CHAPTER 1

Testing of I.C. Engines

1.1. Introduction: - The basic task in the design and development of I.C. Engines is to reduce the cost of production and improve the efficiency and power output. In order to achieve the above task, the engineer has to compare the engine developed by him with other engines in terms of its output and efficiency. Hence he has to test the engine and make measurements of relevant parameters that reflect the performance of the engine. In general the nature and number of tests to be carried out depend on a large number of factors. In this chapter only certain basic as well as important measurements and tests are described.

1.2. Important Performance Parameters of I.C. Engines:- The important performance parameters of I.C. engines are as follows:

(i) Friction Power,

(ii) Indicated Power,

(iii) Brake Power,

(iv) Specific Fuel Consumption,

(v) Air – Fuel ratio

(vi) Thermal Efficiency

(vii) Mechanical Efficiency,

(viii) Volumetric Efficiency,

(ix) Exhaust gas emissions,

(x) Noise

1.3. Measurement of Performance Parameters in a Laboratory

1.3.1. Measurement of Friction Power:- Friction power includes the frictional losses and the pumping losses. During suction and exhaust strokes the piston must move against a gaseous pressure and power required to do this is called the “pumping losses”. The
friction loss is made up of the energy loss due to friction between the piston and cylinder walls, piston rings and cylinder walls, and between the crank shaft and camshaft and their bearings, as well as by the loss incurred by driving the essential accessories, such as water pump, ignition unit etc.

Following methods are used in the laboratory to measure friction power:

(i) Willan’s line method;

(ii) From the measurement of indicated power and brake power;

(iii) Motoring test;

(iv) Retardation test;

(v) Morse Test.

1.3.1.1. Willan’s Line Method:- This method is also known as fuel rate extrapolation method. In this method a graph of fuel consumption (vertical axis) versus brake power (horizontal axis) is drawn and it is extrapolated on the negative axis of brake power (see Fig. 1). The intercept of the negative axis is taken as the friction power of the engine at

![Figure 1: Willan's line method](image)

that speed. As shown in the figure, in most of the power range the relation between the fuel consumption and brake power is linear when speed of the engine is held constant and this permits extrapolation. Further when the engine does not develop power, i.e. brake
power = 0, it consumes a certain amount of fuel. This energy in the fuel would have been spent in overcoming the friction. Hence the extrapolated negative intercept of the horizontal axis will be the work representing the combined losses due to friction, pumping and as a whole is termed as the frictional loss of the engine. This method of measuring friction power will hold good only for a particular speed and is applicable mainly for compression ignition engines.

The main draw back of this method is the long distance to be extrapolated from data between 5 and 40 % load towards the zero line of the fuel input. The directional margin of error is rather wide because the graph is not exactly linear.

1.3.1.2. From the Measurement of Indicated Power and Brake Power:- This is an ideal method by which friction power is obtained by computing the difference between the indicated power and brake power. The indicated power is obtained from an indicator diagram and brake power is obtained by a brake dynamometer. This method requires elaborate equipment to obtain accurate indicator diagrams at high speeds.

1.3.1.3. Morse Test:- This method can be used only for multi – cylinder IC engines. The Morse test consists of obtaining indicated power of the engine without any elaborate equipment. The test consists of making, in turn, each cylinder of the engine inoperative and noting the reduction in brake power developed. In a petrol engine (gasoline engine), each cylinder is rendered inoperative by “shorting” the spark plug of the cylinder to be made inoperative. In a Diesel engine, a particular cylinder is made inoperative by cutting off the supply of fuel. It is assumed that pumping and friction are the same when the cylinder is inoperative as well as during firing.

In this test, the engine is first run at the required speed and the brake power is measured. Next, one cylinder is cut off by short circuiting the spark plug if it is a petrol engine or by cutting of the fuel supply if it is a diesel engine. Since one of the cylinders is cut off from producing power, the speed of the engine will change. The engine speed is brought to its original value by reducing the load on the engine. This will ensure that the frictional power is the same.

