

Reciprocating Compressor

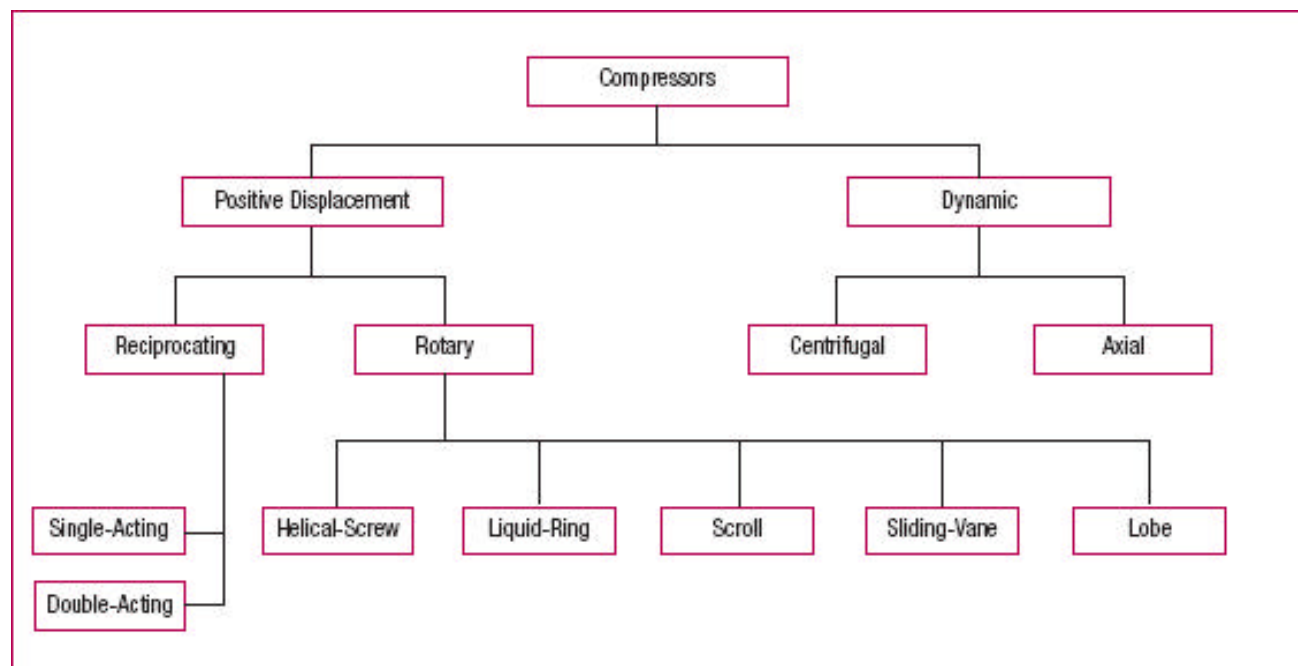
5.1 INTRODUCTION

Compressors are work absorbing devices which are used for increasing pressure of fluid at the expense of work done on fluid.

The compressors used for compressing air are called air compressors. Compressors are invariably used for all applications requiring high pressure air. Some of popular applications of compressor are, for driving pneumatic tools and air operated equipments, spray painting, compressed air engine, supercharging surface cleaning, refrigeration and air conditioning, chemical industry etc. compressors are supplied with low pressure air (or any fluid) at inlet which comes out as high pressure air (or any fluid) at outlet. Work required for increasing pressure of air is available from the prime mover driving the compressor. Generally, electric motor, internal combustion engine or steam engine, turbine etc. are used as prime movers. Compressors are similar to fans and blowers but differ in terms of pressure ratios. Fan is said to have pressure ratio up to 1.1 and blowers have pressure ratio between 1.1 to 4 while compressors have pressure ratios more than 4.

5.2 CLASSIFICATION OF COMPRESSORS

Table-5.1 Types of Compressors



Compressors can be classified in the following different ways.

- (a) **Based on principle of operation:** Based on the principle of operation compressors can be classified as.
 - (i) Positive displacement compressor.
 - (ii) Non-positive displacement compressors.

In positive displacement compressors the compression is realized by displacement of solid boundary and preventing fluid by solid boundary from flowing back in the direction of pressure gradient. Due to solid wall displacement these are capable of providing quite large pressure ratios. Positive displacement compressors can be further classified based on the type of mechanism used for compression. These can be

- (i) Reciprocating type positive displacement compressors
- (ii) Rotary type positive displacement compressors.

Reciprocating compressors generally, employ piston-cylinder arrangement where displacement of piston in cylinder causes rise in pressure. Reciprocating compressors are capable of giving large pressure ratios but the mass handling capacity is limited or small. Reciprocating compressors may also be single acting compressor or double acting compressor. Single acting compressor has one delivery stroke per revolution while in double acting there are two delivery strokes per revolution of crank shaft. Rotary compressors employing positive displacement have a rotary part whose boundary causes positive displacement of fluid and thereby compression. Rotary compressors of this type are available in the names as given below;

- (i) Roots blower
- (ii) Vane type compressors

Rotary compressors of above type are capable of running at higher speed and can handle large mass flow rate than reciprocating compressors of positive displacement type.

Non-positive displacement compressors, also called as steady flow compressors use dynamic action of solid boundary for realizing pressure rise. Here fluid is not contained in definite volume and subsequent volume reduction does not occur as in case of positive displacement compressors. Non-positive displacement compressor may be of 'axial flow type' or 'centrifugal type' depending upon type of flow in compressor.

(b) **Based on number of stages:** Compressors may also be classified on the basis of number of stages. Generally, the number of stages depend upon the maximum delivery pressure. Compressors can be single stage or multistage. Normally maximum compression ratio of 5 is realized in single stage compressors. For compression ratio more than 5 the multistage compressors are used.

Type values of maximum delivery pressures generally available from different type of compressor are,

- (i) Single stage Compressor, for delivery pressure upto 5 bar.
- (ii) Two stage Compressor, for delivery pressure between 5 to 35 bar
- (iii) Three stage Compressor, for delivery pressure between 35 to 85 bar.
- (iv) Four stage compressor, for delivery pressure more than 85 bar

(c) **Based on Capacity of compressors :** Compressors can also be classified depending upon the capacity of Compressor or air delivered per unit time. Typical values of capacity for different compressors are given as;

- (i) Low capacity compressors, having air delivery capacity of $0.15 \text{ m}^3/\text{s}$ or less
- (ii) Medium capacity compressors, having air delivery capacity between 0.15 to $5 \text{ m}^3/\text{s}$.
- (iii) High capacity compressors, having air delivery capacity more than $5 \text{ m}^3/\text{s}$

(d) **Based on highest pressure developed:** Depending upon the maximum pressure available from compressor they can be classified as low pressure, medium pressure, high pressure and super high pressure compressors. Typical values of maximum pressure developed for different compressors are as under:

- (i) Low pressure compressor, having maximum pressure upto 1 bar
- (ii) Medium pressure compressor, having maximum pressure from 1 bar to 8 bar
- (iii) High pressure compressor, having maximum pressure from 8 to 10 bar
- (iv) Super high pressure compressor, having maximum pressure more than 10 bar.

5.3 Reciprocating Compressors

Reciprocating Compressor has piston cylinder arrangement as shown Fig.5.1

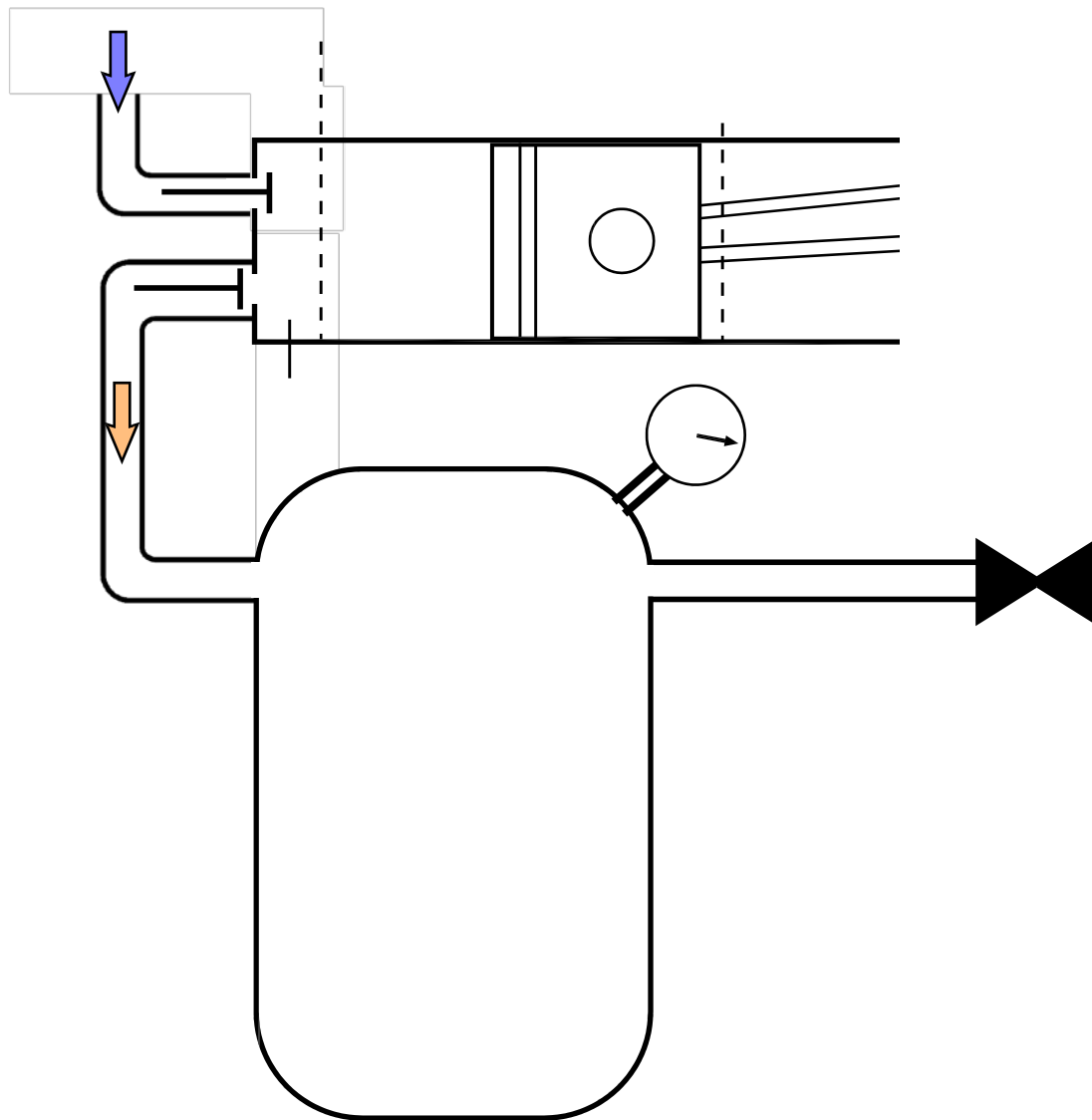


Fig.5.1 Reciprocating Compressor

Reciprocating Compressor has piston, cylinder, inlet valve, exit valve, connecting rod, crank, piston pin, crank pin and crank shaft. Inlet valve and exit valves may be of spring loaded type which get opened and closed due to pressure differential across them. Let us consider piston to be at top dead centre (TDC) and move towards bottom dead centre (BDC). Due to this piston movement from TDC to BDC suction pressure is created causing opening of inlet valve. With this opening of inlet valve and suction pressure the atmospheric air enters the cylinder.

Air gets into cylinder during this stroke and is subsequently compressed in next stroke with both inlet valve and exit valve closed. Both inlet valve and exit valves are of plate type and spring loaded so as to operate automatically as and when sufficient pressure difference is available to cause deflection in spring of valve plates to open them. After piston reaching BDC it reverses its motion and compresses the air inducted in previous stroke. Compression is continued till the pressure of air inside becomes sufficient to cause deflection in exit valve. At the moment when exit valve plate gets lifted the exhaust of compressed air takes place. This piston again reaches TDC from where downward piston movement is again accompanied by suction. This is how reciprocating compressor. Keeps on working as flow device. In order to counter for the heating of piston-cylinder arrangement during compression the provision of cooling the cylinder is there in the form of cooling jackets in the body. Reciprocating compressor described above has suction, compression and discharge as three prominent processes getting completed in two strokes of piston or one revolution of crank shaft.

5.4 Thermodynamic Analysis

Compression of air in compressor may be carried out following number of thermodynamic processes such as isothermal compression, polytropic compressor or adiabatic compressor. Fig.16.3 shows the thermodynamic cycle involved in compressor. Theoretical cycle is shown neglecting clearance volume but in actual cycle clearance volume can not be negligible. Clearance volume is necessary in order to prevent collision of piston with cylinder head, accommodating valve mechanism etc., Compression process is shown by process 1-2, $1-2^1$, $1-2^{11}$ and $1-2^{111}$ following isothermal, polytropic and adiabatic processes.

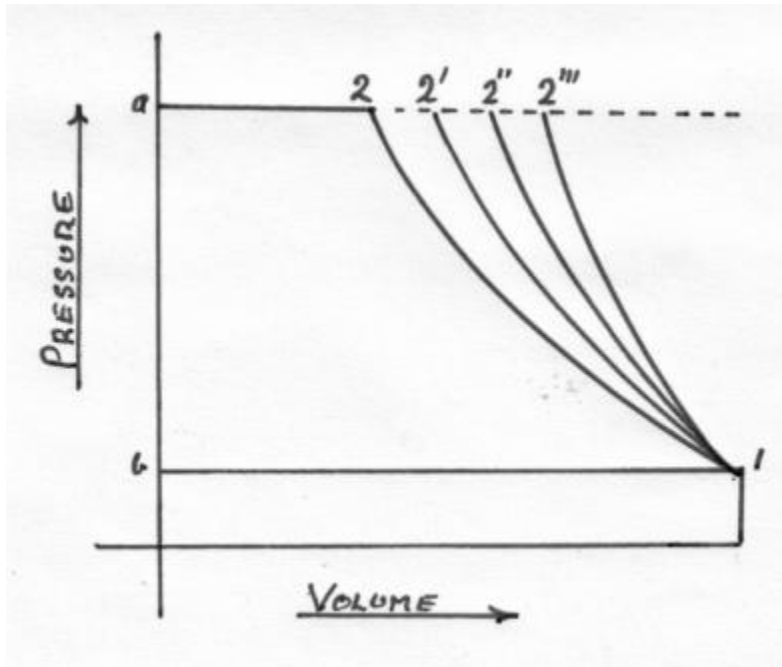


Fig.5.2 P-V diagram for Reciprocating Compressor without Clearance

On P-V diagram process 4-1 shows the suction process followed by compression during 1-2 and discharge through compressor is shown by process 2-3.

Air enters compressor at pressure p_1 and is compressed upto p_2 . Compression work requirement can be estimated from the area below the each compression process. Area on p-V diagram shows that work requirement shall be minimum with isothermal process 1-2'. Work requirement is maximum with process 1-2 ie., adiabatic process. As a designer one shall be interested in a compressor having minimum compression work requirement. Therefore, ideally compression should occur isothermally for minimum work input. In practice it is not possible to have isothermal compression because constancy of temperature during compression can not be realized. Generally, compressors run at substantially high speed while isothermal compression requires compressor to run at very slow speed so that heat evolved during compression is dissipated out and temperature remains constant. Actually due to high speed running of compressor the compression process may be assumed to be near adiabatic or polytropic process following law of compression as $pV^n = C$ with of 'n' varying between 1.25 to 1.35 for air. Compression process following three processes is also shown on T-s diagram in Fig.16.4. it is thus obvious that actual compression process should be compared with isothermal compression process. A mathematical parameter called isothermal efficiency is defined for quantifying the degree of deviation of actual compression process from ideal compression process. Isothermal efficiency is defined by the ratio is isothermal work and actual indicated work in reciprocating compressor.

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

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

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

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

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

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

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

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

In case of compressor having adiabatic compression process,

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

Or

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

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

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

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

- (i) Faster heat dissipation from inside of compressor to outside by use of fins over cylinder. Fins facilitate quick heat transfer from air being compressed to atmosphere so that temperature rise during compression can be minimized.
- (ii) Water jacket may be provided around compressor cylinder so that heat can be picked by cooling water circulating through water jacket. Cooling water circulation around compressor regulates rise in temperature to great extent.
- (iii) The water may also be injected at the end of compression process in order to cool the air being compressed. This water injection near the end of compression process requires special arrangement in compressor and also the air gets mixed with water and needs to be separated out before being used. Water injection also contaminates the lubricant film inner surface of cylinder and may initiate corrosion etc, The water injection is not popularly used.
- (iv) In case of multistage compression in different compressors operating serially, the air leaving one compressor may be cooled upto ambient state or somewhat high temperature before being injected into subsequent compressor. This cooling of fluid being compressed between two consecutive compressors is called intercooling and is frequently used in case of multistage compressors.

Considering clearance volume: With clearance volume the cycle is represented on Fig.5.3 The work done for compression of air polytropically can be given by the area enclosed in cycle 1-2-3-4. Clearance volume in compressors varies from 1.5% to 35% depending upon type of compressor.

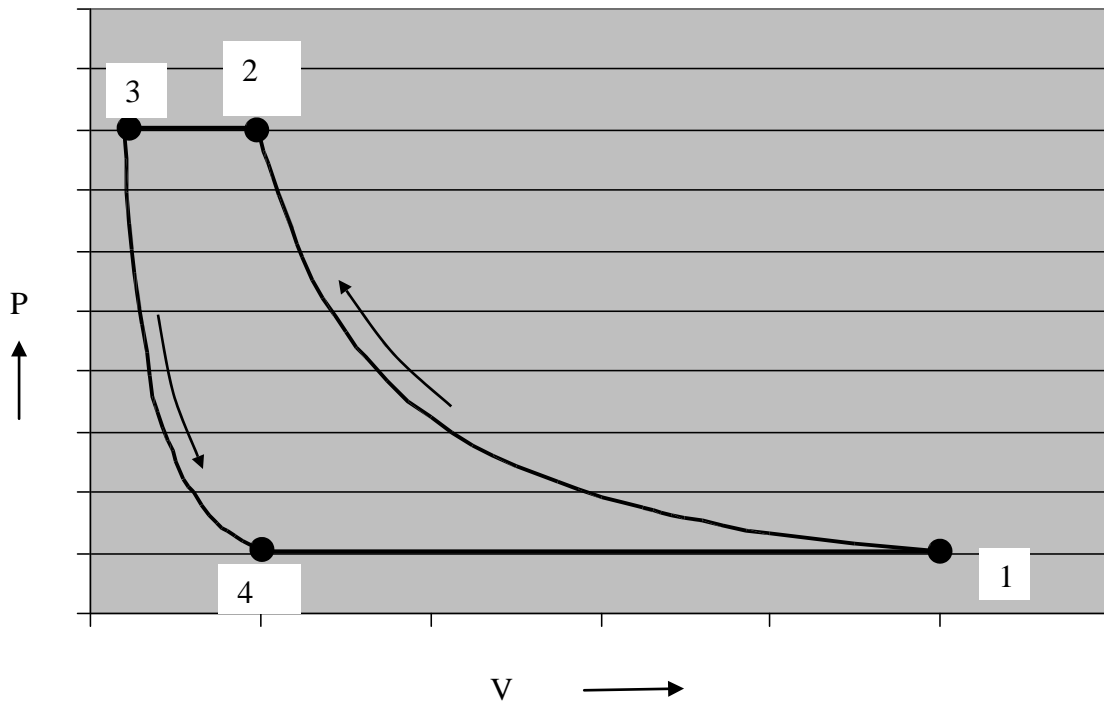


Fig.5.3 P-V diagram for Reciprocating Compressor with Clearance

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

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

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

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

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

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

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

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

So, at state 1, $p_1 V_1 = m_1 R T_1$

at state 2, $p_2 V_2 = m_1 R T_2$

at state 3, $p_3 V_3 = m_2 R T_3$ or $p_2 V_3 = m_2 R T_3$

at state 4, $p_4 V_4 = m_2 R T_4$ or $p_1 V_4 = m_2 R T_4$

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

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

Temperature and pressure can be related as,

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

Substituting

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

Substituting for constancy of temperature during suction and delivery.

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

Or

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

Thus $(m_1 - m_2)$ denotes the mass of air sucked or delivered. For unit mass of air delivered the work done per kg of air can be given as,

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

Thus from above expressions it is obvious that the clearance volume reduces the effective swept volume i.e., the mass of air handled but the work done per kg of air delivered remains unaffected.

From the cycle work estimated as above the theoretical power required for running compressor shall be,

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

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

For double acting compressor, Power

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

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

Volumetric efficiency can be overall volumetric efficiency and absolute volumetric efficiency as given below.

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

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

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

This concept is used for giving the capacity of compressor in terms of ‘free air delivery’ (FAD). “Free air delivery is the volume of air delivered being reduced to free air conditions”.

