stages is 1.25. Also calculate the heat rejected per minute to the intercooler and the saving in power over single stage compression. For air, $c_p = 1$ kJ/kg K and $c_v = 0.718$ kJ/kg K. Neglect clearance volume.

[Ans. 22 kW ; 461.2 kJ/min ; 5.84 kW]

9. A two stage reciprocating air compressor delivers 40.5 kg/min at 9.8 bar. The intake pressure is 1 bar and the intake temperature is 15.5° C. The compression follows $p\,v^{1.31}$ = constant and the intercooler cools the air back to the intake temperature. Neglecting clearance, calculate 1. the optimum intermediate pressure ; 2. the power to be delivered to each cylinder ; and 3. the rate of heat transfer from the cylinders and intercooler.

[Ans. 3.13 bar ; 73.2 kW ; 3625 kJ/min]

10. A three stage reciprocating compressor compresses air from 1 bar and 26° C to 36 bar. The law of compression is $pv^{1.3}$ = constant and is same for all the three stages of compression. Assuming perfect intercooling and neglecting clearance, find the minimum power required to compress 0.25 m³/s of free air. Also find the intermediate pressures. **[Ans. 103.12 kW ; 3.302 bar ; 10.903 bar]**

QUESTIONS

1. Classify air compressors. Describe the working of a single stage reciprocating air compressor.

2. Draw *p-v* and *T-s* diagram for a single stage reciprocating air compressor, without clearance. Derive the expression for the workdone when compression is (*a*) isothermal, and (*b*) isentropic.

3. Sketch the theoretical indicator diagram for a single stage, single cylinder reciprocating compressor with clearance volume showing the various processes. For such a compressor, derive the expression for workdone in terms of mass rate of flow of air, initial temperature, pressure ratio and index of compression.

4. When is multi-stage compression used for air ? What are its advantages ?

5. Explain the effect of intercooling in a multistage reciprocating compressor.

6. Discuss briefly a two stage air compressor with intercooler. Draw the ideal *p-v* diagram. Derive the expression for work done per unit mass of air. Establish that the workdone is minimum when the pressure ratio for each stage is the same and there is complete intercooling.

7. In a two stage air compressor, in which intercooling is perfect, prove that the work done in compression is a minimum when the pressure in the intercooler is the geometric mean between the initial and final pressures. Draw the indicator diagram for two stage compression.

8. Discuss the procedure for obtaining the ratio of cylinder dimensions in an air compressor.

OBJECTIVE TYPE QUESTIONS

1. The volume of air delivered by the compressor is called

   (*a*) free air delivery (*b*) compressor capacity

   (*c*) swept volume (*d*) none of these

2. The volume of air sucked by the compressor during its suction stroke is called

   (*a*) free air delivery (*b*) compressor capacity

   (*c*) swept volume (*d*) none of these

3. The ratio of workdone per cycle to the stroke volume of the compressor is known as

(*a*) compressor capacity (*b*) compression ratio

(*c*) compressor efficiency (*d*) mean effective pressure

4. In a single stage, single acting reciprocating air compressor without clearance volume, the workdone is minimum during

(*a*) isothermal compression (*b*) isentropic compression

(*c*) polytropic compression (*d*) none of these

5. The pressure of air at the beginning of the compression stroke is ....atmospheric pressure.

(*a*) equal to (*b*) less than (*c*) more than

6. A 3 m^3/min compressor means that it

(*a*) compresses 3 m^3/min of standard air (*b*) compresses 3 m^3/min of free air

(*c*) delivers 3 m^3/min of compressed air

(*d*) delivers 3 m^3/min of compressed air at delivery pressure.

7. The multi-stage compression of air as compared to single stage compression

(*a*) improves volumetric efficiency for the given pressure ratio

(*b*) reduces workdone per kg of air (*c*) gives more uniform torque

(*d*) reduces cost of compressor (*e*) all of the above

8. The intercooling in multi-stage compressors is done

(*a*) to cool the air during compression (*b*) to cool the air at delivery

(*c*) to enable compression in two stages (*d*) to minimise the work of compression

9. The intercooler pressure (p_2) for minimum work required, for a two stage reciprocating air compressor, is given by

(*a*) $p_2 = p_1 p_3$ (*b*) $p_2 = p_1/p_3$ (*c*) $p_2 = \sqrt{p_1 p_3}$ (*d*) $p_2 = p_3/p_1$

where p_1 = Intake pressure of air, and

p_3 = Delivery pressure of air.

10. The ratio of cylinder diameters for a single acting, two stage reciprocating air compressor with complete intercooling, is given by

(*a*) $D_1/D_2 = \sqrt{p_1 p_2}$ (*b*) $D_1/D_2 = \sqrt{p_1/p_2}$

(*c*) $D_1/D_2 = \sqrt{p_2/p_1}$ (*d*) none of these

where D_1 = Diameter of L.P. cylinder, and

D_2 = Diameter of H.P. cylinder.

ANSWERS

1. (*b*)	2. (*c*)	3. (*d*)	4. (*a*)	5. (*b*)
6. (*b*)	7. (*e*)	8. (*d*)	9. (*c*)	10. (*c*)