If there are $k$ cylinders, then

Total indicated power
when all the cylinders are working $= \sum_{j=1}^{k} i_{p_j}$

We can write $\sum_{j=1}^{k} i_{p_j} = B_t + F_t$ ..............................................(1)

where $i_{p_j}$ is the indicated power produced by $j$ th cylinder, $k$ is the number of cylinders,
B₁ is the total brake power when all the cylinders are producing power and F₁ is the total frictional power for the entire engine.

If the first cylinder is cut – off, then it will not produce any power, but it will have frictional losses. Then

we can write  \( \sum_{j=2}^{k} \text{ip}_j = B_1 - F_1 \) ..........................(2)

where \( B_1 \) = total brake power when cylinder 1 is cut - off and

\( F_1 \) = Total frictional power.

Subtracting Eq. (2) from Eq. (1) we have the indicated power of the cut off cylinder. Thus

\( \text{ip}_1 = B_1 - B_1 \) ..........................(3).

Similarly we can find the indicated power of all the cylinders, viz., \( \text{ip}_2, \text{ip}_3, \ldots \cdot \text{ip}_k \) Then the total indicated power is calculated as

\( (\text{ip})_{\text{total}} = \sum_{j=2}^{k} \text{ip}_j \) ..........................(4)

The frictional power of the engine is therefore given by

\( F_t = (\text{ip})_{\text{total}} - B_t \) ..........................(5)

The procedure is illustrated by some examples worked out at the end of the chapter.

1.4. MEASUREMENT OF INDICATED POWER

The power developed in the cylinder is known as Indicated Horse Power and is designated as IP.

The IP of an engine at a particular running condition is obtained from the indicator diagram. The indicator diagram is the \( p\-v \) diagram for one cycle at that load drawn with the help of indicator fitted on the engine. The construction and use of mechanical indicator for obtaining \( p\-v \) diagram is already explained.

A typical \( p\-v \) diagram taken by a mechanical indicator is shown in Figure 2.
The areas, the positive loop and negative loop, are measured with the help of a planimeter and let these be $A_p$ and $A_n$ cm$^2$ respectively, the net positive area is $(A_p - A_n)$. Let the actual length of the diagram as measured be $L$ cm, then the average height of the net positive area is given by

$$h = \frac{(A_p - A_n)}{L} \text{ in centimetre}$$

The height multiplied by spring-strength (or spring number) gives the indicated mean effective pressure of the cycle.

$$\text{Imep} = \frac{(A_p - A_n) \cdot S}{L} \quad \ldots(6)$$

Where $S$ is spring scale and it is defined as a force per unit area required to compress the spring through a height of one centimeter (N/m$^2$/cm).

Generally the area of negative loop $A_n$ is negligible compared with the positive loop and it cannot be easily measured especially when it is taken with the spring used for taking positive loop. Special light springs are used to obtain the negative loop. When two different springs are used for taking the $p$-$v$ diagram of positive and negative loop, then the net indicated mean effective pressure is given by

$$P_n = \frac{A_p \cdot S_p}{L} - \frac{A_n \cdot S_n}{L} \quad \ldots(7)$$

Where $S_p$ = Spring strength used for taking $p$-$v$ diagram of positive loop, (N/m$^2$/per cm) 
$S_n$ = Spring strength used for taking $p$-$v$ diagram of negative loop, (N/m$^2$/per cm) 
$A_p$ = Area in Cm$^2$ of positive loop taken with spring of strength $S_p$ 
$A_n$ = Area in Cm$^2$ of positive loop taken with spring of strength $S_n$

Sometimes spring strength is also noted as spring constant.

The IP developed by the engine is given by

$$IP = P_n L A_n \quad \ldots(8)$$

Where ‘$n$’ is the number of working strokes per second.
The explanation of this expression is already given in the last chapter.