In case of air the free air delivery can be obtained using perfect gas equation as,

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

Where subscript a or p_a , V_a , T_a denote properties at free air conditions

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

This volume V_a gives ‘free air delivered’ per cycle by the compressor.

Absolute volumetric efficiency can be defined, using NTP conditions in place of free air conditions.

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

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

Here V_s is the swept volume = $V_1 - V_3$

V_c is the clearance volume = V_3

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

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

Let the ratio of clearance volume to swept volume be given by $C = \frac{V_c}{V_s}$

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

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

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

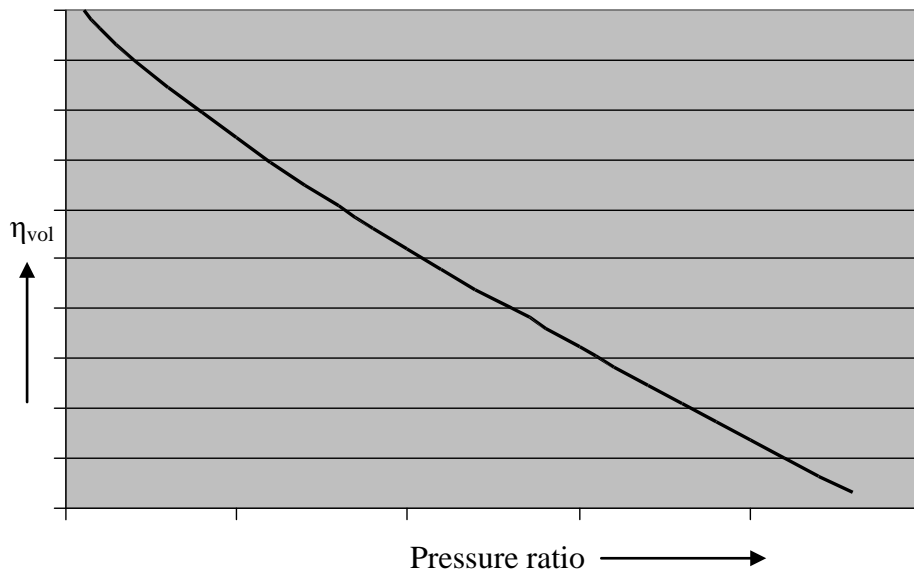


Fig.5.4 Volumetric efficiency v/s Pressure ratio

Multistage Compression

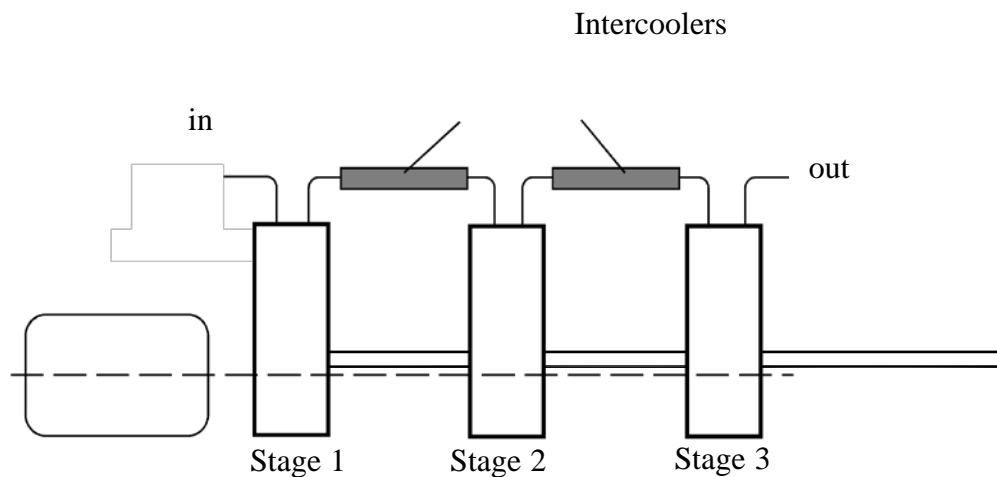


Fig.5.5 Multistage Compressor with inter coolers

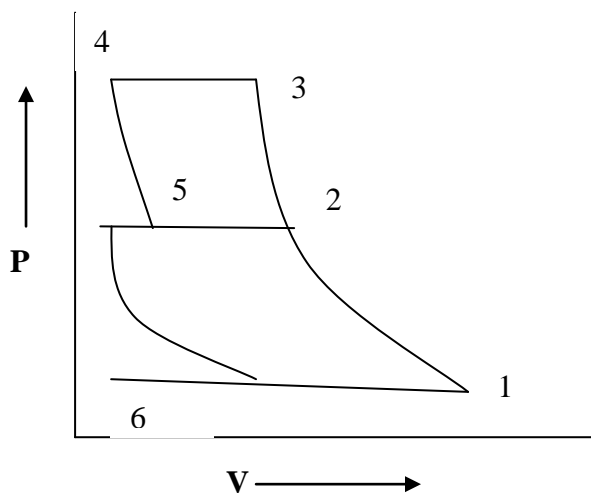


Fig.5.6 P-V diagram for Multistage Compressor

Multistage compression refers to the compression process completed in more than one stage i.e., a part of compression occurs in one cylinder and subsequently compressed air is sent to subsequent cylinders for further compression. In case it is desired to increase the compression ratio of compressor then multi-stage compression becomes inevitable. If we look at the expression for volumetric efficiency then it shows that the volumetric efficiency decreases with increase in pressure ratio. This aspect can also be explained using p-V representation shown in Fig.5.6.

Therefore, the volumetric efficiency reduces with increasing pressure ratio in compressor with single stage compression. Also for getting the same amount of free air delivery the size of cylinder is to be increased with increasing pressure ratio. The increase in pressure ratio also requires sturdy structure from mechanical strength point of view for withstanding large pressure difference.

The solution to number of difficulties discussed above lies in using the multistage compression where compression occurs in parts in different cylinders one after the other. Fig.16.6b, shows the multistage compression occurring in two stages. Here first stage of compression occurs in cycle 12671 and after first stage compression partly compressed enters second stage of compression and occurs in cycle 2345. In case of multistage compression the compression in first stage occurs at low temperature and subsequent compression in following stages occurs at higher temperature. The compression work requirement depends largely upon the average temperature during compression. Higher average temperature

during compression has larger work requirement compared to low temperature so it is always desired to keep the low average temperature during compression.

Apart from the cooling during compression the temperature of air at inlet to compressor can be reduced so as to reduce compression work. In multistage compression the partly compressed air leaving first stage is cooled upto ambient air temperature in intercooler and then sent to subsequent cylinder (stage) for compression. Thus, intercoolers when put between the stages reduce the compression work and compression is called intercooled compression. Intercooling is called perfect when temperature at inlet to subsequent stages of compression is reduced to ambient temperature. Fig.16.6c, shows multi-stage (two stage) intercooled compression. Intercooling between two stages causes temperature drop from 2 to 2' i.e discharge from first stage (at 2) is cooled upto the ambient temperature stage (at 2') which lies on isothermal compression process 1-2'-3''. In the absence of intercooling the discharge from first stage shall enter at 2. Final discharge from second stage occurs at 3' in case of intercooled compression compared to discharge at 3 in case of non-intercooled compression. Thus, intercooling offers reduced work requirement by the amount shown by area 22'3'3 on p-V diagram. If the intercooling is not perfect then the inlet state to second/subsequent stage shall not lie on the isothermal compression process line and this stage shall lie between actual discharge state from first stage and isothermal compression line.

Fig.16.7 shows the schematic of multi stage compressor (double stage) with inter cooler between stage T-s representation is shown in Fig.16.8. The total work requirement for running this shall be algebraic summation of work required for low pressure (LP) and high pressure (HP) stages. The size of HP cylinder is smaller than LP cylinder as HP cylinder handles high pressure air having smaller specific volume.

Mathematical analysis of multistage compressor is done with following assumptions:

- (i) Compression in all the stages is done following same index of compression and there is no pressure drop in suction and delivery pressures in each stage. Suction and delivery pressure remains constant in the stages.
- (ii) There is perfect intercooling between compression stages.

- (iii) Mass handled in different stages is same i.e, mass of air in LP and HP stages are same.
- (iv) Air behaves as perfect gas during compression.

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

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

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

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

$$W_{HP} = \left(\frac{n}{n-1} \right) P_2 V_{2'} \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{(n-1)}{n}} - 1 \right\}$$

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

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

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

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

Power = $W_c \times N$ - Watts

If we look at compressor work then it shows that with the initial and final pressures p_1 and P_2 , remaining same the intermediate pressure p_2 may have value floating between p_1 and P_2 , and change the work requirement W_c . Thus, the compressor work can be optimized with respect to intermediate pressure P

2. Mathematically, it can be differentiated with respect to P_2 .

$$\frac{dW_c}{dP_2} = \frac{d}{dP_2} \left[\left(\frac{n}{n-1} \right) P_1 V_1 \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_{2'}}{P_2} \right)^{\frac{n-1}{n}} - 2 \right\} \right]$$

$$\frac{dW_c}{dP_2} = \left[\left(\frac{n}{n-1} \right) P_1 V_1 \frac{d}{dP_2} \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} + \left(\frac{P_{2'}}{P_2} \right)^{\frac{n-1}{n}} - 2 \right\} \right]$$

$$\frac{dW_c}{dP_2} = \left(\frac{n}{n-1} \right) P_1 V_1 \left\{ \left(\frac{n-1}{n} \right) P_1^{\frac{1-n}{n}} \cdot P_2^{\frac{-1}{n}} - \left(\frac{n-1}{n} \right) \cdot P_{2'}^{\frac{1-n}{n}} \cdot P_2^{\frac{1-2n}{n}} \right\}$$

Equating to zero yields

$$P_1^{\frac{1-n}{n}} \cdot P_2^{\frac{-1}{n}} = P_{2'}^{\frac{1-n}{n}} \cdot P_2^{\frac{1-2n}{n}}$$

$$P_2^{\frac{-2+2n}{n}} = P_{2'}^{\frac{1-n}{n}} \cdot P_1^{\frac{n-1}{n}}$$

$$P_2^{2\left(\frac{n-1}{n}\right)} = (P_1 \cdot P_{2'})^{\left(\frac{n-1}{n}\right)}$$

$$P_2^2 = (P_1 \cdot P_{2'}), P_2 = \sqrt{P_1 \cdot P_{2'}}$$

Pressure ratio in Ist stage = Pressure ratio in IInd stage

Thus, it is established that the compressor work requirement shall be minimum when the pressure ratio in each stage is equal.

In case of multiple stages, say i number of stages, for the delivery and suction pressures of P_{i+1} and P_1 the optimum stage pressure ratio shall be,

$$\text{Optimum stage pressure ratio} = \left(\frac{P_{i+1}}{P_1} \right)^{\frac{1}{i}} \text{ for pressure at stages being } P_1, P_2, P_3, \dots, P_{i-1}, P_i,$$

P_{i+1}

Minimum work required in two stage compressor can be given by

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

For i number of stages, minimum work,

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

It also shows that for optimum pressure ratio the work required in different stages remains same for the assumptions made for present analysis. Due to pressure ratio being equal in all stages the temperature ratios and maximum temperature in each stage remains same for perfect intercooling.

Cylinder dimensions: In case of multistage compressor the dimension of cylinders can be estimated basing upon the fact that the mass flow rate of air across the stages remains same. For perfect intercooling the temperature of air at suction of each stage shall be same.

If the actual volume sucked during suction stroke is V_1, V_2, V_3, \dots for different stages they by perfect gas law, $P_1 V_1 = RT_1, P_2 V_2 = RT_2, P_3 V_3 = RT_3$

For perfect intercooling

$$P_1 V_1 = RT_1, P_2 V_2 = RT_1, P_3 V_3 = RT_1$$

$$P_1 V_1 = P_2 V_2 = RT_2, P_3 V_3 = \dots$$

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

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

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

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

Substituting,

$$P_1 \cdot \eta_{V_1} \cdot V_{1,th} = P_2 \cdot \eta_{V_2} \cdot V_{2,th} = P_3 \cdot \eta_{V_3} \cdot V_{3,th} = \dots$$

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

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

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

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

$$\text{Or } P_1 \cdot \eta_{V_1} \cdot \frac{\pi}{4} \cdot D_1^2 \cdot L_1 = P_2 \cdot \eta_{V_2} \cdot \frac{\pi}{4} \cdot D_2^2 \cdot L_2$$

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

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

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

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

Then, $D_1^2 P_1 = D_2^2 P_2 = D_3^2 P_3 = \dots$

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

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

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

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

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

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

For intercooling (Fig. 5.5) between LP and HP stage steady flow energy equation shall be;

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

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

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

For HP stage (Fig.5.5) the steady flow energy equation yields.

$$m \cdot h_{2'} + W_{HP} = m \cdot h_{3'} + Q_{HP}$$

$$Q_{HP} = W_{HP} + m(h_{2'} - h_{3'})$$

$$Q_{HP} = W_{HP} + m \cdot C_p (T_{2'} - T_{3'}) = W_{HP} - m \cdot C_p (T_{3'} - T_{2'})$$

In case of perfect intercooling and optimum pressure ratio, $T_{2'} = T_1$ and $T_2 = T_{3'}$

Hence for these conditions,

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

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

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

Total heat rejected during compression shall be the sum of heat rejected during compression and heat extracted in intercooler for perfect intercooling.

Heat rejected during compression for polytropic process $= \left(\frac{\gamma - n}{\gamma - 1} \right) \times Work$

UNIT-5 Air Compressors

Review of equations

Work done in a single stage compressor

$$= \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] = \frac{n}{n-1} mR(T_2 - T_1)$$

Work done in a two stage compressor for perfect inter cooling

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

Work done in a two stage compressor

$$\eta_v = \frac{P_1 T_a}{P_a T_1} \left[1 + c - c \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \right]$$

Volumetric Efficiency

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

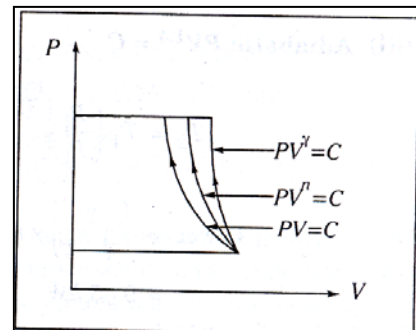
Problem 1

A single stage acting air compressor 30 cm bore and 40 cm stroke is running at a speed of 100 RPM. It takes in air at 1 bar and 20°C and drive it when the compresses it to a pressure of 5 bar. Find the power required to drive it when compression is (i) isothermal (ii) $PV^{1.2} = C$ and (iii) adiabatic. Also find the isothermal efficiencies for the cases (ii) and (iii) Neglect clearance.

$$\begin{aligned} N_1 &= 100 \text{ RPM}, d = 30 \text{ cm} \\ L &= 40 \text{ cm} \quad P_1 = 1 \text{ bar} \\ T_1 &= 20^\circ \text{C} \quad P_2 = 5 \text{ bar} \\ V_s &= \frac{\Pi}{4} d^2 L = \frac{\Pi}{4} 0.3^2 \times 0.4 \\ &= 0.028 \text{ m}^3 / \text{cycle} \end{aligned}$$

$$V_1 = V_s = 0.028 \times \frac{100}{60} = 0.047 \text{ m}^3 / \text{s}$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{1 \times 10^5 \times 0.047}{0.287 \times 293} = 0.055 \text{ kg/s}$$



(i) Isothermal $PV = C$

$$\begin{aligned}\text{Power} &= P_1 V_1 \ln \frac{P_2}{P_1} = m R T_1 \ln \frac{P_2}{P_1} \\ &= 0.055 \times 0.287 \times 293 \ln \frac{5}{1} = 7.56 \text{ kW}\end{aligned}$$

(i) $PV^{1.2} = C$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = 293 \left(\frac{5}{1} \right)^{\frac{1.2-1}{1.2}} = 383.14 \text{ K}$$

$$\begin{aligned}\text{Power} &= \frac{n}{n-1} (P_2 V_2 - P_1 V_1) = \frac{n}{n-1} \times m R (T_2 - T_1) \\ &= \frac{1.2}{1.2-1} \times 0.055 \times 0.287 (383.14 - 293) = 8.53 \text{ kW}\end{aligned}$$

$$\begin{aligned}\text{Isothermal efficiency} &= \frac{\text{Isothermal Power}}{\text{Actual power}} \\ &= \frac{7.56}{8.53} = 0.8854\end{aligned}$$

$$\eta_{\text{Isothermal}} = 88.54 \%$$

(iii) Adiabatic $PV^{1.4} = C$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = 293 \left(\frac{5}{1} \right)^{\frac{1.4-1}{1.4}} = 464 \text{ K}$$

$$\text{Isothermal efficiency} = \frac{\text{Isothermal Power}}{\text{Actual power}}$$

$$= \frac{7.56}{8.53} = 0.8854$$

$$\eta_{\text{Isothermal}} = 88.54 \%$$

$$(iii) \text{ Adiabatic } PV^{1.4} = C$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = 293 \left(\frac{5}{1} \right)^{\frac{1.4-1}{1.4}} = 464 \text{ K}$$

$$\text{Power} = \frac{\gamma}{\gamma-1} \times mR(T_2 - T_1)$$

$$= \frac{1.4}{1.4-1} \times 0.055 \times 0.287(464 - 293) = 9.45 \text{ kW}$$

$$\eta_{\text{Isothermal}} = \frac{7.56}{9.45} = 0.80$$

$$= 80 \%$$

Problem 2

A single acting stage acting air compressor with clearance running at 360 rpm has a bore of 10 cm. The compression and expansion are polytropic with $n = 1.25$ for each. The clearance volume is 80 cm^3 . If the suction and delivery pressures are 98.1 kPa and 706.32 kPa absolute, find the free air at 101 kPa and 15°C delivered per minute. What is the work done per cycle? The temperature at the beginning of compression may be taken as 30°C . Find also the power required to drive the compressor.

$$T_a = 15^\circ\text{C}, \quad d = 10 \text{ cm}$$

$$L = 8.5 \text{ cm} \quad P_1 = 98.1 \text{ kPa}$$

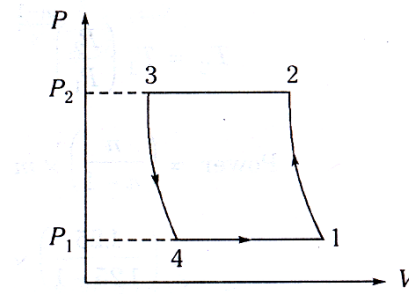
$$T_1 = 30^\circ\text{C} \quad P_2 = 706.32 \text{ kPa}$$

$$P_a = 101 \text{ kPa}$$

$$V_s = \frac{\pi}{4} d^2 L = \frac{\pi}{4} 0.1^2 \times 0.85$$

$$= 6.675 \times 10^{-4} \text{ m}^3$$

$$V_c = 80 \text{ cm}^3 = 0.8 \times 10^{-4} \text{ m}^3$$



$$\text{Clearance ratio } C = \frac{V_c}{V_s} = \frac{0.8 \times 10^{-4}}{6.675 \times 10^{-4}} = 0.12$$

Volumetric efficiency

$$\eta_v = \frac{P_1 T_a}{P_a T_1} \left[1 + c - c \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \right]$$

$$= \frac{98.1 \times 288}{101 \times 303} \left[1 + 0.12 - 0.12 \left(\frac{706.32}{98.1} \right)^{\frac{1}{1.25}} \right] = 0.496$$

$$m = \frac{P_a T_a}{R T_a} = \frac{101 \times 0.1193}{0.287 \times 288}$$

$$= 0.1457 \text{ kg/min}$$

$$\eta_v = \frac{V_a}{V_s}$$

$$\therefore \text{Volume of free air } V_a = 0.496 \times 6.675 \times 10^{-4}$$

$$= 0.0003314 \text{ m}^3 / \text{cycle}$$

$$= 0.0003314 \times 360 = 0.1193 \text{ m}^3 / \text{min}$$

$$m = \frac{0.1457}{60} = 0.00242 \text{ kg/sec}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = 303 \left(\frac{706.32}{98.1} \right)^{\frac{1.25-1}{1.25}} = 449.67 \text{ K}$$

$$\text{Power} = \left(\frac{n}{n-1} \right) \times m R (T_2 - T_1)$$

$$= \left(\frac{1.25}{1.25-1} \right) \times 0.00242 \times 0.287 (449.67 - 303)$$

$$= 0.5 \text{ kJ/s or kW}$$

$$\text{Workdone/cycle} = \frac{0.5}{360/60} = 0.0084 \text{ kJ/cycle}$$

Problem 3

A single stage double acting air compressor is required to deal with 17 m³/min of air measured at 1 bar and 15°C. the pressure and temperature at the end of suction is 0.98 bar and 30°C. The delivery pressure is 6.3bar. The rpm of the compressor is 500.assuming a clearance volume of 5% of the stroke volume, laws of the compression and expansion as $PV^{1.32}=C$, calculate the necessary stroke of volume, temperature of the air delivered and power of the compressor.

$$T_a = 15^\circ\text{C}, \quad V_a = 17\text{m}^3/\text{min}$$

$$P_a = 1\text{bar} \quad P_1 = 0.98\text{bar}$$

$$T_1 = 30^\circ\text{C} \quad P_2 = 6.3\text{bar}$$

$$N = 500\text{rpm}$$

$$C = \frac{5}{100} = 0.05$$

$$\eta_v = \frac{P_1 T_a}{P_a T_1} \left[1 + C - C \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \right]$$

$$\eta_v = \frac{98.1 \times 288}{1 \times 303} \left[1 + 0.05 - 0.05 \left(\frac{6.3}{0.98} \right)^{\frac{1}{1.32}} \right] = 0.8452$$

$$\eta_v = \frac{V_a}{V_s}$$

$$V_s = \text{Stroke Volume} = \frac{17}{0.8452} = 20.11\text{m}^3/\text{min}$$

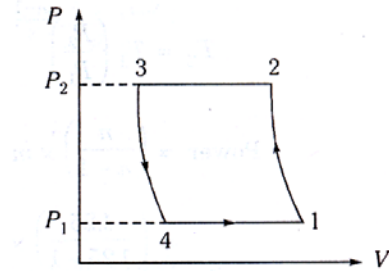
$$V_s = \frac{20.11}{500} = 0.0402\text{m}^3/\text{min}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = 303 \left(\frac{6.3}{0.98} \right)^{\frac{1.32-1}{1.32}}$$

$$\text{Temperature of air delivered} = 475.71\text{K}$$

$$m = \frac{P_a T_a}{R T_1} = \frac{1 \times 100 \times 17}{0.287 \times 288} = 20.56\text{kg/min}$$

$$= \frac{20.56}{60} = 0.3427\text{ kg/sec}$$



$$\begin{aligned}
 \text{Power} &= \left(\frac{n}{n-1} \right) \times mR(T_2 - T_1) \\
 &= \left(\frac{1.32}{1.32-1} \right) \times 0.3427 \times 0.287(495.71 - 303) \\
 &= 70 \text{ kJ/s or kW}
 \end{aligned}$$

Problem 4

A single stage double acting air compressor delivers 15 m³ of air per min of air measured at 1.013 bar and 27°C. delivers at 7bar. The condition at the end of the suction stroke are pressure 0.98 bar and temperature 4°C. The a clearance volume is 4% of the swept volume, and stroke to bore ratio is 1.3:1 and compressor runs at 300rpm. calculate the Volumetric efficiency of the compressor. Assume the index of compression and expansion to be 1.3.

$$\begin{aligned}
 T_a &= 27^\circ\text{C}, & V_a &= 15 \text{ m}^3 / \text{min} \\
 P_a &= 1.013 \text{ bar} & P_1 &= .98 \text{ bar} \\
 T_1 &= 40^\circ\text{C} & P_2 &= 7 \text{ bar} \\
 N &= 300 \text{ rpm}, \text{Clearanceratio} &= 0.04 \\
 L/d &= 1.3
 \end{aligned}$$

$$\eta_v = \frac{P_1 T_a}{P_a T_1} \left[1 + C - C \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \right]$$

$$\begin{aligned}
 \eta_v &= \frac{98.1 \times 300}{1.013 \times 313} \left[1 + 0.04 - 0.04 \left(\frac{7}{0.98} \right)^{\frac{1}{1.3}} \right] = 0.796 \\
 &= 79.6\%
 \end{aligned}$$

$$\eta_v = \frac{V_a}{V_s}$$

$$V_s = \text{Swept Volume} = \frac{15}{0.796} = 18.84 \text{ m}^3 / \text{min}$$

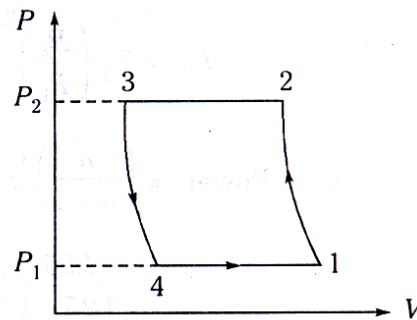
But for double acting compressor

$$V_s = \frac{\Pi}{4} d^2 \times L \times 2NL = 1.3d$$

$$18.84 = \frac{\Pi}{4} d^2 \times 1.3d \times 2 \times 300$$

$$d = \sqrt[3]{\frac{18.8 \times 44}{\Pi \times 1.3 \times 2 \times 300}} = 0.313 \text{ m} = 31.3 \text{ cm}$$

$$L = 1.3 \times 31.3 = 40.72 \text{ cm}$$



$$m = \frac{P_a T_a}{R T_a} = \frac{1.013 \times 100 \times 15}{0.287 \times 300} = 17.6 \text{ kg/min}$$

$$= \frac{17.6}{60} = 0.294 \text{ kg/s}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = 313 \left(\frac{7}{0.98} \right)^{\frac{1.3-1}{1.3}} = 492.7 \text{ K}$$

$$\text{Power} = \left(\frac{n}{n-1} \right) \times m R (T_2 - T_1)$$

$$\text{Power} = \left(\frac{1.3}{1.3-1} \right) \times 0.294 \times 0.287 (492.71 - 313)$$

$$= 65.7 \text{ kJ/s or kW}$$

$$\text{Isothermal Power} = P_1 V_1 \ln \frac{P_2}{P_1} = m R T_1 \ln \frac{P_2}{P_1}$$

$$= 0.294 \times 0.287 \times 313 \ln \frac{7}{0.98} = 51.92 \text{ kW}$$

$$\text{Isothermal efficiency} = \frac{51.92}{65.7} \times 100 = 79\%$$

Problem 5

A two-stage compressor compresses 1kg/min of air from 1bar to 42.18 bar. Initial temperature is 15°C. At the intermediate pressure the intercooling is perfect. The compression takes place according to $PV^{1.35} = C$. Neglecting the effect of clearance, determine the minimum power required to run the compressor. Also find the mass of cooling water required in the intercooler, if the temperature rise of water is limited to 5°C.

$$M = 1 \text{ kg/min} = \frac{1}{60} = 0.0166 \text{ kg/s}$$

$$P_1 = 1 \text{ bar} \quad T_1 = 15^\circ \text{C} \quad P_3 = 42.18 \text{ bar}$$

For perfect intercooling

$$P_2 = \sqrt{P_1 P_3} = \sqrt{1 \times 42.18} = 6.48 \text{ bar}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = 288 \left(\frac{6.49}{1} \right)^{\frac{1.35-1}{1.35}} = 467.7 \text{ K}$$

$$\text{Maximum Power} = \left(\frac{n}{n-1} \right) \times mR(T_2 - T_1)$$

$$= \left(\frac{2 \times 1.35}{1.35 - 1} \right) \times 0.0166 \times 0.287(467.7 - 288)$$

$$= 6.6 \text{ kJ/s or kW}$$

$$\text{Heat rejected in the intercooler } Q = mC_p(T_2 - T_1)$$

$$= 0.0166 \times 1.005(467.7 - 288)$$

$$= 2.99 \text{ kJ/Sec}$$

But Q is also = $m_{\omega} \times C_p \times \text{temperature rise}$

$$299 = m_{\omega} \times 4.187 \times 5$$

$$m_{\omega} = 0.1432 \text{ kg/sec}$$

Problem 6

A two stage reciprocating compressor delivers 150 m³/hr of free air measured at 1.03 bar and 15°C. The final pressure 18 bar. The pressure and temperature of the air in LP cylinder before compression is 1 bar and 30°C. the diameter of the LP cylinder is twice that of HP cylinder and air enters the HP cylinder at 40°C. If compression follows the law $PV^{1.22} = C$, determine

- Intermediate pressure and power required if the intercooler is imperfect.
- Ration of cylinder diameter and minimum power required for perfect intercooling.

$$T_a = 15^\circ\text{C}, \quad V_a = 150 \text{ m}^3 / \text{min}$$

$$P_a = 1.03 \text{ bar} \quad P_1 = 1 \text{ bar}$$

$$T_1 = 30^\circ\text{C} \quad P_3 = 18 \text{ bar}$$

$$T_2 = 40^\circ\text{C}$$

Neglect the effect of clearance

(i) Imperfect intercooling

$$d_{LP} = 2d_{HP}$$

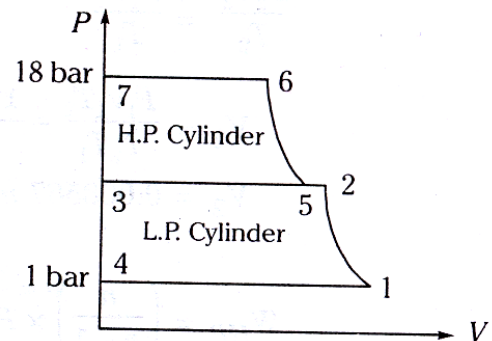
$$P_1 V_1^{1.22} = P_2 V_2^{1.22}$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2} \right)^{1.22} = \left(\frac{\Pi / 4 \times d_{LP}^2 \times L}{\Pi / 4 \times d_{HP}^2 \times L} \right)^{1.22}$$

$$\frac{P_2}{P_1} = \left(\frac{d_{LP}}{d_{HP}} \right)^{2 \times 1.22}$$

$$P_2 = 1 \times (2)^{2.44}$$

$$\text{Intercooling pressure } P_2 = 5.426 \text{ bar}$$



$$\frac{P_1 V_1}{T_1} = \frac{P_a V_a}{T_a}$$

$$V_1 = \frac{P_a V_a T_1}{T_a P_1} = \frac{1.03 \times 10^2 \times 0.02916 \times 303}{1 \times 10^2 \times 288}$$

$$= 0.045 \text{ m}^3 / \text{sec}$$

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

$$W_{LP} = \frac{1.22}{1.22-1} 1 \times 10^2 \times 0.045 \left[\left(\frac{5.426}{1} \right)^{\frac{1.22-1}{1.22}} - 1 \right]$$

$$= 8.89 \text{ kW}$$

Air enters HP cylinder at $T_2 = 40^\circ \text{C}$

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

$$V_2 = \frac{P_1 V_1 T_2}{T_1 P_2} = \frac{1 \times 10^2 \times 0.045 \times 313}{5.426 \times 10^2 \times 303}$$

$$= 0.008567 \text{ m}^3 / \text{sec}$$

$$W_{HP} = \frac{n}{n-1} P_2 V_2 \left[\left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$W_{HP} = \frac{1.22}{1.22-1} 5.426 \times 10^2 \times 0.008567 \left[\left(\frac{18}{5.426} \right)^{\frac{1.22-1}{1.22}} - 1 \right]$$

$$= 6.12 \text{ kW}$$

$$\text{Total Power} = 8.89 + 6.12$$

$$= 15.01 \text{ kW}$$

Problem 7

A multi stage compressor compressing air is to be designed to elevate the pressure from 1 bar to 120 bar such that the stage pressure ratio should not exceed 4. Determine

- (i) The number of stages
- (ii) Exact stage pressure ratio
- (iii) Intermediate pressure

Solution:

$$P_1 = 1\text{bar} \quad P_{N+1} = 120\text{bar}$$

$$\text{Stage pressure ratio} = \frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = \frac{P_{N+1}}{P_N} = 4$$

Assuming the intercooling to be perfect we have

$$\frac{P_{N+1}}{P_N} = \left(\frac{P_{N+1}}{P_1} \right)^{\frac{1}{N}}$$

$$4 = \left(\frac{120}{1} \right)^{\frac{1}{N}}$$

$$N = \frac{\ln 120}{\ln 4} = 3.453$$

\therefore Number of stages = 4

$$\text{Exact stage pressure ratio} = \frac{P_{N+1}}{P_N} = (120)^{\frac{1}{4}} = 3.31$$

$$\frac{P_5}{P_4} = 3.31, \quad P_4 = \frac{120}{3.31} = 36.25\text{ bar}$$

$$\frac{P_4}{P_3} = 3.31, \quad P_3 = \frac{36.25}{3.31} = 10.95\text{ bar}$$

$$\frac{P_3}{P_2} = 3.31, \quad P_2 = \frac{10.95}{3.31} = 3.308\text{ bar}$$

\therefore Intermediate pressures are 36.25 bar, 10.95 bar
and 3.308 bar

Problem 8

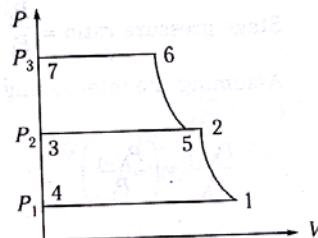
A two-stage compressor delivers air at pressure of 19 bar. The free air conditions are 1.03 bar and 25°C. The pressure of the air before compression is 0.98 bar. The intermediate pressure is 4.5 bar. The temperature of the air entering each cylinder is 35°C. The law of compression and expansion being $PV^{1.25} = C$. The clearance volume is 5% of the swept volume. Determine the volumetric efficiency and the work done per kg of air.

$$T_a = 25^\circ\text{C}, \quad P_2 = 4.5\text{bar}$$

$$P_a = 1\text{bar} \quad P_1 = 0.98\text{bar}$$

$$T_1 = 35^\circ\text{C} \quad P_3 = 19\text{bar}$$

$$\text{Clearance ratio } C = 0.05$$



$$\eta_v = \frac{P_1 T_a}{P_a T_1} \left[1 + C - C \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} \right]$$

$$\eta_v = \frac{98.1 \times 298}{1 \times 308} \left[1 + 0.05 - 0.05 \left(\frac{4.5}{0.98} \right)^{\frac{1}{1.25}} \right] = 0.835$$

$$= 83.5\%$$

Even though the temperature of air entering each cylinder is same, work is not same in both cylinders since $P_2 \neq \sqrt{P_1 P_3}$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = 308 \left(\frac{4.5}{0.98} \right)^{\frac{1.25-1}{1.25}} = 416.1 K$$

$$W_{LP} = \frac{n}{n-1} R (T_2 - T_1) = \frac{1.25}{1.25-1} \times 0.287 (416.1 - 308)$$

$$= 155.12 \text{ kJ/kg}$$

Temperature of air entering HP cylinder $T_5 = 35^\circ C$

$$T_6 = T_5 \left(\frac{P_6}{P_5} \right)^{\frac{n-1}{n}} = 308 \left(\frac{19}{4.5} \right)^{\frac{1.25-1}{1.25}} = 410.82 K$$

$$W_{HP} = \frac{n}{n-1} R (T_6 - T_5) = \frac{1.25}{1.25-1} \times 0.287 (410.82 - 308)$$

$$= 147.55 \text{ kJ/kg}$$

$$\text{Total work} = 155.12 + 147.55$$

$$= 302.75 \text{ kJ/kg of air}$$

Problem 9

A two-stage double acting air compressor takes in air at 1 bar and $25^\circ C$. It runs at 200 rpm. The diameter of LP cylinder is 35cm. The stroke of both LP and HP cylinders are 40cm. The clearance volume of both the cylinders is 4%. The index of compression is 1.3. The LP cylinder discharges air at a pressure of 4 bar. The air passes through the intercooler so that it enters the HP cylinder at $27^\circ C$ and 3.6 bar, finally it is discharged from the compressor at 14.4 bar.

Calculate

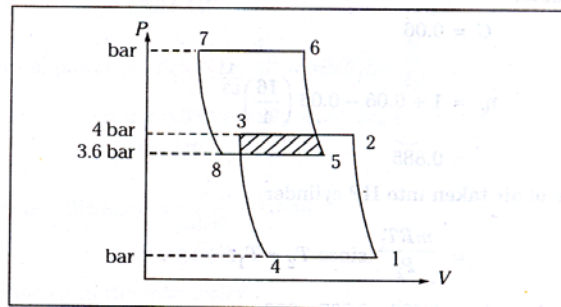
- (i) Diameter of HP cylinder
- (ii) The heat rejected in the intercooler
- (iii) The power required to drive the HP cylinder

$P_1 = 1 \text{ bar}$ $T_1 = 25^\circ\text{C}$ $N = 200$ For a double acting compressor, swept volume of LP cylinder

$$= \frac{\pi}{4} D_{LP}^2 \times L_{LP} \times \frac{2N}{60}$$

$$= \frac{\pi}{4} \times 0.35^2 \times 0.4 \times \frac{2 \times 220}{60}$$

$$V_s = 0.2822 \text{ m}^3/\text{sec}$$



Volumetric efficiency referred to the suction condition 1

$$\eta_v = 1 + C - C \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}}$$

Volume of air referred to condition 1

$$V_1 = 0.9238 \times 0.2822 = 0.26071 \text{ m}^3 / \text{s}$$

$$\text{Mass of air } m = \frac{P_1 V_1}{RT_1} = \frac{1 \times 10^2 \times 0.26071}{0.287 \times 298}$$

$$= 0.30483 \text{ kg/s}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = 298 \left(\frac{4}{1} \right)^{\frac{1.3-1}{1.3}} = 410.34 \text{ K}$$

Heat rejected in the intercooler

$$= m C_p (T_2 - T_1) = 0.30483 \times 1.005 (410.34 - 298)$$

$$= 34.14 \text{ kJ/s}$$

Volume of air draw to HP cylinder

$$V_5 = \frac{mRT_5}{P_5} = \frac{0.30483 \times 0.287 \times 298}{3.6 \times 10^2} = 0.0724 \text{ m}^3 / \text{s}$$

Pressure ratio in LP cylinder = 4

Pressure ratio in HP cylinder = $14.4/3.6=4$

Since the pressure ratio and clearance percentage in both HP and LP cylinders are the same, the volumetric efficiency of the cylinders referred to the condition at the start of compression is same.

$$\begin{aligned} \therefore \text{Swept volume of HP cylinder} &= \frac{V_5}{\eta_v} = \frac{0.0724}{0.9238} \\ &= 0.07837 \text{ m}^3 / \text{s} \end{aligned}$$

$$\frac{\pi}{4} D_{HP}^2 L_{HP} \times \frac{2 \times 200}{60} = 0.7837$$

$$\begin{aligned} \frac{\pi}{4} D_{HP}^2 \times 0.4 \times \frac{2 \times 200}{60} &= 0.1934 \text{ m} \\ &= 19.34 \text{ cm} \end{aligned}$$

Since pressure ratio is same $T_6 = T_2$

Power required for HP cylinder

$$\begin{aligned} &= \left(\frac{n}{n-1} \right) \times mR(T_6 - T_5) \\ &= \left(\frac{n}{n-1} \right) \times mR(T_2 - T_1) \\ &= \left(\frac{1.3}{1.3-1} \right) \times 0.30483 \times 0.287 (410.34 - 298) \\ &= 42.58 \text{ kW} \end{aligned}$$

Problem 10

A two stage air compressor compresses air from 17°C and 1 bar to 63 bar. The air is cooled in the intercooler to 30°C and intermediate pressure is steady at 7.7 bar. The low pressure cylinder is 10 cm diameter and the stroke for both cylinders is 11.25 cm. Assuming a compression law of $PV^{1.35} = \text{constant}$, and that the volume of air at atmospheric conditions drawn in per stroke is equal to the low pressure cylinder

swept volume, find the power of the compressor while running at 250 rpm. Find also the diameter of HP cylinder.

Solution:

$$\begin{aligned}d_{LP} &= 10\text{cm} \quad L = 11.25\text{cm} \\P_1 &= 1 \text{ bar} \quad P_2 = 7.7 \text{ bar} \quad T_1 = 17^\circ\text{C} \\P_3 &= 63 \text{ bar}\end{aligned}$$

Volume of LP cylinder

$$V_1 = \pi/4 \times 0.1^2 \times 0.1125 = 0.00088 \text{ m}^3$$

$$m = P_1 V_1 / R T_1 = (1 \times 100 \times 0.00088) / (0.287 \times 290) = 0.00106 \text{ kg}$$

$$T_2 = 30^\circ\text{C}$$

Volume of air entering the HP cylinder

$$V_2 = m R T_2 / P_2 = (0.00106 \times 0.283 \times 303) / (7.7 \times 10^2) = 0.0001198 \text{ m}^3$$

$$\begin{aligned}V_2 &= \pi/4 \times d_2^2 \times L \\0.0001198 &= \pi/4 \times d_2^2 \times 0.1125 \\d_2 &= \text{diameter of HP cylinder} \\&= 0.0368 \text{ m}\end{aligned}$$

$$\text{Diameter of HP cylinder} = 3.68 \text{ cm}$$

$$\text{Work required/cycle} = (n/n-1) [P_1 V_1 \{ (P_2 / P_1)^{n-1/n} - 1 \} + P_2 V_2 (P_3 / P_2)^{n-1/n} - 1]$$

$$W = (1.35/1.35-1) [1 \times 10^2 \times 0.00088 \{ (7.7/1)^{1.35-1/1.35} - 1 \} + 7.7 \times 10^2 \times 0.0001198 \{ (63/7.7)^{1.35-1/1.35} - 1 \}]$$

$$= 0.49456 \text{ kJ/cycle}$$

$$\text{Power} = 0.49456 \times \text{Number of cycles/sec}$$

$$= 0.49456 \times 250/60$$

$$\text{Power} = 2.06 \text{ kW}$$

Problem 11

A three stage air compressor draws $8 \text{ m}^3/\text{min}$ of air at 1 bar and 18°C and delivers the same at 55 bar and 20°C . The index of compression is 1.32. The air while passing through the intercoolers and aftercoolers suffers a pressure loss of 4% and is cooled to the initial temperature. Determine the shaft power required to drive the compressor if mechanical efficiency is 85%.