1.5. MEASUREMENT OF B.P
Part of the power developed in the engine cylinder is used to overcome the internal friction. The net power available at the shaft is known as brake power and it is denoted by B.P. The arrangement used for measuring the BP of the engine is described below:

(a) Prony Brake. The arrangement of the braking system is shown in Figure 3. It consists of brake shoes made of wood and these are clamped on to the rim of the brake wheel by means of the bolts. The pressure on the rim is adjusted with the help of nut and springs as shown in Fig. 2. A load bar extends from top of the brake and a load carrier is attached to the end of the load bar. Weight kept on this load carrier is balanced by the torque reaction in the shoes. The load arm is kept horizontal to keep the arm length constant.

![Figure.3](image)

The energy supplied by engine to the brake is eventually dissipated as heat. Therefore, most of the brakes are provided with a means of supply of cooling water to the inside rim of the brake drum.

The BP of the engine is given by

\[
B.P \ (\text{brake power}) = \frac{2\pi NT}{60} \ \text{watts} = \frac{2\pi NT}{60} \times 1000 \ \text{Kw} \ldots \ (9)
\]

Where \(T = (W \cdot L) \ (\text{N-m})\)

Where \(W = \) Weight on load carrier, (N)

And \(L = \) Distance from the centre of shaft to the point of load-meter in meters.
The prony brake is inexpensive, simple in operation and easy to construct. It is, therefore, used extensively for testing of low speed engines. At high speeds, grabbing and chattering of the band occur and lead to difficulty in maintaining constant load. The main disadvantage of the prony brake is its constant torque at any one band pressure and therefore its inability to compensate for varying conditions.

1.5.1 Hydraulic Dynamometer.
The BP of an engine coupled to the dynamometer is given by

\[
B.P = 2\pi N W R / (60 \times 1000) = WN(2\pi R / (60 \times 1000)) \text{ Kw}
\]

The working of a prony brake dynamometer is shown in figure 4

![Hydraulic Dynamometer Diagram](image)

Figure 4: Hydraulic dynamometer

In the hydraulic dynamometer, as the arm length \( R \) is fixed, the factor \( [2 R / (60 \times 1000)] \) is constant and its value is generally given on the name plate of the dynamometer by the manufacturer and is known as brake or dynamometer constant. Then the BP measured by the dynamometer is given by

\[
B.P = \frac{WN}{K} \quad \text{...(10)}
\]

Where \( W \) = Weight measured on the dynamometer, N
\( K \) = Dynamometer constant \( (60 \times 1000 / 2\pi R) \)
and \( N \) = RPM of the engine.

The arm length ‘\( R \)’ is selected in such a way that \( K \) is a whole number. These dynamometers are directly coupled with the engine shaft.

1.5.2 Electric Dynamometer:
The electric generator can also be used for measured BP of the engine. The output of the generator must be measured by electrical instruments and corrected for generator efficiency. Since the efficiency of the generator depends upon load, speed and temperature, this device is rather inconvenient to use in the laboratory for obtaining precise measurement. To overcome these difficulties, the generator stator may be supported in ball bearing trunnions and the reaction force exerted on the stator of the generator may be measured by a suitable balance. The tendency to rotate or the reaction of the stator will be
equal and opposite to the torque exerted on the armature, which is driven by the engine which is shown in Figure 5.

The electric dynamometer can also be used as a motor to start and drive the engine at various speeds.

There are other types of dynamometers like eddy current dynamometer, fan brake and transmission dynamometers used for measurement of large power output.

1.5.3 Eddy current Type Dynamometer
The ‘eddy-current’ dynamometer is an effect, a magnetic brake in which a toothed steel rotor turns between the poles of an electromagnet attached to a trunioned stator. The resistance to rotation is controlled by varying the current through the coils and hence, the strength of the magnetic field. The flux tends to follow the smaller air gaps at the ends of the rotor teeth and eddy currents are set up within the metal of the pole pieces, resulting in heating the stator. The heat energy is removed by circulating water through a water jacket formed in the stator. Figure 6 shows the “Heenan eddy-current dynamometer”.
The power output of eddy-current dynamometer is given by the equation where \( C \) is eddy-current dynamometer constant.