Solution:

$$\begin{aligned}V_1 &= 8 \text{ m}^3/\text{min} = 8/60 = 0.133 \text{ m}^3/\text{sec} \\P_1 &= 1 \text{ bar} \quad T_1 = 18^\circ\text{C} \quad P_4 = 55 \text{ bar}\end{aligned}$$

The pressure drop of 4% in the intercooler is accounted for by the factor $C = 0.96$

For 3 stage compressor,

$$\begin{aligned}\text{Power} &= 3(n/n-1) \times P_1 V_1 [\{P_4/C^3 P_1\}^{n-1/3n} - 1] \\ &= 3 (1.32/1.32-1) \times 1 \times 10^2 \times 0.133 [\{55/0.96^3 \times 1\}^{1.32-1/1.32} - 1] \\ &= 283.32 \text{ kW}\end{aligned}$$

$$\text{Actual shaft power required} = \text{Power}/\eta_{\text{mech}} = 283.32/0.85 = \mathbf{333.32 \text{ kW}}$$

STEAM NOZZLES

8.1 Introduction

In the impulse steam turbine, the overall transformation of heat into mechanical work is accomplished in two distinct steps. The available energy of steam is first changed into kinetic energy, and this kinetic energy is then transformed into mechanical work. The first of these steps, viz., the transformation of available energy into kinetic energy is dealt with in this chapter.

A nozzle is a passage of varying cross-sectional area in which the potential energy of the steam is converted into kinetic energy. The increase of velocity of the steam jet at the exit of the nozzle is obtained due to decrease in enthalpy (total heat content) of the steam. The nozzle is so shaped that it will perform this conversion of energy with minimum loss.

8.2 General Forms of Nozzle Passages

A nozzle is an element whose primary function is to convert enthalpy (total heat) energy into kinetic energy. When the steam flows through a suitably shaped nozzle from zone of high pressure to one at low pressure, its velocity and specific volume both will increase.

The equation of the continuity of mass may be written thus :

$$\dots(8.1)$$

where m mass flow in kg/sec.,

V = velocity of steam in m/sec.,

A = area of cross-section in m^2 , and

v = specific volume of steam in m^3/kg .

In order to allow the expansion to take place properly, the area at any section of the nozzle must be such that it will accommodate the steam whatever volume and velocity may prevail at that point.

As the mass flow (m) is same at all sections of the nozzle, area of cross-section (A) varies as $\frac{1}{Vv}$. The manner in which both V and v vary depends upon the properties of the substance flowing. Hence, the contour of the passage of nozzle depends upon the nature of the substance flowing.

For example, consider a *liquid*- a substance whose specific volume v remains almost constant with change of pressure. The value of V will go on increasing with change of pressure. Thus, from eqn. (8.1), the area of cross-section should decrease with the decrease of pressure. Fig. 8-1(a) illustrates the proper contour of longitudinal section of

a nozzle suitable for liquid. This also can represent convergent nozzle for a fluid whose peculiarity is that while both velocity and specific volume increase, the rate of specific volume increase is less than that of the velocity, thus resulting in increasing value of $\frac{V}{v}$.

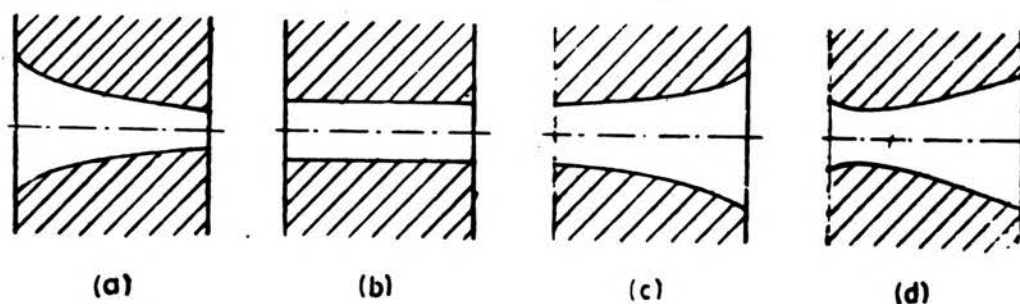


Fig. 8-1. General forms of Nozzles.

Fig. 8-1(b) represents the correct contour for some hypothetical substance for which both velocity and specific volume increase at the same rate, so that their ratio $\frac{V}{v}$ is a constant at all points. The area of cross-section should therefore, be constant at all points, and the nozzle becomes a plain tube.

Fig. 8-1(c) represents a divergent nozzle for a fluid whose peculiarity is that $\frac{V}{v}$ decreases with the drop of pressure, i.e., specific volume increases at a faster rate than velocity with the drop of pressure. The area of cross-section should increase as the pressure decreases.

Table 8-1

Properties of steam at various pressures when expanding dry saturated steam from 14 bar to 0.15 bar through a nozzle, assuming frictionless adiabatic flow.

Pressure p bar	Dryness fraction x	Enthalpy drop $H_1 - H_2$ kJ	Velocity V m/sec.	Specific Volume v_s m^3/kg	Discharge per unit area kg/m^2	Area A m^2	Diameter D metre
14	1.000	-	-	-	-	-	-
12	0.988	38.6	278	0.1633	1,723	0.00058	0.0272
10	0.974	84.1	410	0.1944	2,165	0.00046	0.0242
7	0.950	164.7	574	0.2729	*2,214	0.00045	x0.0239
3.5	0.908	309	786	0.5243	1,651	0.00061	0.0279
1.5	0.872	441.2	939	1.1593	929	0.0011	0.0374
0.70	0.840	555.6	1,054	2.365	531	0.00188	0.049
0.15	0.790	736.7	1,214	10.022	153	0.0065	0.091

* Maximum discharge per unit area

x Smallest diameter

Fig. 8-1(d) shows the general shape of convergent-divergent nozzle suitable for gases and vapours. It can be shown that in practice, while velocity and specific volume both increase from the start, velocity first increases faster than the specific volume, but after

UNIT 7 IC ENGINE TESTING

Structure

- 7.1 Introduction
 - Objectives
- 7.2 Performance Measurements
- 7.3 Basic Parameters
 - 7.3.1 Measurement of Speed
 - 7.3.2 Fuel Consumption Measurement
 - 7.3.3 Measurement of Air Consumption
 - 7.3.4 Measurement of Exhaust Smoke
- 7.4 Measurement of Exhaust Emission
- 7.5 Measurement of Brake Power
- 7.6 Measurement of Friction Horse Power
- 7.7 Blowby Loss
- 7.8 Performance of SI Engines
- 7.9 Performance of CI Engines
- 7.10 Summary
- 7.11 Key Words
- 7.12 Answers to SAQs

7.1 INTRODUCTION

At a design and development stage an engineer would design an engine with certain aims in his mind. The aims may include the variables like indicated power, brake power, brake specific fuel consumption, exhaust emissions, cooling of engine, maintenance free operation etc. The other task of the development engineer is to reduce the cost and improve power output and reliability of an engine. In trying to achieve these goals he has to try various design concepts. After the design the parts of the engine are manufactured for the dimensions and surface finish and may be with certain tolerances. In order to verify the designed and developed engine one has to go for testing and performance evaluation of the engines.

Thus, in general, a development engineer will have to conduct a wide variety of engine tests starting from simple fuel and air-flow measurements to taking of complicated injector needle lift diagrams, swirl patterns and photographs of the burning process in the combustion chamber. The nature and the type of the tests to be conducted depend upon various factors, some of which are: the degree of development of the particular design, the accuracy required, the funds available, the nature of the manufacturing company, and its design strategy. In this chapter, only certain basic tests and measurements will be considered.

Objectives

After studying this unit, you should be able to

- understand the performance parameters in evaluation of IC engine performance,
- calculate the speed of IC engine, fuel consumption, air consumption, etc.,
- evaluate the exhaust smoke and exhaust emission, and
- differentiate between the performance of SI engine and CI engines.

7.2 PERFORMANCE PARAMETERS

Engine performance is an indication of the degree of success of the engine performs its assigned task, i.e. the conversion of the chemical energy contained in the fuel into the useful mechanical work. The performance of an engine is evaluated on the basis of the following :

- (a) Specific Fuel Consumption.
- (b) Brake Mean Effective Pressure.
- (c) Specific Power Output.
- (d) Specific Weight.
- (e) Exhaust Smoke and Other Emissions.

The particular application of the engine decides the relative importance of these performance parameters.

For Example : For an aircraft engine specific weight is more important whereas for an industrial engine specific fuel consumption is more important.

For the evaluation of an engine performance few more parameters are chosen and the effect of various operating conditions, design concepts and modifications on these parameters are studied. The basic performance parameters are the following :

- (a) Power and Mechanical Efficiency.
- (b) Mean Effective Pressure and Torque.
- (c) Specific Output.
- (d) Volumetric Efficiency.
- (e) Fuel-air Ratio.
- (f) Specific Fuel Consumption.
- (g) Thermal Efficiency and Heat Balance.
- (h) Exhaust Smoke and Other Emissions.
- (i) Specific Weight.

Power and Mechanical Efficiency

The main purpose of running an engine is to obtain mechanical power.

- Power is defined as the rate of doing work and is equal to the product of force and linear velocity or the product of torque and angular velocity.
- Thus, the measurement of power involves the measurement of force (or torque) as well as speed. The force or torque is measured with the help of a dynamometer and the speed by a tachometer.

The power developed by an engine and measured at the output shaft is called the brake power (bp) and is given by,

$$bp = \frac{2\pi NT}{60} \quad \dots (7.1)$$

where, T is torque in N-m and N is the rotational speed in revolutions per minute.

The total power developed by combustion of fuel in the combustion chamber is, however, more than the bp and is called indicated power (ip). Of the power developed by the engine, i.e. ip , some power is consumed in overcoming the friction between moving parts, some in the process of inducing the air and removing the products of combustion from the engine combustion chamber.

It is the power developed in the cylinder and thus, forms the basis of evaluation of combustion efficiency or the heat release in the cylinder.

$$IP = \frac{p_{im} LANk}{60}$$

where, p_m = Mean effective pressure, N/m²,

L = Length of the stroke, m,

A = Area of the piston, m²,

N = Rotational speed of the engine, rpm (It is $N/2$ for four stroke engine), and

k = Number of cylinders.

Thus, we see that for a given engine the power output can be measured in terms of mean effective pressure.

The difference between the ip and bp is the indication of the power lost in the mechanical components of the engine (due to friction) and forms the basis of mechanical efficiency; which is defined as follows :

$$\text{Mechanical efficiency} = \frac{bp}{ip} \quad \dots (7.2)$$

The difference between ip and bp is called friction power (fp).

$$fp = ip - bp \quad \dots (7.3)$$

$$\therefore \text{Mechanical efficiency} = \frac{bp}{(bp + fp)} \quad \dots (7.4)$$

Mean Effective Pressure and Torque

Mean effective pressure is defined as a hypothetical/average pressure which is assumed to be acting on the piston throughout the power stroke. Therefore,

$$p_m = \frac{ip \times 60}{LANk} \quad \dots (7.5)$$

where, P_m = Mean effective pressure, N/m²,

I_p = Indicated power, Watt,

L = Length of the stroke, m,

A = Area of the piston, m²,

N = Rotational speed of the engine, rpm (It is $N/2$ for four stroke engine), and

k = Number of cylinders.

If the mean effective pressure is based on bp it is called the brake mean effective pressure ($bmeP$ P_{mb} replace ip by bp in Eq. 5.5), and if based on ihp it is called indicated mean effective pressure ($imeP$). Similarly, the friction mean effective pressure ($fmeP$) can be defined as,

$$fmeP = imeP - bmeP \quad \dots (7.6)$$

The torque is related to mean effective pressure by the relation

$$bp = \frac{2\pi NT}{60} \quad \dots (7.7)$$

$$ip = \frac{p_{im} LANk}{60}$$

By Eq. (5.5),

$$\frac{2\pi NT}{60} = \left(b_{emp} \cdot A \cdot L \cdot \frac{Nk}{60} \right)$$

$$\text{or, } T = \frac{(b_{emp} \cdot A \cdot L \cdot k)}{2\pi} \quad \dots (7.8)$$

Thus, the torque and the mean effective pressure are related by the engine size. A large engine produces more torque for the same mean effective pressure. For this reason, torque is not the measure of the ability of an engine to utilize its displacement for producing power from fuel. It is the mean effective pressure which gives an indication of engine displacement utilization for this conversion. Higher the mean effective pressure, higher will be the power developed by the engine for a given displacement.

Again we see that the power of an engine is dependent on its size and speed. Therefore, it is not possible to compare engines on the basis of either power or torque. Mean effective pressure is the true indication of the relative performance of different engines.

Specific Output

Specific output of an engine is defined as the brake power (output) per unit of piston displacement and is given by,

$$\begin{aligned} \text{Specific output} &= \frac{bp}{A \times L} \\ &= \text{Constant} \times b_{mep} \times rpm \end{aligned} \quad \dots (7.9)$$

- The specific output consists of two elements – the b_{mep} (force) available to work and the speed with which it is working.
- Therefore, for the same piston displacement and b_{mep} an engine operating at higher speed will give more output.
- It is clear that the output of an engine can be increased by increasing either speed or b_{mep} . Increasing speed involves increase in the mechanical stress of various engine parts whereas increasing b_{mep} requires better heat release and more load on engine cylinder.

Volumetric Efficiency

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

Volumetric efficiency, η_v

$$= \frac{\text{Mass of charge actually sucked in}}{\text{Mass of charge corresponding to the cylinder intake } P \text{ and } T \text{ conditions}} \quad \dots (5.10)$$

The amount of air taken inside the cylinder is dependent on the volumetric efficiency of an engine and hence puts a limit on the amount of fuel which can be efficiently burned and the power output.

For supercharged engine the volumetric efficiency has no meaning as it comes out to be more than unity.

Fuel-air ratio (F/A) is the ratio of the mass of fuel to the mass of air in the fuel-air mixture. Air-fuel ratio (A/F) is reciprocal of fuel-air ratio. Fuel-air ratio of the mixture affects the combustion phenomenon in that it determines the flame propagation velocity, the heat release in the combustion chamber, the maximum temperature and the completeness of combustion.

Relative fuel-air ratio is defined as the ratio of the actual fuel-air ratio to that of the stoichiometric fuel-air ratio required to burn the fuel supplied. Stoichiometric fuel-air ratio is the ratio of fuel to air is one in which case fuel is completely burned due to minimum quantity of air supplied.

$$\text{Relative fuel-air ratio, } F_R = \frac{\text{Actual fuel - Air ratio}}{\text{Stoichiometric fuel - Air ratio}} \quad \dots (7.11)$$

Brake Specific Fuel Consumption

Specific fuel consumption is defined as the amount of fuel consumed for each unit of brake power developed per hour. It is a clear indication of the efficiency with which the engine develops power from fuel.

$$\text{Brake specific fuel consumption (bsfc)} = \frac{\text{Actual fuel - Air ratio}}{\text{Stoichiometric fuel - Air ratio}} \quad \dots (7.12)$$

This parameter is widely used to compare the performance of different engines.

Thermal Efficiency and Heat Balance

Thermal efficiency of an engine is defined as the ratio of the output to that of the chemical energy input in the form of fuel supply. It may be based on brake or indicated output. It is the true indication of the efficiency with which the chemical energy of fuel (input) is converted into mechanical work. Thermal efficiency also accounts for combustion efficiency, i.e., for the fact that whole of the chemical energy of the fuel is not converted into heat energy during combustion.

$$\text{Brake thermal efficiency} = \frac{bp}{m_f \times C_v} \quad \dots (7.13)$$

where, C_v = Calorific value of fuel, kJ/kg, and

m_f = Mass of fuel supplied, kg/sec.

- The energy input to the engine goes out in various forms – a part is in the form of brake output, a part into exhaust, and the rest is taken by cooling water and the lubricating oil.
- The break-up of the total energy input into these different parts is called the heat balance.
- The main components in a heat balance are brake output, coolant losses, heat going to exhaust, radiation and other losses.
- Preparation of heat balance sheet gives us an idea about the amount of energy wasted in various parts and allows us to think of methods to reduce the losses so incurred.

Exhaust Smoke and Other Emissions

Smoke and other exhaust emissions such as oxides of nitrogen, unburned hydrocarbons, etc. are nuisance for the public environment. With increasing emphasis on air pollution control all efforts are being made to keep them as minimum as it could be.

Smoke is an indication of incomplete combustion. It limits the output of an engine if air pollution control is the consideration.

Exhaust emissions have of late become a matter of grave concern and with the enforcement of legislation on air pollution in many countries; it has become necessary to view them as performance parameters.

Specific Weight

Specific weight is defined as the weight of the engine in kilogram for each brake power developed and is an indication of the engine bulk. Specific weight plays an important role in applications such as power plants for aircrafts.

7.3 BASIC MEASUREMENTS

The basic measurements to be undertaken to evaluate the performance of an engine on almost all tests are the following :

- (a) Speed
- (b) Fuel consumption
- (c) Air consumption
- (d) Smoke density
- (e) Brake horse-power
- (f) Indicated horse power and friction horse power
- (g) Heat going to cooling water
- (h) Heat going to exhaust
- (i) Exhaust gas analysis.

In addition to above a large number of other measurements may be necessary depending upon the aim of the test.

7.3.1 Measurement of Speed

One of the basic measurements is that of speed. A wide variety of speed measuring devices are available in the market. They range from a mechanical tachometer to digital and triggered electrical tachometers.

The best method of measuring speed is to count the number of revolutions in a given time. This gives an accurate measurement of speed. Many engines are fitted with such revolution counters.

A mechanical tachometer or an electrical tachometer can also be used for measuring the speed.

The electrical tachometer has a three-phase permanent-magnet alternator to which a voltmeter is attached. The output of the alternator is a linear function of the speed and is directly indicated on the voltmeter dial.

Both electrical and mechanical types of tachometers are affected by the temperature variations and are not very accurate. For accurate and continuous measurement of speed a magnetic pick-up placed near a toothed wheel coupled to the engine shaft can be used. The magnetic pick-up will produce a pulse for every revolution and a pulse counter will accurately measure the speed.

7.3.2 Fuel Consumption Measurement

Fuel consumption is measured in two ways :

- (a) The fuel consumption of an engine is measured by determining the volume flow in a given time interval and multiplying it by the specific gravity of the fuel which should be measured occasionally to get an accurate value.
- (b) Another method is to measure the time required for consumption of a given mass of fuel.

Accurate measurement of fuel consumption is very important in engine testing work.

As already mentioned two basic types of fuel measurement methods are :

- Volumetric type
- Gravimetric type.

Volumetric type flowmeter includes Burette method, Automatic Burette flowmeter and Turbine flowmeter.

Gravimetric Fuel Flow Measurement

The efficiency of an engine is related to the kilograms of fuel which are consumed and not the number of litres. The method of measuring volume flow and then correcting it for specific gravity variations is quite inconvenient and inherently limited in accuracy. Instead if the weight of the fuel consumed is directly measured a great improvement in accuracy and cost can be obtained.

There are three types of gravimetric type systems which are commercially available include Actual weighing of fuel consumed, Four Orifice Flowmeter, etc.

7.3.3 Measurement of Air Consumption

One can say the mixture of air and fuel is the food for an engine. For finding out the performance of the engine accurate measurement of both is essential.

In IC engines, the satisfactory measurement of air consumption is quite difficult because the flow is pulsating, due to the cyclic nature of the engine and because the air a compressible fluid. Therefore, the simple method of using an orifice in the induction pipe is not satisfactory since the reading will be pulsating and unreliable.

All kinetic flow-inferring systems such as nozzles, orifices and venturies have a square law relationship between flow rate and differential pressure which gives rise to severe errors on unsteady flow. Pulsation produced errors are roughly inversely proportional to the pressure across the orifice for a given set of flow conditions. The various methods and meters used for air flow measurement include

- (a) Air box method, and
- (b) Viscous-flow air meter.

7.3.4 Measurement of Exhaust Smoke

All the three widely used smokemeters, namely, Bosch, Hartridge, and PHS are basically soot density (g/m^3) measuring devices, that is, the meter readings are a function of the mass of carbon in a given volume of exhaust gas.

Hartridge smokemeter works on the light extinction principle.

The basic principles of the Bosch smokemeter is one in which a fixed quantity of exhaust gas is passed through a fixed filter paper and the density of the smoke stains on the paper are evaluated optically. In a recent modification of this type of smokemeter units are used for the measurement of the intensity of smoke stain on filter paper.

In Von Brand smokemeter which can give a continuous reading a filter tape is continuously moved at a uniform rate to which the exhaust from the engine is fed. The smoke stains developed on the filter paper are sensed by a recording head. The single obtained from the recording head is calibrated to give smoke density.

7.4 MEASUREMENT OF EXHAUST EMISSION

Substances which are emitted to the atmosphere from any opening of the exhaust port of the engine are termed as exhaust emissions. If combustion is complete and the mixture is

stoichiometric the products of combustion would consist of carbon dioxide (CO_2) and water vapour only.

However, there is no complete combustion of fuel and hence the exhaust gas consists of variety of components, the most important of them are carbon monoxide (CO), unburned hydrocarbons (UBHC) and oxides of nitrogen (NO_x). Some oxygen and other inert gases would also be present in the exhaust gas.

Over the decade numerous devices have been developed for measuring these various exhaust components. A brief discussion of some of the more commonly used instruments is given below.

7.4.1 Flame Ionization Detector (FID)

The schematic diagram of a flame ionization detector burner is shown in Figures 7.1(a) and (b) shows burner.

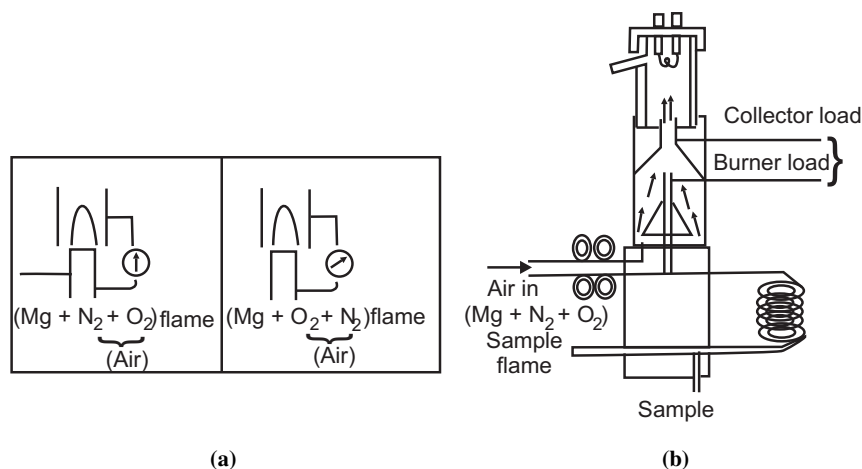


Figure 7.1 : Flame Ionization Detector Burner

The working principle of this burner is as follows: A hydrogen-air flame contains a negligible amount of ions. However, if even trace amounts of an organic compound such as HC are introduced into the flame, a large number of ions are produced. If a polarized voltage is applied across the burner jet and an adjacent collector, an ion migration will produce a current proportional to the number of ions and thus to the HC concentration present in the flame.

The output of the FID depends on the number of carbon atoms passing through the flame in a unit time. Doubling the flow velocity would also double the output. Hexane (C_6H_{14}) would give double the output of propane (C_3H_8). Therefore, FID output is usually referred to a standard hydrocarbon, usually as ppm of normal hexane.

Presences of CO, CO_2 , NO_x , water and nitrogen in the exhaust have to effect on the FID reading. Oxygen slightly affects the reading of FID.

FID analyzer is a rapid, continuous and accurate method of measuring HC in the exhaust gas. Concentration as low as 1 ppb can be measured.

7.4.2 Spectroscopic Analyzers

- A spectrum shows the light absorbed as a function of wavelength (or frequency).
- Each compound shows a different spectrum for the light absorbed.
- All the spectroscopic analyzers work on the principle that the quantity of energy absorbed by a compound in a sample cell is proportional to the concentration of the compound in the cell. There are two types of spectroscopic analyzers.

These analyzers use only a narrow dispersed frequency of light spectrum to analyze a compound. These are usually not use for exhaust emission measurements.

Non-Dispersive Infra-red (NDIR) Analyzers

In the NDIR analyzer the exhaust gas species being measuring is used to detect itself. This is done by selective absorption. The infrared energy of a particular wavelength or frequency is peculiar to a certain gas in that the gas will absorb the infracted energy of this wavelength and transmit and infrared energy of other wavelengths. For example, the absorption band for carbon monoxide is between 4.5 and 5 microns. So the energy absorbed at this wavelength is an indication of the concentration of CO in the exhaust gas.

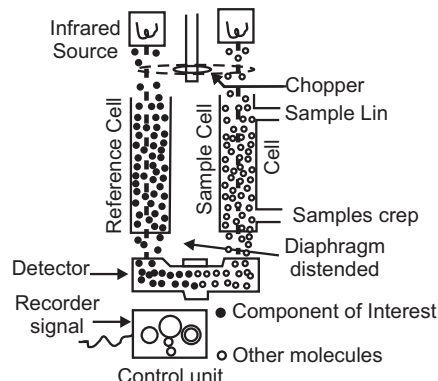


Figure 7.2 : Schematic of Non-dispersive Infra-red Analyzer (NDIR)

The NDIR analyzer as shown in Figure 7.2 consists of two infrared sources, interrupted simultaneously by an optical chopper. Radiation from these sources passes in parallel paths through a reference cell and a sample cell to opposite side of a common detector. The sample cell contains the compounds to be analyzed, whereas this compound is not present in the reference cell. The latter is usually filled with an inert gas, usually nitrogen, which does not absorb the infrared energy for the wavelength corresponding to the compound being measured. A closed container filled with only the compound to be measured works as a detector.

The detector is divided into two equal volumes by a thin metallic diaphragm. When the chopper blocks the radiation, the pressure in both parts of the detector is same and the diagram remains in the neutral position. As the chopper blocks and unblocks the radiation, the radiant energy from one source passes through the reference cell unchanged whereas the sample cell absorbs the infrared energy at the wavelength of the compound in cell. The absorption is proportional to the concentration of the compound to be measured in the sample cell. Thus unequal amounts of energy are transmitted to the two volumes of the detector and the pressure differential so generated causes movement of the diaphragm and a fixed probe, thereby generating an a.c., displayed on a meter. The signal is a function of the concentration of the compound to be measured.

The NDIR can accurately measure CO, CO₂ and those hydrocarbons which have clear infrared absorption peaks. However, usually the exhaust sample to be analyzed contains other species which also absorb infrared energy at the same frequency. For example, an NDIR analyzer sensitized to n-hexane for detection of HC responds equally well to other paraffin HC but not to olefins, acetylenes or aromatics. Therefore, the reading given by such analyzer is multiplied by 1.8 to correct it to the total UBHC as measured by an FID analyzer in the same exhaust stream.

7.4.3 Gas Chromatography

Gas chromatography is first a method of separating the individual constituents of a mixture and then a method of assured their concentration. After separation, each

compound can be separately analyzed for concentration. This is the only method by which each component existing in an exhaust sample can be identified and analyzed. However, it is very time consuming and the samples can be taken only in batches. Gas chromatograph is primarily a laboratory tool.

In addition to the above methods such as mass spectroscopy, chemiluminescent analyzers, and electrochemical analyzer are also used for measuring exhaust emissions.

7.5 MEASUREMENT OF BRAKE POWER

The brake power measurement involves the determination of the torque and the angular speed of the engine output shaft. The torque measuring device is called a dynamometer.

Dynamometers can be broadly classified into two main types, power absorption dynamometers and transmission dynamometer.

Figure 7.3 shows the basic principle of a dynamometer. A rotor driven by the engine under test is electrically, hydraulically or magnetically coupled to a stator. For every revolution of the shaft, the rotor periphery moves through a distance $2\pi r$ against the coupling force F . Hence, the work done per revolution is .

$$W = 2 \pi R F$$

The external moment or torque is equal to $S \times L$ where, S is the scale reading and L is the arm. This moment balances the turning moment $R \times F$, i.e.

$$S \times L = R \times F$$

$$\therefore \text{Work done/revolution} = 2\pi SL$$

$$\text{Work done/minute} = 2\pi SLN$$

where, N is rpm. Hence, power is given by

$$\text{Brake power } P = 2\pi NT$$

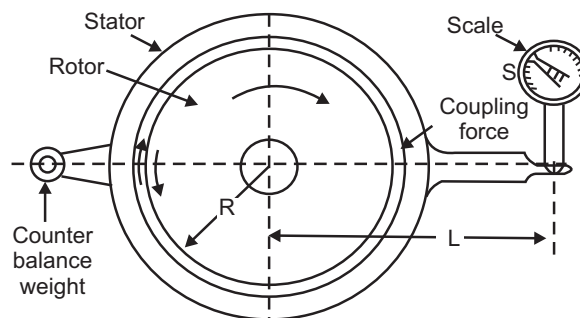


Figure7.3 : Principle of a Dynamometer

Absorption Dynamometers

These dynamometers measure and absorb the power output of the engine to which they are coupled. The power absorbed is usually dissipated as heat by some means. Example of such dynamometers is prony brake, rope brake, hydraulic dynamometer, etc.

Transmission Dynamometers

In transmission dynamometers, the power is transmitted to the load coupled to the engine after it is indicated on some type of scale. These are also called torque-meters.

7.5.1 Absorption Dynamometers

These include Prony brake type, Rope brake type, and Hydraulic type.

Prony Brake

One of the simplest methods of measuring brake power (output) is to attempt to stop the engine by means of a brake on the flywheel and measure the weight which an arm attached to the brake will support, as it tries to rotate with the flywheel. This system is known as the prony brake and forms its use; the expression brake power has come.

The Prony brake shown in Figure 7.4 works on the principle of converting power into heat by dry friction. It consists of a wooden block mounted on a flexible rope or band. The wooden block when pressed into contact with the rotating drum takes the engine torque and the power is dissipated in frictional resistance. Spring-loaded bolts are provided to tighten the wooden block and hence increase the friction. The whole of the power absorbed is converted into heat and hence this type of dynamometer must be cooled. The brake horsepower is given by

$$BP = 2\pi NT$$

where, $T = W \times l$

W being the weight applied at a radius l .

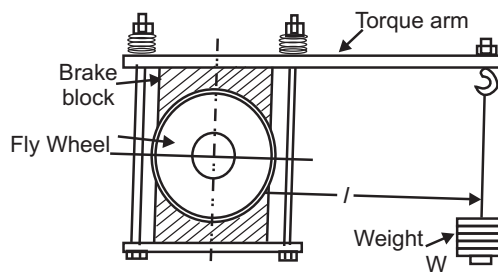


Figure 7.4 : Prony Brake

Rope Brake

The rope brake as shown in Figure 7.5 is another simple device for measuring bp of an engine. It consists of a number of turns of rope wound around the rotating drum attached to the output shaft. One side of the rope is connected to a spring balance and the other to a loading device. The power is absorbed in friction between the rope and the drum. The drum therefore requires cooling.

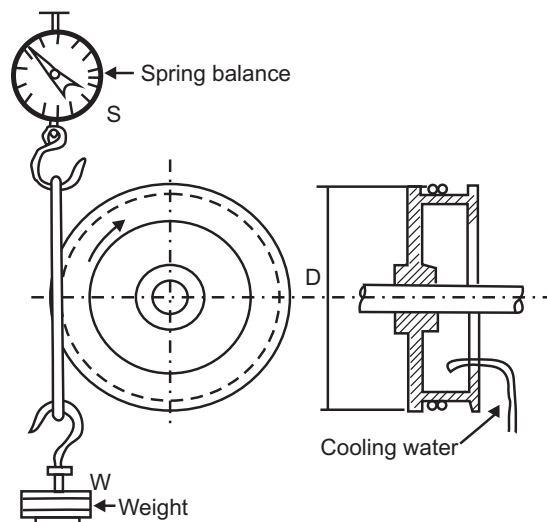


Figure 7.5 : Rope Brake

Rope brake is cheap and easily constructed but not a very accurate method because of changes in the friction coefficient of the rope with temperature.

The bp is given by

$$bp = \pi DN (W - S)$$

where, D is the brake drum diameter, W is the weight in Newton and S is the spring scale reading.

Hydraulic Dynamometer

Hydraulic dynamometer shown in Figure 7.6 works on the principle of dissipating the power in fluid friction rather than in dry friction.

- In principle its construction is similar to that of a fluid flywheel.
- It consists of an inner rotating member or impeller coupled to the output shaft of the engine.
- This impeller rotates in a casing filled with fluid.
- This outer casing, due to the centrifugal force developed, tends to revolve with the impeller, but is resisted by a torque arm supporting the balance weight.
- The frictional forces between the impeller and the fluid are measured by the spring-balance fitted on the casing.
- The heat developed due to dissipation of power is carried away by a continuous supply of the working fluid, usually water.
- The output can be controlled by regulating the sluice gates which can be moved in and out to partially or wholly obstruct the flow of water between impeller, and the casing.

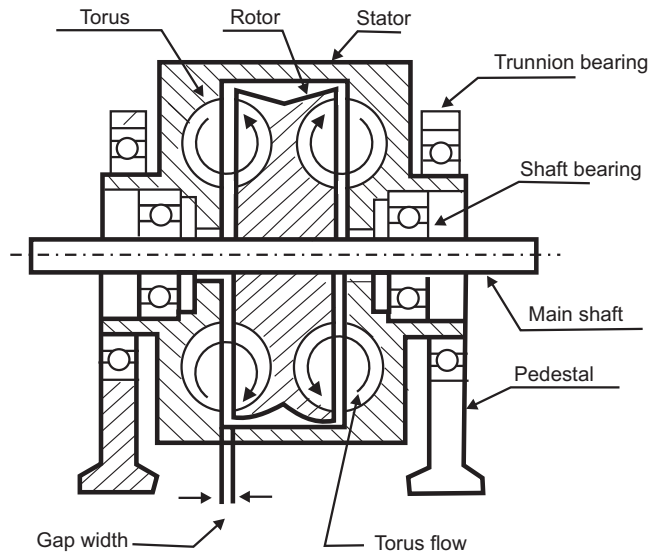


Figure 7.6 : Hydraulic Dynamometer

Eddy Current Dynamometer

The working principle of eddy current dynamometer is shown in Figure 7.7.

It consists of a stator on which are fitted a number of electromagnets and a rotor disc made of copper or steel and coupled to the output shaft of the engine. When the rotor rotates eddy currents are produced in the stator due to magnetic flux set up by the passage of field current in the electromagnets. These eddy currents are dissipated in producing heat so that this type of dynamometer also requires some cooling arrangement. The torque is measured exactly as in other types of

absorption dynamometers, i.e. with the help of a moment arm. The load is controlled by regulating the current in the electromagnets.

The following are the main advantages of eddy current dynamometers :

- (a) High brake power per unit weight of dynamometer.
- (b) They offer the highest ratio of constant power speed range (up to 5 : 1).
- (c) Level of field excitation is below 1% of total power being handled by dynamometer, thus, easy to control and programme.
- (d) Development of eddy current is smooth hence the torque is also smooth and continuous under all conditions.
- (e) Relatively higher torque under low speed conditions.
- (f) It has no intricate rotating parts except shaft bearing.
- (g) No natural limit to size-either small or large.

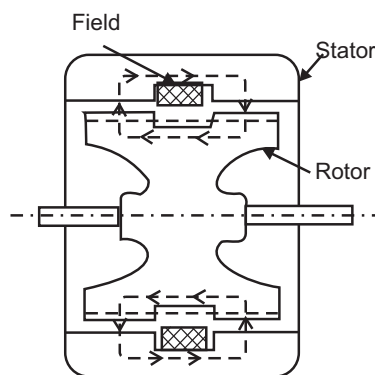


Figure 7.7 : Eddy Current Dynamometer

Swinging Field d.c. Dynamometer

Basically, a swinging field d.c. dynamometer is a d.c. shunt motor so supported on trunnion bearings to measure their action torque that the outer case and field coils tend to rotate with the magnetic drag. Hence, the name swinging field. The torque is measured with an arm and weighing equipment in the usual manner.

Many dynamometers are provided with suitable electric connections to run as motor also. Then the dynamometer is reversible, i.e. works as motoring as well as power absorbing device.

- When used as an absorption dynamometer it works as a d.c. generator and converts mechanical energy into electric energy which is dissipated in an external resistor or fed back to the mains.
- When used as a motoring device an external source of d.c. voltage is needed to drive the motor.

The load is controlled by changing the field current.

7.5.2 Fan Dynamometer

It is also an absorption type of dynamometer in that when driven by the engine it absorbs the engine power. Such dynamometers are useful mainly for rough testing and running-in. The accuracy of the fan dynamometer is very poor. The power absorbed is determined by using previous calibration of the fan brake.

7.5.3 Transmission Dynamometers

Transmission dynamometers, also called torque meters, mostly consist of a set of strain-gauges fixed on the rotating shaft and the torque is measured by the angular deformation of the shaft which is indicated as strain of the strain gauge. Usually, a four

arm bridge is used to reduce the effect of temperature to minimum and the gauges are arranged in pairs such that the effect of axial or transverse load on the strain gauges is avoided.

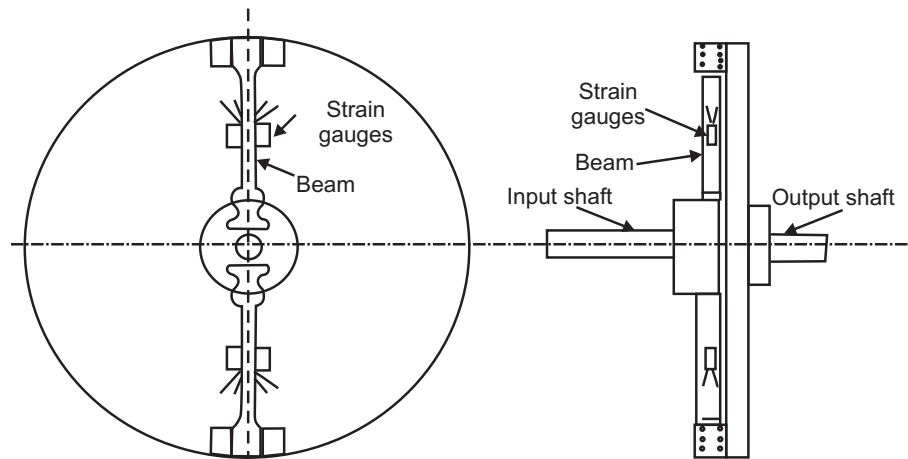


Figure 7.8 : Transmission Dynamometer

Figure 7.8 shows a transmission dynamometer which employs beams and strain-gauges for a sensing torque.

Transmission dynamometers are very accurate and are used where continuous transmission of load is necessary. These are used mainly in automatic units.

7.6 MEASUREMENT OF FRICTION HORSE POWER

- The difference between indicated power and the brake power output of an engine is the friction power.
- Almost invariably, the difference between a good engine and a bad engine is due to difference between their frictional losses.
- The frictional losses are ultimately dissipated to the cooling system (and exhaust) as they appear in the form of frictional heat and this influences the cooling capacity required. Moreover, lower friction means availability of more brake power; hence brake specific fuel consumption is lower.
- The *bsfc* rises with an increase in speed and at some speed it renders the sue of engine prohibitive. Thus, the level of friction decides the maximum output of the engine which can be obtained economically.

In the design and testing of an engine; measurement of friction power is important for getting an insight into the methods by which the output of an engine can be increased. In the evaluation of *ip* and mechanical efficiency measured friction power is also used.

The friction force power of an engine is determined by the following methods :

- Willan's line method.
- Morse test.
- Motoring test.
- Difference between *ip* and *bp*.

Willan's Line Method or Fuel Rate Extrapolation

In this method, gross fuel consumption vs. *bp* at a constant speed is plotted and the graph is extrapolated back to zero fuel consumption as illustrated in Figure 7.9.

The point where this graph cuts the *bp* axis in an indication of the friction power of the engine at that speed. This negative work represents the combined loss due to mechanical friction, pumping and blowby.

The test is applicable only to compression ignition engines.

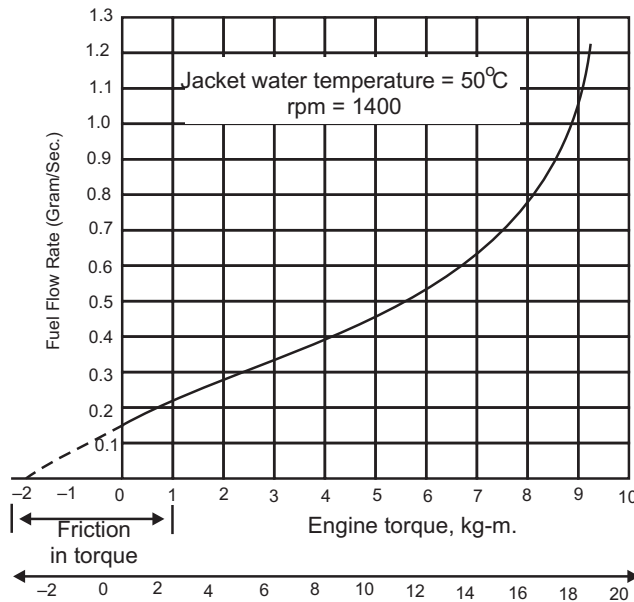


Figure 7.9 : Willan's Line Method

- The main drawback of this method is the long distance to be extrapolated from data measured between 5 and 40% load towards the zero line of fuel in put.
- The directional margin of error is rather wide because of the graph which may not be a straight line many times.
- The changing slope along the curve indicates part efficiencies of increments of fuel. The pronounced change in the slope of this line near full load reflects the limiting influence of the air-fuel ratio and of the quality of combustion.
- Similarly, there is a slight curvature at light loads. This is perhaps due to difficulty in injecting accurately and consistently very small quantities of fuel per cycle.
- Therefore, it is essential that great care should be taken at light loads to establish the true nature of the curve.
- The Willan's line for a swirl-chamber CI engine is straighter than that for a direct injection type engine.
- The accuracy obtained in this method is good and compares favorably with other methods if extrapolation is carefully done.

Morse Test

The Morse test is applicable only to multicylinder engines.

- In this test, the engine is first run at the required speed and the output is measured.
- Then, one cylinder is cut out by short circuiting the spark plug or by disconnecting the injector as the case may be.
- Under this condition all other cylinders 'motor' this cut-out cylinder. The output is measured by keeping the speed constant at its original value.
- The difference in the outputs is a measure of the indicated horse power of the cut-out cylinder.
- Thus, for each cylinder the ip is obtained and is added together to find the total ip of the engine.

The ip of n cylinder is given by

$$ip_n = bp_n + fp \quad \dots (7.17)$$

ip for $(n - 1)$ cylinders is given by

$$ip_{n-1} = bp_{n-1} + fp \quad \dots (7.18)$$

Since, the engine is running at the same speed it is quite reasonable to assume that fp remains constant.

From Eqs. (7.17) and (7.18), we see that the ihp of the n^{th} cylinder is given by

$$(ip) n^{\text{th}} = bp_n - bp_{n-1} \quad \dots (7.19)$$

and the total ip of the engine is,

$$hp_n = \Sigma (ihp) n^{\text{th}} \quad \dots (7.20)$$

By subtracting bp_n from this, fp of the engine can be obtained.

This method though gives reasonably accurate results and is liable to errors due to changes in mixture distribution and other conditions by cutting-out one cylinder. In gasoline engines, where there is a common manifold for two or more cylinders the mixture distribution as well as the volumetric efficiency both change. Again, almost all engines have a common exhaust manifold for all cylinders and cutting-out of one cylinder may greatly affect the pulsations in exhaust system which may significantly change the engine performance by imposing different back pressures.

Motoring Test

- In the motoring test, the engine is first run up to the desired speed by its own power and allowed to remain at the given speed and load conditions for some time so that oil, water, and engine component temperatures reach stable conditions.
- The power of the engine during this period is absorbed by a swinging field type electric dynamometer, which is most suitable for this test.
- The fuel supply is then cut-off and by suitable electric-switching devices the dynamometer is converted to run as a motor to drive for 'motor' the engine at the same speed at which it was previously running.
- The power supply to the motor is measured which is a measure of the fhp of the engine. During the motoring test the water supply is also cut-off so that the actual operating temperatures are maintained.
- This method, though determines the fp at temperature conditions very near to the actual operating temperatures at the test speed and load, does, not give the true losses occurring under firing conditions due to the following reasons.
 - (a) The temperatures in the motored engine are different from those in a firing engine because even if water circulation is stopped the incoming air cools the cylinder. This reduces the lubricating oil temperature and increases friction increasing the oil viscosity. This problem is much more severe in air-cooled engines.
 - (b) The pressure on the bearings and piston rings is lower than the firing pressure. Load on main and connecting rod bearings are lower.
 - (c) The clearance between piston and cylinder wall is more (due to cooling). This reduces the piston friction.
 - (d) The air is drawn at a temperature less than when the engine is firing because it does not get heat from the cylinder (rather loses heat to the cylinder). This makes the expansion line to be lower than the compression line on the p - v diagram. This loss is however counted in the indicator diagram.

- (e) During exhaust the back pressure is more because under motoring conditions sufficient pressure difference is not available to impart gases the kinetic energy is necessary to expel them from exhaust.

Motoring method, however, gives reasonably good results and is very suitable for finding the losses due to various engine components. This insight into the losses caused by various components and other parameters is obtained by progressive stripping-off of the under progressive dismantling conditions keeping water and oil circulation intact. Then the cylinder head can be removed to evaluate, by difference, the compression loss. In this manner piston ring, piston etc. can be removed and evaluated for their effect on overall friction.

Difference between ip and bp

- (a) The method of finding the fp by computing the difference between ip , as obtained from an indicator diagram, and bp , as obtained by a dynamometer, is the ideal method. However, due to difficulties.
- (b) In obtaining accurate indicator diagrams, especially at high engine speeds, this method is usually only used in research laboratories. Its use at commercial level is very limited.

Comments on Methods of Measuring fp

- The Willan's line method and Morse tests are very cheap and easy to conduct.
- However, both these tests give only an overall idea of the losses whereas motoring test gives a very good insight into the various causes of losses and is a much more powerful tool.
- As far as accuracy is concerned the $ip - bp$ method is the most accurate if carefully done.
- Motoring method usually gives a higher value for fhp as compared to that given by the Willian's line method.

7.7 BLOWBY LOSS

Blowby is the escape of unburned air-fuel mixture and burned gases from the combustion chamber, past the piston rings, and into the crank-case. High blowby is quite harmful in that it results in higher ring temperatures and contamination of lubricating oil.

7.8 PERFORMANCE OF SI ENGINES

The performance of an engine is usually studied by heat balance-sheet. The main components of the heat balance are :

- Heat equivalent to the effective (brake) work of the engine,
- Heat rejected to the cooling medium,
- Heat carried away from the engine with the exhaust gases, and
- Unaccounted losses.

The unaccounted losses include the radiation losses from the various parts of the engine and heat lost due to incomplete combustion. The friction loss is not shown as a separate item to the heat balance-sheet as the friction loss ultimately reappears as heat in cooling water, exhaust and radiation.

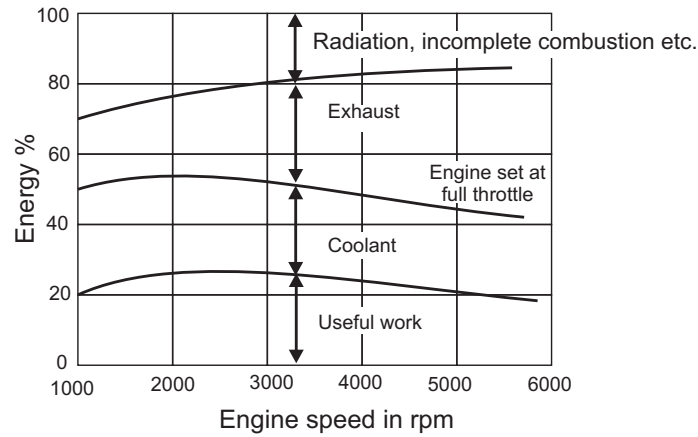


Figure 7.10 : Heat Balance Vs. Speed for a Petrol Engine at Full Throttle

The following Table 7.1 gives the approximate percentage values of various losses in SI and CI engines.

Table 7.1 : Components of Heat Balance in Percent at Full Load

Engine Type	Brake Load Efficiency %	Heat Rejected to Cooling Water %	Heat Rejected through Exhaust Gases %	Unaccounted Heat %
SI	21-28	12-27	30-55	3-55 (including incomplete combustion loss 0-45)
CI	29-42	15-35	25-45	21-0 (including incomplete combustion loss 0-5)

Figure 7.10 shows the heat balance for a petrol engine run at full throttle over its speed range. In SI engines, the loss due to incomplete combustion included on unaccounted form can be rather high. For a rich mixture (A/F ratio = 12.5 to 13) it could be 20%. Figure 7.11 shows the heat balance of uncontrolled Otto engine at different loads.

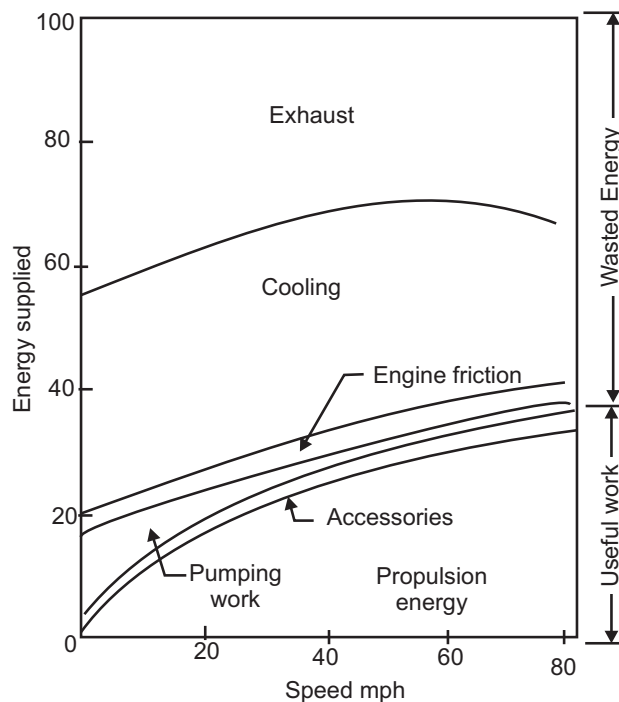


Figure 7.11 : Uncontrolled Otto Engine

Figure 7.12 shows the brake thermal efficiency, indicated thermal efficiency, mechanical efficiency and specific fuel consumption for the above SI engine.

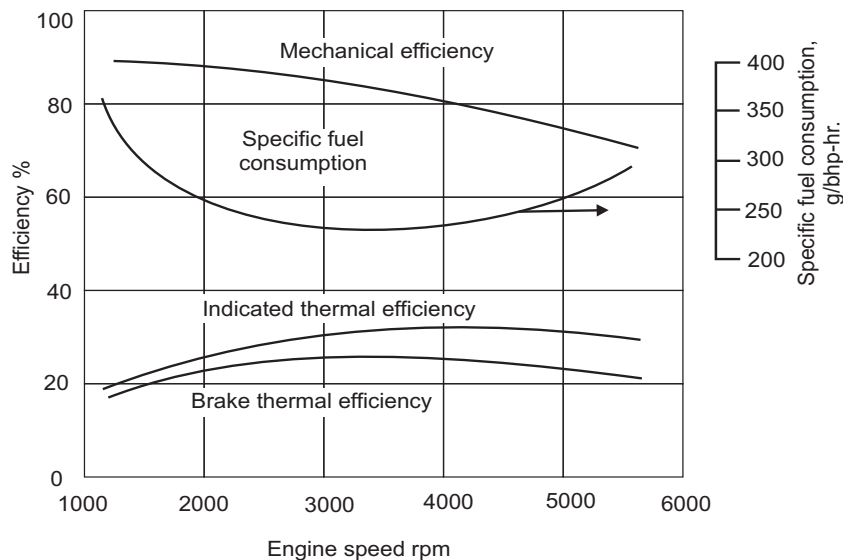


Figure 7.12 : Efficiency and Specific Fuel Consumption Vs.

Figure 7.13 shows the ip , bp , fp (by difference) brake torque, brake mean effective pressure and brake specific fuel consumption of a high compression ratio (9) automotive SI engine at full or Wide Open Throttle (W.O.T.).

Speed for a Petrol Engine at Full Throttle

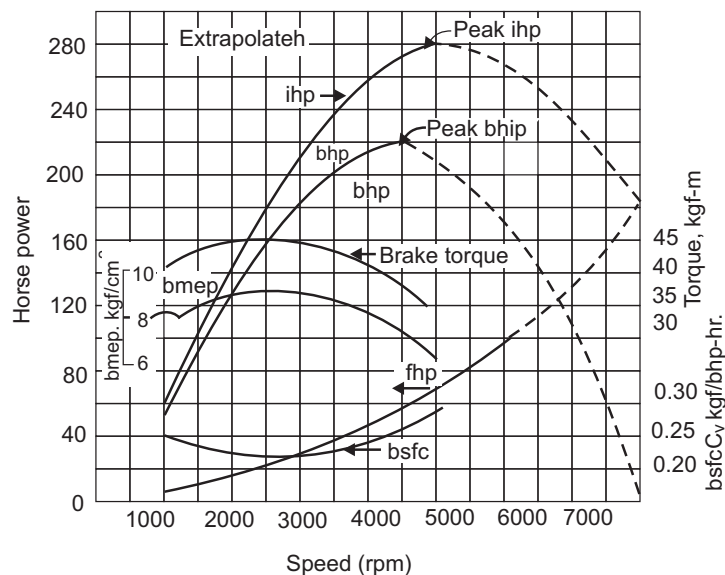


Figure 7.13 : Variable Speed Test of Automotive SI Engine at Full Throttle ($CR = 9$)

Referring to the Figure 7.10 through Figure 7.13 the following conclusions can be drawn :

- At full throttle the brake thermal efficiency at various speeds varies from 20 to 27 percent, maximum efficiency being at the middle speed range.
- The percentage heat rejected to coolant is more at lower speed (≈ 35 percent) and reduces at higher speeds (≈ 25 percent). Considerably more heat is carried by exhaust at higher speeds.
- Torque and mean effective pressure do not strongly depend on the speed of the engine, but depend on the volumetric efficiency and friction losses. Maximum torque position corresponds with the maximum air charge or minimum volumetric efficiency position.

Torque and mep curves peak at about half that of the brake-power.

Note : If size (displacement) of the engine were to be doubled, torque would also double, but mean effective pressure (mep) is a 'specific' torque, a variable independent of the size of the engine.

- (d) High power arises from the high speed. In the speed range before the maximum power is obtained, doubling the speed doubles the power.
- (e) At low engine speed the friction power is relatively low and bhp is nearly as large as ip (Figure 7.13). As engine speed increases, however, fp increases at continuously greater rate and therefore bp reaches a peak and starts reducing even though ip is rising. At engine speeds above the usual operating range, fp increases very rapidly. Also, at these higher speeds ip will reach a maximum and then fall off. At some point, ip and fp will be equal, and bp will then drop to zero.

Performance of SI Engine at Constant Speed and Variable Load

The performance of SI engine at constant speed and variable loads is different from the performance at full throttle and variable speed. Figure 7.14 shows the heat balance of SI engine at constant speed and Figure 7.14 variable load. The load is varied by altering the throttle and the speed is kept constant by resetting the dynamometer.

Closing the throttle reduces the pressure inside the cylinders but the temperature is affected very little because the air/fuel ratio is substantially constant, and the gas temperatures throughout the cycle are high. This results in high loss to coolant at low engine load. This is reason of poor part load thermal efficiency of the SI engine compared with the CI engine.

- At low loads the efficiency is about 10 percent, rising to about 25 percent at full load.
- The loss to coolant is about 60 percent at low loads and 30 percent at full load.
- The exhaust temperature rises very slowly with load and as mass flow rate of exhaust gas is reduced because the mass flow rate of fuel into the engine is reduced, the percentage loss to exhaust remains nearly constant (about 21% at low loads to 24% at full load).
- Percentage loss to radiation increases from about 7% at loads or 20% at full load.

7.9 PERFORMANCE OF CI ENGINES

The performance of a CI engine at constant speed variable load is shown in Figure 7.15.

- As the efficiency of e^{th} CI engine is more than the SI engine the total losses are less. The coolant loss is more at low loads and radiation, etc. losses are more at high loads.
- The bme_p , bp and torque directly increase with load, as shown in Figure 7.16. Unlike the SI engine bhp and bme_p are continuously raising curves and are limited only by the load. The lowest brake specific fuel consumption and hence the maximum efficiency occurs at about 80 percent of the full load.

Figure 7.17 shows the performance curves of variable speed GM 7850 cc. four cycle V-6 Toro-flow diesel engine. The maximum torque value is at about 70 percent of maximum speed compared to about 50 percent in the SI engine. Also, the $bsfc$ is low through most of the speed range for the diesel engine and is better than the SI engine.

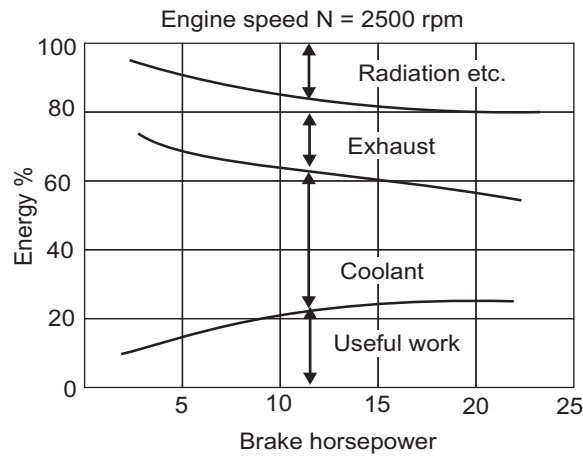


Figure 7.14 : Heat Balance Vs. Load for a Petrol Engine

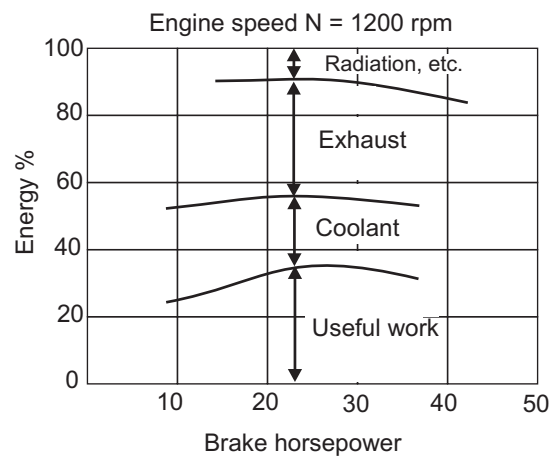


Figure 7.15 : Heat Balance Vs. Load for a CI Engine

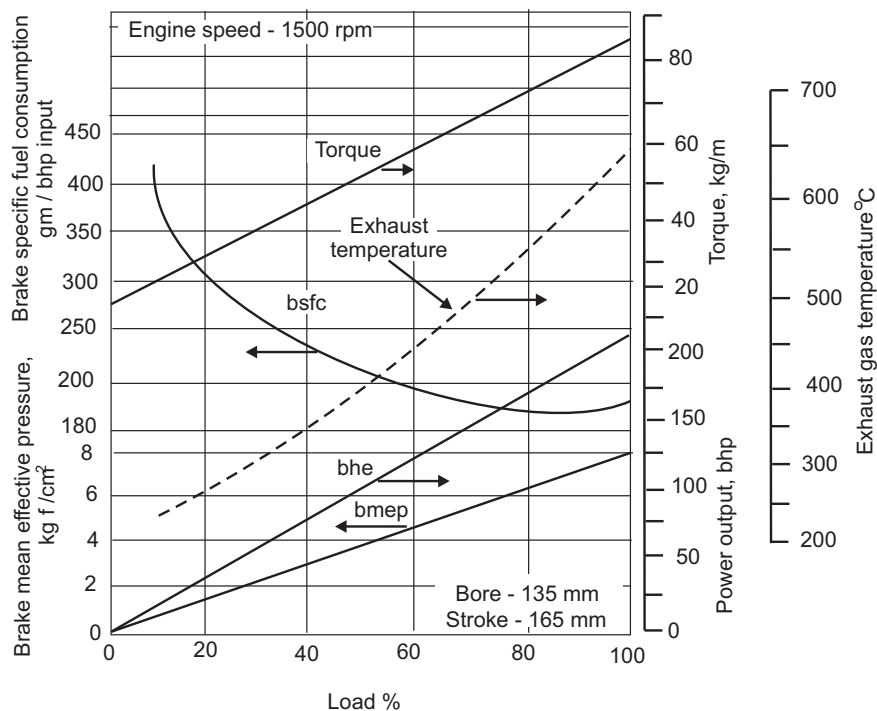


Figure 7.16 : Performance Curves of a Six Cylinder Four-stroke Cycle Automotive Type CI Engine at Constant Speed

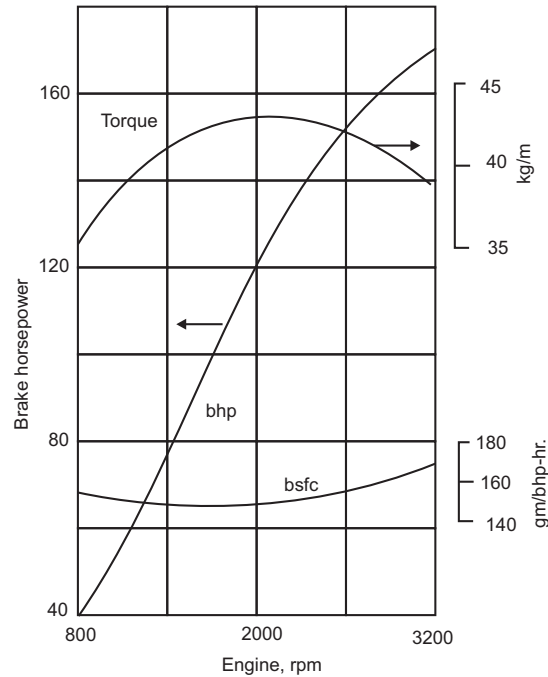


Figure 7.17 : Performance Curves of GM-four Cycle Toro-flow Diesel Engine

Example 7.1

A gasoline engine works on Otto cycle. It consumes 8 litres of gasoline per hour and develops power at the rate of 25 kW. The specific gravity of gasoline is 0.8 and its calorific value is 44000 kJ/kg. Find the indicated thermal efficiency of the engine.

Solution

Heat liberated at the input

$$\begin{aligned}
 &= m C_v \\
 &= 8 \times \frac{0.8}{60 \times 60} \\
 &= \frac{6.4}{3600}
 \end{aligned}$$

$$\text{Power at the input} = \frac{6.4}{3600} \times 44000 \text{ kW}$$

$$\begin{aligned}
 \eta_{\text{ith}} &= \frac{\text{Output power}}{\text{Input power}} \\
 &= \frac{25}{\frac{6.4 \times 44000}{3600}} \\
 &= \frac{25 \times 3600}{6.4 \times 44000} = 0.3196
 \end{aligned}$$

$$\text{or,} \quad = 31.96\%$$

Example 7.2

A single cylinder engine operating at 2000 rpm develops a torque of 8 N-m. The indicated power of the engine is 2.0 kW. Find loss due to friction as the percentage of brake power.

Solution**IC Engine Testing**

$$\begin{aligned}\text{Brake power} &= \frac{2\pi NT}{60000} = \frac{2 \times \pi \times 2000 \times 8}{60000} \\ &= 1.6746 \text{ kW}\end{aligned}$$

$$\begin{aligned}\text{Friction power} &= 2.0 - 1.6746 \\ &= 0.3253\end{aligned}$$

$$\% \text{ loss} = \frac{0.3253}{2} \times 100$$

$$\% \text{ loss} = 16.2667\%$$

Example 7.3

A diesel engine consumes fuel at the rate of 5.5 gm/sec. and develops a power of 75 kW. If the mechanical efficiency is 85%. Calculate *bsfc* and *isfc*. The lower heating value of the fuel is 44 MJ/kg.

Solution

$$\begin{aligned}b_{sec} &= \frac{\text{kW heat input}}{\text{kW heat output}} \\ &= \frac{C_v \times m_f}{P} = C_v \times bsfc\end{aligned}$$

$$bsfc = \frac{5.55}{75} = 0.074 \text{ g/kWs}$$

$$= 0.074 \times 10^{-3} \text{ kg/kWs}$$

$$C_v = 44 \text{ MJ/kg} = 44 \times 10^3 \text{ kJ/kg}$$

$$b_{sec} = bsfc \times C_v = 44 \times 10^3 \times 0.074 \times 10^{-3} = 3.256$$

$$i_{sec} = b_{sec} \times \eta_n = 3.256 \times 0.85$$

$$i_{sec} = 2.7676.$$

Example 7.4

Find the air-fuel ratio of a 4-stroke, 1 cylinder, air cooled engine with fuel consumption time for 10 cc as 20.0 sec. and air consumption time for 0.1 m³ as 16.3 sec. The load is 16 kg at speed of 3000 rpm. Also find brake specific fuel consumption in g/kWh and thermal brake efficiency. Assume the density of air as 1.175 kg/m³ and specific gravity of fuel to be 0.7. The lower heating value of fuel is 44 MJ/kg and the dynamometer constant is 5000.

Solution

$$\text{Air consumption} = \frac{0.1}{16.3} \times 1.175 = 7.21 \times 10^{-3} \text{ kg/s}$$

$$\text{Fuel consumption} = \frac{10}{20} \times 0.7 \times \frac{1}{1000} = 0.35 \times 10^{-3} \text{ kg/s}$$

$$\text{Air-fuel ratio} = \frac{7.21 \times 10^{-3}}{0.35 \times 10^{-3}} = 20.6$$

$$\begin{aligned}\text{Power output (P)} &= \frac{WN}{\text{Dynamometer constant}} \\ &= \frac{16 \times 3000}{5000} = 9.6 \text{ kW}\end{aligned}$$

$$bsfc = \frac{\text{Fuel consumption (h/hr)}}{\text{Power output}}$$

$$= \frac{0.35 \times 10^{-3} \times 3600 \times 1000}{9.6}$$

$$bsfc = 131.25 \text{ g/kWh}$$

$$= \frac{9.6}{0.35 \times 10^{-3} \times 44000} = 100$$

$$\eta_{bth} = 62.3377$$

Example 7.5

A six-cylinder, gasoline engine operates on the four-stroke cycle. The bore of each cylinder is 80 mm and the stroke is 100 mm. The clearance volume per cylinder is 70 cc. At the speed of 4100 rpm, the fuel consumption is 5.5 gm/sec. [or 19.8 kg/hr.) and the torque developed is 160 Nm.

Calculate : (i) Brake power, (ii) The brake mean effective pressure, (iii) Brake thermal efficiency if the calorific value of the fuel is 44000 kJ/kg and (iv) The relative efficiency on a brake power basis assuming the engine works on the constant volume cycle $r = 1.4$ for air.

Solution

$$bp = \frac{2\pi NT}{60000} = \frac{2 \times \pi \times 4100 \times 160}{60000} = 68.66$$

$$P_{bm} = \frac{bp \times 6000}{LANK}$$

$$= \frac{68.66 \times 60000}{0.1 \times \frac{\pi}{4} \times (0.08)^2 \times \frac{4100}{2} \times 6}$$

$$= 6.66 \times 10^5 \text{ Pa}$$

$$P_{bm} = 6.66 \text{ bar}$$

$$\eta_{bth} = \frac{bp}{m_f \times C_v} = \frac{68.66 \times 3600}{19.8 \times 43000} \times 100 = 29.03\%$$

$$\text{Compression ratio, } r = \frac{V_s + V_d}{V_d}$$

$$V_s = \frac{\pi}{4} D^2 L = \frac{\pi}{4} \times 8^2 \times 10 = 502.65 \text{ cc}$$

$$r = \frac{502.65 + 70}{70}$$

$$r = 8.18$$

$$\text{Air-standard efficiency, } \eta_{otto} = 1 - \frac{1}{(8.18)^{0.4}} = 1 - \frac{1}{2.3179} = 0.56858$$

$$\text{Relative efficiency, } \eta_{rel} = \frac{0.2903}{0.568} \times 100 = 51.109\%$$

$$\eta_{bth} = \frac{bp}{m_f \times C_v}$$

$$= \frac{119.82 \times 60}{\frac{4.4}{10} \times 44000} \times 100$$

$$\eta_{bth} = 37.134\%$$

Volume flow rate of air at intake condition.

$$a = \frac{6 \times 287 \times 300}{1 \times 10^5} = 5.17 \text{ m}^3/\text{min}$$

Swept volume per minute,

$$\begin{aligned} V_s &= \frac{\pi}{4} D^2 L n K \\ &= \frac{\pi}{4} \times (0.1)^2 \times 0.9 \times \frac{4500}{2} \times 9 \\ &= 127.17 \text{ m}^3/\text{min}. \end{aligned}$$

$$\text{Volumetric efficiency, } \eta_v = \frac{5.17}{127.17} \times 100$$

$$\eta_v = 4.654\%$$

$$\text{Air-fuel ratio, } \frac{A}{F} = \frac{6.0}{0.44} = 13.64$$

Example 7.6

A gasoline engine is specified to be 4-stroke and four-cylinder. It has a bore of 80 mm and a stroke of 100 mm. On test it develops a torque of 75 Nm when running at 3000 rpm. If the clearance volume in each cylinder is 60 cc the relative efficiency with respect to brake thermal efficiency is 0.5 and the calorific value of the fuel is 42 MJ/kg; determine the fuel consumption in kg/hr. and the brake mean effective pressure.

Solution

$$\begin{aligned} \text{Swept volume, } V_s &= \frac{\pi}{4} \times 0.08^2 \times 0.1 = 5.024 \times 10^{-4} \text{ m}^3/\text{cylinder} \\ &= 502.4 \text{ cc/cylinder} \end{aligned}$$

$$\text{Compression ratio} = \frac{502.4 + 60}{60} = 9.373$$

$$\text{Air-standard efficiency} = 1 - \frac{1}{(9.373)^{0.4}} = 0.5914$$

$$\begin{aligned} \eta_{bth} &= \text{Relative } \eta \times \text{Air-standard } \eta \\ &= 0.5 \times 0.5914 \\ &= 0.2957 \end{aligned}$$

$$bp = \frac{2 \times \pi \times 3000 \times 75}{60000} = 23.55 \text{ kW}$$

$$\text{Heat supplied} = \frac{23.55}{0.2957} = 79.64 \text{ kJ/s}$$

$$\text{Fuel consumption} = \frac{79.64 \times 3600}{42000} = 6.8264 \text{ kg/hr}$$

$$P_{bm} = \frac{P \times 60000}{V_s n K}$$

$$= \frac{23.55 \times 60000}{5.024 \times 10^{-4} \times \frac{3000}{2} \times 4} = 4.6875 \times 10^5 \text{ N/m}^2$$

$$= 4.6875 \text{ bar}$$

Example 7.7

A six-cylinder, four-stroke engine gasoline engine having a bore of 90 mm and stroke of 100 mm has a compression ratio 8. The relative efficiency is 60%. When the indicated specific fuel consumption is 3009 g/kWh. Estimate (i) The calorific value of the fuel and (ii) Corresponding fuel consumption given that $imep$ is 8.5 bar and speed is 2500 rpm.

Solution

$$\text{Air-standard efficiency} = 1 - \frac{1}{r^{r-1}} = 1 - \frac{1}{8^{0.4}} = 0.5647$$

$$\text{Relative efficiency} = \frac{\text{Thermal efficiency}}{\text{Air-standard efficiency}}$$

$$\text{Indicated thermal efficiency} = 0.6 \times 0.5647 = 0.3388$$

$$\eta_{ith} = \frac{1}{i_{sfc} \times C_v}$$

$$C_v = \frac{1}{\eta_{ith} \times i_{sfc}} = \frac{3600}{0.3 \times 0.3388}$$

$$C_v = 35417.035 \text{ kJ/kg}$$

$$ip = \frac{P_{im} \text{ LAnK}}{60000}$$

$$= \frac{8.5 \times 10^5 \times 0.1 \times \frac{\pi}{4} \times 0.09^2 \times \frac{2500}{2} \times 6}{60000} = 67.6 \text{ kW}$$

$$\text{Fuel consumption} = i_{sfc} \times ip = 0.3 \times 67.6$$

$$ip = 20.28 \text{ kg/h.}$$

Example 7.8

The observations recorded after the conduct of a retardation test on a single-cylinder diesel engine are as follows :

Rated power = 10 kW

Rated speed = 500 rpm

Sl. No.	Drip in Speed	Time for Fall of Speed at no Load, t_2 (s)	Time for Fall of Speed at 50% Load, t_3 (s)
1.	500-400	7	2.2
2.	500-350	10.6	3.7
3.	500-325	12.5	4.8
4.	500-300	15.0	5.4
5.	500-275	16.6	6.5
6.	500-250	18.9	7.2

First we draw a graph of drop in speed versus time taken for the drop.

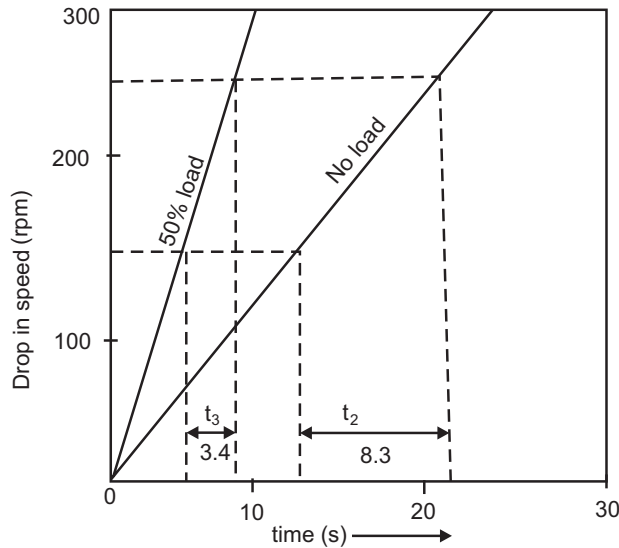


Figure 7.18 : Speed Vs Time

$$P = \frac{2\pi NT}{60000}$$

$$\text{Full load torque, } T = \frac{P \times 60000}{2\pi N} = \frac{10 \times 60000}{2 \times \pi \times 500} = 191.083 \text{ Nm}$$

$$\text{Torque at half load, } T_{1/2} = 95.5415 \text{ Nm}$$

From graph :

Time for the fall of 100 rpm at no load, $t_2 = 8.3$ sec.

Time for the fall of same 100 rpm at half load, $t_3 = 3.4$ sec.

$$\begin{aligned} T_f &= \frac{t_3}{t_2 - t_3} \times \text{Torque at 50\% load} \\ &= \frac{t_3}{t_2 - t_3} \times T_{1/2} = \frac{3.4}{(8.3 - 3.4)} \times 95.5415 = 66.294 \text{ Nm} \end{aligned}$$

$$\text{Friction power} = \frac{2\pi N T_f}{60000} = \frac{2\pi \times 500 \times 66.294}{60000} = 3.469 \text{ kW}$$

$$\eta_m = \frac{bp}{bp + fp} = \frac{10}{10 + 3.469} \times 100 = 74.24\%$$

Example 7.9

A 4-cylinder, 4-stroke cycle engine having cylinder diameter 100 mm and stroke 120 mm was tested at 1600 rpm and the following readings were obtained.

Fuel consumption = 0.27 litres/minute, Specific gravity fuel = 0.74,
B.P. = 31.4 kW, Mechanical efficiency = 80%, Calorific value of fuel
= 44000 kJ/kg.

Determine :

- (i) $bsfc$,
- (ii) $imep$, and
- (iii) Brake thermal efficiency.

Solution

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

$$L = 120 \text{ mm} = 0.12 \text{ m}$$

$$\eta_m = 80\% = 0.8$$

(i) Brake Specific Fuel Consumption (*bsfc*) :

$$= \frac{0.27 \times 0.74 \times 60}{31.4} = 0.38174 \text{ kJ/kW.hr}$$

(ii) Indicated Power :

$$I.P = \frac{n \times P_{imp} \times L \times A \times N}{2 \times 60}$$

$$\frac{B.P}{\eta_m} = \frac{n \times P_{imp} \times 0.12 \times \frac{\pi}{4} \times (0.1)^2 \times 1600}{2 \times 60}$$

$$\therefore \frac{31.4}{0.8} = 4 \times P_{imp} \times 0.01256637$$

$$\therefore P_{imp} = 780.85 \text{ kN/m}^2$$

(iii) Brake Thermal Efficiency :

$$\eta_{bth} = \frac{\text{Brake power}}{\text{Heat supplied}}$$

$$= \frac{31.4}{\frac{0.27 \times 0.74}{60} \times 44000} \times 100 = 21.43\%$$

Example 7.10

A single cylinder and stroke cycle I.C. engine when tested, the following observations available :

Area of indicator diagram = 3 sq.cm, Length of indicator diagram = 4 cm, Spring constant = 10 bar/cm, Speed of engine = 400 rpm, Brake drum diameter = 120 cm, Dead weight on brake = 380 N, Spring balance reading = 50 N, Fuel consumption = 2.8 kg/hr., $C_v = 42000 \text{ kJ/kg}$, Cylinder diameter = 16 cm, Piston stroke = 20 cm.

Find :

- (i) F.P.,
- (ii) Mechanical efficiency,
- (iii) *bsfc*, and
- (iv) Brake thermal efficiency.

Solution

Indicated mean effective pressure,

$$P_{imp} = \frac{\text{Area of indicate diameter}}{\text{Length of indicated diameter}} \times \text{Spring constant}$$

$$= \frac{A_i}{L_i} \times K_i$$

$$= \frac{3}{4_i} \times 10$$

$$P_{imep} = 7.5 \text{ bar} \quad \dots (1)$$

$$\begin{aligned} \text{Indicated power} = I.P &= \frac{P_{imp} \times L \times A \times N}{60} \\ &= \frac{7.5 \times 10^5 \times 0.2 \times \frac{\pi}{4} (0.16)^2 \times 400}{60 \times 2} = 10.05 \text{ kW} \quad \dots (2) \end{aligned}$$

$$\begin{aligned} \text{Brake Power} = B.P &= \frac{2\pi NT}{60} = \frac{2\pi N (W - S) \frac{b}{2}}{60} \\ B.P &= \frac{2\pi \times 400 (380 - 50)}{60} \times \frac{1.2}{2} = 8.294 \text{ W} \quad \dots (3) \end{aligned}$$

$$\begin{aligned} \text{(i) Frictional Power} = F.P. &= I.P. - B.P. \\ &= 10.05 - 8.294 \\ &= 1.756 \text{ kW} \end{aligned}$$

$$\text{(ii) Mechanical Efficiency } \eta_m = \frac{B.P.}{I.P.} = \frac{8.294}{10.05} \times 100 = 82.53\%$$

$$\begin{aligned} \text{(iii) Brake Specific Fuel Consumption (bsfc):} \\ &= \frac{2.8}{8.294} = 0.3376 \text{ kg/kW.hr} \end{aligned}$$

$$\begin{aligned} \text{(iv) Brake Thermal Efficiency } (\eta_{bth}) \\ &= \frac{B.P.}{\text{Heat supplied}} = \frac{8.294}{\frac{2.8}{3600} \times 42000} \times 100 = 25.39\% \end{aligned}$$

Example 7.11

A six-cylinder 4-stroke petrol engine having a bore of 90 mm and stroke of 100 mm has a compression ratio of 7. The relative efficiency with reference to indicated thermal efficiency is 55% when indicated mean specific fuel consumption is 0.3 kg/kWh. Estimate the calorific value of the fuel and fuel consumption in kg/hr. Given that indicated mean effective pressure is 8.5 bar and speed is 2500 r.p.m.

Solution

Number of cylinders = $n_1 = 6$, $L = 100 \text{ mm} = 0.1 \text{ m}$

$d = 90 \text{ mm} = 0.09 \text{ m}$, $r = 7$ (P.U. May 2006)

$\eta_r = 55\% = 0.55$ [based on indicated thermal efficiency]

$isfc = 0.3 \text{ kg/kWh}$

$P_{mi} = 8.5 \text{ bar}$

$N = 2500 \text{ rpm}$

$$I.P = \frac{P_{imp} \times A \times L \times N}{60000} \times n_1$$

where, $n = \frac{N}{2} = \frac{2500}{2} = 1250 \text{ strokes/mm}$ [for 4 stroke engine]

$$A = \frac{\pi}{4} d^2$$

From Eq. (1), we have,

$$I.P. = (8.5 \times 10^5) \times \frac{\pi}{4} (0.09)^2 \times 0.1 \times 1250 \times 6 \times \frac{1}{60000} = 67.593 \text{ kW}$$

(i) Fuel consumption, f :

$$Isfc = \text{i.e. } 0.3 =$$

$$f = 20.278 \text{ kg/hr}$$

(ii) Calorific Value (C_v) of fuel :

$$\text{Air standard efficiency, } \eta_a = 1 - \frac{1}{(r)^{(\gamma-1)}} = 1 - \frac{1}{(7)^{(1.4-1)}} = 0.42467$$

$$\text{Relative efficiency, } \eta_r = \frac{\text{Indicate thermal efficiency, } \eta_i}{\text{Air standard efficiency, } \eta_a}$$

$$\eta_i = \eta_r \times \eta_a = 0.55 \times 0.42467 = 0.2346$$

$$\text{But, } \eta_i = 0.2346 = \frac{67.593}{\left(\frac{20.278}{3600}\right) \times C_v}$$

$$\therefore C_v = 51150.6 \text{ kJ/kg.}$$

Example 7.12

A two stroke diesel engine was motored when the meter reading was 1.5 kW. Then the test on the engine was carried out for one hour and the following observations were recorded: Brake torque = 120 Nm; Speed = 600 rpm; Fuel used = 2.5 kg; calorific value of fuel = 40.3 MJ/kg; Cooling water used = 818 kg; Rise in temperature of cooling water = 10°C.

Exhaust gas temperature = 345°C. Room temperature = 25°C; A/F = 32 : 1.

Determine :

- (i) bp ,
- (ii) ip ,
- (iii) Mechanical efficiency,
- (iv) Indicated thermal efficiency, and
- (v) Draw heat balance sheet on minute basis and also in percentage. (P.U. Dec. 2006).

Solution

$$(a) \quad B.P. = \frac{2\pi NT}{1000} = 2\pi \times \frac{600}{60} \times \frac{120}{1000} = 7.54 \text{ kW}$$

$$I.P. = B.P. + F.P. = 7.54 + 15 = 9.04 \text{ kW}$$

$$\text{Mechanical } \eta = \frac{7.54}{9.04} = 0.834 = 83.4\%$$

$$\begin{aligned} \text{Indicated thermal } \eta &= \frac{I.P.}{\text{Heat supplied}} \\ &= \frac{9.04 \times 3600}{2.5 \times 40.3 \times 10^3} = 0.323 = 32.3\% \end{aligned}$$

(b) Heat Balance for the Engine :

- (i) Energy supplied $= \frac{2.5 \times 40300}{60} = 1680 \text{ kJ/min}$
- (ii) Energy Distributed
- (a) Heat in *B.P.* $= 7.54 \times 60 = 452.4 \text{ kJ/min.}$
- (b) Heat in cooling water $= \frac{818}{60} \times 4.2 \times 10 = 570.8 \text{ kJ/min}$
- (c) Heat in exhaust gases $= m_g C_{pg} (\Delta T)$
 $= \frac{33 \times 2.5}{60} \times 1.05 (345 - 25) = 462 \text{ kJ/min}$
- (d) Heat unaccounted (by difference)
 $= 1680 - (452.4 + 570.8 + 462) = \mathbf{197.8 \text{ kJ/min}}$

Percentage Heats

Heat supplied = 1679 kJ/min. = 100%

- (a) Heat in *B.P.* $= \frac{452.4}{1679} \times 100 = 20.94\%$
- (b) Heat in cooling water $= \frac{462}{1679} \times 100 = 33.97\%$
- (c) Heat in exhaust gases $= \frac{462}{1679} \times 100 = 27.51\%$
- (d) Heat unaccounted $= \frac{197.8}{1679} \times 100 = 11.78\%$

Example 7.13

The following observations were recorded during a trial on a 4-stroke diesel engine :

Power absorbed by non-firing engine when
 Driven by an electric motor = 10 kW
 Speed of the engine = 1750 rpm
 Brake torque = 327.4 Nm
 Fuel used = 15 kg/hr.
 Calorific value of fuel = 42000 kJ/kg
 Air supplied = 4.75 kg/min.
 Cooling water circulated = 16 kg/min.
 Outlet temperature of cooling water = 65.8°C
 Temperature of exhaust gas = 400°C
 Room temperature = 20.8°C
 Specific heat of water = 4.19 kJ/kg·K
 Specific heat of exhaust gas = 1.25 kJ/kg·K

Determine :

- (i) *bp*,
 (ii) Mechanical efficiency,

- (iii) *bsfc*,
(iv) Draw up heat balance sheet on kW basis.

Solution

- (i) Brake Power (*b.p.*) :

$$b.p. = 2\pi NT = 2 \times \pi \times \frac{1750}{60} \times 327.4 \times 10^{-3} = 60.01 \text{ kW}$$

- (ii) Mechanical Efficiency (η_m) :

$$\eta_m = \frac{b.p.}{i.p.}$$

But, $i.p. = b.p. + f.p.$

$$f.p. = 10 \text{ kW}$$

Given that power absorbed by non-firing engine when driven by electric motor. This is frictional power.

This type of testing is done in a motoring test which is used to calculate the frictional power of an engine.

Hence, $f.p. = 10 \text{ kW}$

$$\therefore i.p. = b.p. + f.p. \\ = 60.01 + 10$$

$$\therefore i.p. = 70.01 \text{ kW}$$

$$\therefore \eta_m = \frac{60.01}{70.01} = 0.8571 = 85.71\%$$

- (iii) *bsfc* : Brake Specific Fuel Consumption :

$$\therefore bsfc = \frac{m_f / \text{hr.}}{b.p.} = \frac{15}{60.01} = 0.25 \text{ kg/kW.hr}$$

- (iv) Heat Balance Sheet in kW basis :

- (i) Power supplied by fuel = $m_f \times C_v$

$$= \frac{15}{3600} \times 42000 = 175 \text{ kW}$$

- (ii) Brake power = 60.01 kW

- (iii) Power to cooling water = $m_w C_{pw} \Delta T$

$$= \frac{16}{60} \times 4.19 \times (T_o - T_{in})$$

$$T_o = 65.8 + 273 = 338.8 \text{ K}$$

$$T_{in} = 20.8 + 273 = 293.8 \text{ K}$$

Power lost to cooling water = 50.28 kW

- (iv) Power to exhaust = $m_E C_{pE} \Delta T$

Here, mass of exhaust gases

$$m_E = m_a + m_f$$

$$= \frac{4.75}{60} + \frac{15}{3600} = 0.0833 \text{ kg/s}$$

$$\begin{aligned}\text{Power lost to exhaust} &= 0.0833 \times 1.25 \times (400 - 20.8) \\ &= 39.48 \text{ kW}\end{aligned}$$

Heat Balance Sheet :

	Input (kW)	%	Output	kW	%
01	Power from fuel 175 kW	100%	Brake power	60.01	34.29
			Power lost to cooling water	50.28	28.73
			Power lost to exhaust	39.48	22.56
			Unaccounted power	25.23	14.42
Total	174 kW	100%	Total	175	100%

Example 7.14

A single cylinder engine running at 1800 rpm develops a torque of 8 Nm. The indicated power of the engine 1.8 kW. Find the loss due to friction power as the percentage of brake power.

Solution

Given Data : Single cylinder engine

Speed of engine = $N = 1800$ rpm

Torque = $T = 8$ Nm

$I.P.$ = 1.8 kW

$$\begin{aligned}\text{Brake power} = B.P. &= \frac{2\pi NT}{60} = \frac{2\pi \times 1800 \times 8}{60} \\ &= 1507.96 \text{ W} = 1.50796 \text{ kW}\end{aligned}$$

Friction power = $F.P. = I.P. - B.P.$

$$= 1.8 - 1.50796$$

$$= 0.29204 \text{ kW}$$

Loss due to friction power as the percentage of brake power

$$= \frac{0.29204}{1.50796} \times 100$$

$$= 19.37\% \text{ of brake power.}$$

SAQ 1

- A vertical single cylinder four stroke diesel engine has a bore = 80 mm and stroke = 100 mm respectively. It is water cooled and develops a torque of 3.5 N-m. Calculate the mean effective of the engine.
- A diesel engine consumes 5 grams fuel per second and develops a brake power 75 kW. It has a mechanical efficiency of 85%. Find (a) Brake specific fuel consumption in kg/hWhr, (b) Indicated specific fuel consumption.
- A four stroke gas engine has a bore of 20 cm and stroke of 35 cm and runs at 400 rpm firing every cycle. The air-fuel ratio is 4 : 1 by volume. Its volumetric efficiency at NTP conditions is 80%, determine the volume of gas used per minute. If the calorific value of the gas is 8 MJ/m³ at NTP and the brake thermal efficiency is 25%. Determine brake power of engine.

- (d) The following readings are taken during a test of a four-cylinder, two stroke gasoline engine. Diameter = 10 cm, Stroke = 15 cm, Speed = 1700 rpm, Area of positive loop of the indicator diagram = 5.75 sq.cm; Area of the negative loop of the indicator diagram = 0.25 cm²; Length of indicator diagram = 5.5 cm, Spring constant = 4.0 bar/cm. Find the indicated power of the engine.
- (e) A four cylinder engine running at 1250 rpm delivers 21 kW power. The average torque when one cylinder was cut is 110 N-m. The calorific value of the fuel is 43 MJ/hr. The engine uses 360 gms of gasoline per kWh. Find indicated thermal efficiency.

SAQ 2

- (a) An 8-cylinder, four stroke engine of bore 10 cm and 9 cm stroke has a compression ratio of 7 is 4500 rpm on a dynamometer which has 54 cm arm. During a 10 minutes test the dynamometer scale beam reading was 48 kg and the engine consumed 4.4 kg of gasoline having a calorific value of 44000 kJ/kg. Air at 27°C temperature and 1 bar pressure was supplied to the carburetor at the rate of 6 kg/min. Find (i) the brake power delivered, (ii) The brake mean effective pressure, (iii) The brake specific fuel consumption, (iv) The brake specific air consumption, (v) The brake thermal efficiency, (vi) The volumetric efficiency, (vii) The air-fuel ratio.
- (b) In a test for four-cylinders, four-stroke engine has a diameter of 100 mm, stroke = 120 mm, speed of engine = 1800 rpm, fuel consumption of 0.2 kg/min, calorific value of fuel is 44000 kJ/kg. Difference in tension on either side of brake pulley = 40 kg, Brake circumference is 300 cm. If the mechanical efficiency is 90%. Calculate (i) Brake-thermal efficiency, (ii) Indicated thermal efficiency, (iii) Indicated mean effective pressure and (iv) Brake specific fuel consumption.
- (c) A 4-stroke cycle gas engine has a bore of 20 cm and a stroke of 35 cm. The compression ratio is given to be 8. In a test on the engine the indicated mean effective pressure is 5 bar, the air to gas ratio is 6 : 1 and the calorific value of the gas is 12 MJ/m³ at NTP. At the beginning of the compression stroke the temperature is 77°C and pressure is 0.98 bar. Neglecting residual gases, determine the indicated power, the thermal efficiency and the relative efficiency of the engine at 250 rpm.
- (d) An indicator diagram taken from a single-cylinder, four-stroke CI engine has a length of 100 mm and an area 2000 mm². The indicator pointer deflects a distance of 10 mm for pressure increment of 2 bar in the cylinder. If the bore and stroke of the engine cylinder are both 100 mm and the engine speed is 1200 rpm. Calculate the mean effective pressure and the indicated power. If the mechanical efficiency is 85%. What is the brake power developed?
- (e) A gasoline engine working on 4-stroke develops a brake power of 22 kW. A Morse test was conducted on this engine and the brake power (kW) obtained when each cylinder was made inoperative by short circuiting. The spark plugs are 14.9, 14.3, 14.8 and 14.5 respectively. The test was conducted at constant speed. Find the indicated power, mechanical efficiency and *b MEP* when all the cylinders are firing. The bore of engine is 80 mm and stroke is 90 mm. The engine is running at 3000 rpm.

7.10 SUMMARY

Let us summarise what we have learnt in this unit. In this unit, we have understood in detail about IC engine testing. In evaluation of engine performance, certain basic parameters, we chosen and we studies about measurement of fuel consumption, air consumption, etc. Measurements of exhaust smoke as well as exhaust emission where also highlighted. Lastly performance of SI engine and CI engine were discussed.

7.11 KEY WORDS

Engine Performance	: It is a indication of the degree of success with which it does its assigned job, i.e. conversion of chemical energy contained in the fuel into the useful mechanical work.
Power	: Power is defined as the rate of doing work.
Indicated Power	: The total power developed by combustion of fuel in the combustion chamber.
Mean Effective Pressure	: It is defined as hypothetical pressure which is thought to be acting on the piston throughout the power stroke.
Volumetric Efficiency	: It is defined as the ratio of actual volume to the charge drawn in during the suction stroke to the swipt volume of the piston.
Fuel Air Ratio	: It is the ratio of the mass of fuel to the mass of air in the fuel air mixture.

7.12 ANSWERS TO SAQs

SAQ 1

$$\begin{aligned}
 \text{(a)} \quad P &= \frac{2\pi NT}{60000} = \frac{P_{bm} LA_n}{60000} \\
 P_{bm} &= \frac{2\pi NT}{LA_n} = \frac{2\pi NT}{L \times \frac{\pi}{4} \times D^2 \times \frac{N}{2}} = \frac{16T}{D^2 L} \\
 &= \frac{16 \times 22.5}{(0.08)^2 \times 0.1} = 5.875 \times 10^5 \text{ Pa} \\
 &= 5.875 \text{ bar} \\
 P_{bm} &= \frac{P \times 6 \times 10^4}{\frac{\pi}{4} D^2 L \times \frac{N}{2}} = 10^{-5} \text{ bar} \\
 &= \frac{4 \times 60000}{\frac{\pi}{4} \times (0.08)^2 \times 0.1 \times \frac{1500}{2}} \times 10^{-5} \\
 &= \frac{24 \times 10^4 \times 4 \times 2 \times 10^{-5}}{\pi \times (0.08)^2 \times 0.1 \times 1500} = 6.369 \text{ bar}
 \end{aligned}$$

$$T = \frac{P \times 60000}{2\pi N} = \frac{4 \times 6 \times 10^4}{2 \times 3.14 \times 1500} = 25.477 \text{ Nm}$$

$$(b) \quad bsfc = \frac{m_f}{pb} = \frac{5}{75} = 0.066 \text{ g/kWs}$$

$$= \frac{0.066}{1000} \text{ g} \times 3600 = 0.24 \text{ kg/kWh}$$

$$isfc = bsfc \times \eta_m$$

$$= 0.24 \times 0.85 = 0.204 \text{ kg/kWh}$$

$$(c) \quad \text{Swept volume, } V_s = \frac{\pi}{4} D^2 L$$

$$= \frac{\pi}{4} \times 20^2 \times 25$$

$$= 7853.93 \text{ cc}$$

Total charge taken in per cycle

$$V_c = 0.8 \times 7853.98$$

$$= 6.2832 \times 10^{-3} \text{ m}^3$$

Volume of gas used per minute

$$V_g = \frac{6.2832 \times 10^{-3}}{4 + 1} \times \frac{400}{2}$$

$$= 0.25133 \text{ m}^3 \text{ at NTP/min.}$$

Heat input = $8000 \times 0.25133 = 2010.64 \text{ kJ/min}$

$$bp = \eta_{th} \times \text{Heat input}$$

$$= \frac{0.25 \times 2010.64}{60}$$

$$bp = 8.377 \text{ kW}$$

$$(d) \quad \text{Net area of diagram} = 5.75 - 0.25$$

$$= 5.5 \text{ cm}^2$$

$$\text{Average height of the diagram} = \frac{5.5}{5.5} = 1 \text{ cm}$$

$$P_{im} = \text{Average height of the diagram} \times \text{spring constant}$$

$$= 1 \times 4 = 4 \text{ bar}$$

$$ip = \frac{P_{im} L A_n}{60000}$$

$$= \frac{4 \times 10^5 \times 0.15 \times \frac{\pi}{4} \times 0.1^2 \times 1700 \times 4}{60000}$$

$$i_p = 53.38 \text{ kW.}$$

$$\begin{aligned}
 \text{(e) Average } bp \text{ for 3 cylinders} &= \frac{2\pi NT}{60000} \\
 &= \frac{2\pi \times 1250 \times 110}{60000} \\
 &= 14.39 \text{ kW}
 \end{aligned}$$

$$\begin{aligned}
 \text{Average } ip \text{ with 1 cylinder} &= 21 - 14.39 \\
 &= 6.608 \text{ kW}
 \end{aligned}$$

$$\text{Total input} = 4 \times 6.608 = 26.433 \text{ kW}$$

$$\begin{aligned}
 isfc &= bsfc \times \frac{bp}{ip} = 360 \times \frac{21}{26.433} \\
 &= 286.006 \approx 286 \text{ g/kWh}
 \end{aligned}$$

$$\begin{aligned}
 \text{Fuel combustion} &= \frac{isfc \times ip}{3600 \times 1000} \\
 &= \frac{286 \times 26.433}{3600 \times 1000} \\
 &= 2.099 \times 10^{-3} \text{ kg/sec.}
 \end{aligned}$$

$$\begin{aligned}
 \eta_{\text{ith}} &= \frac{ip}{m_f \times C_v} \\
 &= \frac{26.433}{2.009 \times 10^{-3} \times 43000} \times 100 \\
 \eta_{\text{ith}} &= 29.29\%
 \end{aligned}$$

SAQ 2

$$\begin{aligned}
 \text{(a) } bp &= \frac{2\pi NT}{60000} = \frac{2\pi \times 4500 \times 48 \times 0.54 \times 9.81}{60000} = 119.82 \text{ kW} \\
 b_{mep} &= \frac{bp \times 6000}{LANK} = \frac{119.82 \times 60000}{0.09 \times \frac{\pi}{4} (0.1)^2 \times \frac{4500}{2} \times 8} = 5.653 \times 10^5 \text{ Pa}
 \end{aligned}$$

$$b_{mep} = 5.653 \text{ bar}$$

$$bsfc = \frac{\frac{4.4}{10} \times 60}{119.82}$$

$$bfsc = 0.2203 \text{ kg/kWh}$$

$$bsfc = \frac{6 \times 60}{119.82} = 3.004 \text{ kg/kWh}$$

$$\begin{aligned}
 \text{(b) } bp &= \frac{2\pi NT}{60000} = \frac{2\pi NWR}{60000} = \frac{WN 2\pi R}{60000} \\
 &= \frac{40 \times 9.81 \times 1800 \times 3}{60000} = 35.316 \text{ kW}
 \end{aligned}$$

$$\eta_{bth} = \frac{bp}{m_f \times C_v} \times \frac{35.316 \times 60}{0.2 \times 44000} \times 100 = 24.079\%$$

$$\eta_{ith} = \frac{\eta_{bth}}{\eta_m} \times 100 = \frac{24.079}{0.9} \times 100 = 26.75\%$$

$$imep = \frac{\frac{bp}{\eta_m} \times 6000}{LANK} = \frac{\frac{35.316}{0.8} \times 60000}{0.12 \times \frac{\pi}{4} (0.1)^2 \times \frac{1800}{2} \times 4} = 6.94 \times 10^5 \text{ Pa}$$

$$imep = 6.94 \text{ bar}$$

$$b_{sfc} = \frac{0.2 \times 60}{35.316} = 0.339 \text{ kg/kWh}$$

(c) Swept volume, $V_s = \frac{\pi}{4} D^2 L = \frac{\pi}{4} \times 20^2 \times 35 = 10990 \text{ cc}$

$$\text{Volume of gas in cylinder} = \frac{1}{1 + \frac{A}{F}} \times V_1$$

$$V_1 = V_s + \frac{V_s}{r-1} = V_s \frac{8}{5} = \frac{1}{8+1} \times 10990 \times \frac{8}{5}$$

Since, the residual gases are to be neglected; one can assume a volumetric efficiency of 100%.

Normal pressure = 1 bar

$$\left(\frac{PV}{T} \right)_{NTP} = \left(\frac{P_1 V_1}{T_1} \right)_{\text{Working}}$$

$$\text{Volume of gas at NTP condition} = 1953.7 \times 0.98 \times \frac{273}{350} = 1493.4 \text{ cc}$$

$$\begin{aligned} \text{Heat added} &= 1493.4 \times 10^{-6} \times 12 \times 10^3 \\ &= 17.92 \text{ kJ/cycle} \end{aligned}$$

$$ip = \frac{P_{im} \times V_s \eta}{60000} = \frac{5 \times 10^5 \times 10990 \times 10^{-6} \times \frac{250}{2}}{60000} = 11.44 \text{ kW}$$

$$\begin{aligned} \eta_{ith} &= \frac{ip}{\text{Heat added (in kW)}} \times 100 \\ &= \frac{11.44}{17.92 \times \frac{250}{2 \times 60}} \times 100 = 30.66\% \end{aligned}$$

$$\text{Air-standard efficiency} = 1 - \frac{1}{8^{0.4}} = 0.5647$$

$$\text{Relative efficiency} = \frac{0.3066}{0.5467} \times 100 = 54.29\%$$

(d) Mean height of the indicator diagram = $\frac{2000}{100} = 20 \text{ mm}$

$$\text{Mean effective pressure} = \frac{20}{10} \times 2 = 4 \text{ bar}$$

$$\text{Indicated power, } ip = \frac{P_{im} L A n}{60000}$$

$$= \frac{4 \times 10^5 \times 0.1 \times \frac{\pi}{4} \times 0.1^2 \times \frac{1200}{2}}{60000} = 3.14 \text{ kW}$$

$$bp = ip \times \eta_m = 3.14 \times 0.85 = 2.669 \text{ kW}$$

$$\begin{aligned} \text{(e) } ip_1 &= bp_{1234} - bp_{234} \\ &= 22 - 14.9 = 7.1 \text{ kW} \end{aligned}$$

$$\begin{aligned} ip_2 &= bp_{1224} - bp_{134} \\ &= 22 - 14.3 = 7.7 \text{ kW} \end{aligned}$$

$$\begin{aligned} ip_3 &= bp_{1234} - bp_{124} \\ &= 22 - 14.8 = 7.2 \text{ kW} \end{aligned}$$

$$\begin{aligned} ip_4 &= bp_{1234} - bp_{123} \\ &= 22 - 14.5 = 7.5 \text{ kW} \end{aligned}$$

$$\begin{aligned} ip_1 + ip_2 + ip_3 + ip_4 &= ip_{1234} = 7.1 + 7.7 + 7.2 + 7.5 \\ &= 29.5 \text{ kW} \end{aligned}$$

$$\eta_w = \frac{22}{29.5} \times 100 = 74.57$$

$$\begin{aligned} P_{bm} &= \frac{bp \times 60000}{\angle \text{ AnK}} \\ &= \frac{22 \times 60000}{0.09 \times \frac{\pi}{4} \times (0.08)^2 \times \frac{3000}{2} \times 4} = 4.8665 \times 10^5 \text{ Pa} \end{aligned}$$

$$P_{bm} = 4.8655 \text{ bar.}$$