The advantages of eddy-current dynamometer are listed below:

1. High absorbing power per unit weight of dynamometer.
2. Level of field excitation is below 1% of the total power handled by the dynamometer.
3. The torque development is smooth as eddy current developed smooth.
4. Relatively higher torque is provided under low speed conditions.
5. There is no limit to the size of dynamometer.

1.5.4 Swinging Field Dynamometer
The arrangement of swinging field dynamometer and corresponding diagram of electric connections are shown in Figure 7.
A swinging field DC dynamometer is basically a DC shunt motor. It is supported on trunnion bearings to measure the reaction torque that the outer casing and field coils tend to rotate with the magnetic drag. Therefore, it is named as “Swinging field”. The Torque is measured with an arm and weighting equipment in the usual manner.

The choice of dynamometer depends on the use for which the machine is purchased. An electric dynamometer is preferred as it can operate as motor used for pumping or generator for testing the engine. Also, engine friction power can also be measured by operating the dynamometer in the motoring mode.

An eddy-current or hydraulic dynamometer may be used because of low initial coast and an ability to operate at high speeds. The armature of the electric dynamometer is large and heavy compared with eddy-current dynamometer and requires strong coupling between dynamometer and engine.

**1.6 MEASUREMENT OF I.P OF MULTI-CYLINDER ENGINE (MORSE TEST)**

This method is used in multi-cylinder engines to measure I.P without the use of indicator. The BP of the engine is measured by cutting off each cylinder in turn. If the engine consists of 4-cylinders, then the BP of the engine should be measured four times cutting each cylinder turn by turn. This is applicable to petrol as well as for diesel engines. The cylinder of a petrol engine is made inoperative by “shorting” the spark plug whereas in case of diesel engine, fuel supply is cut-off to the required cylinder.

If there are ‘n’ cylinders in an engine and all are working, then

\[(B.P)_n = (I.P)_n - (F.P)_n \quad \ldots \ldots \ldots \quad (11)\]

Where F.P is the frictional power per cylinder.
If one cylinder is inoperative then the power developed by that cylinder (IP) is lost and the speed of the engine will fall as the load on the engine remains the same. The engine speed can be resorted to its original value by reducing the load on the engine by keeping throttle position same. This is necessary to maintain the FP constant, because it is assumed that the FP is independent of load and depends only on speed of the engine.

When cylinder “1” is cut off; then
\[(B.P)_n - (I.P)_n = (I.P)_{n-1} - (F.P)_{n-1} \ldots (12)\]

By subtracting Eq. (23.7) from Eq.(23.6), we obtain the IP of the cylinder which is not firing i.e., \((B.P)_n - (B.P)_{n-1} = (IP)_n - (IP)_{n-1} = I.P_1\)

Similarly IP of all other cylinders can be measured one by one then the sum of IPs of all cylinders will be the total IP of the engine.

This method of obtaining IP of the multicylinder engine is known as ‘Morse Test’.

1.7 MEASUREMENT OF AIR-CONSUMPTION

The method is commonly used in the laboratory for measuring the consumption of air is known as ‘Orifice Chamber Method’. The arrangement of the system is shown in Figure 8.

It consists of an air-tight chamber fitted with a sharp-edged orifice of known coefficient of discharge. The orifice is located away from the suction connection to the engine.

Due to the suction of engine, there is pressure depression in the chamber which causes the flow through orifice for obtaining a steady flow, the volume of chamber should be sufficiently large compared with the swept volume of the cylinder; generally 500 to 600 times the swept volume. A rubber diaphragm is provided to further reduce the pressure pulsations.

It is assumed that the intermittent suction of the engine will not affect the air pressure in the air box as the volume of the box is sufficiently large, and pressure in the box remains constant.

The pressure different causing the flow through the orifice is measured with the help of a water monometer. The pressure difference should be limited to 10cm of water to make the compressibility effect negligible. Let
- \(A_o\) = Area orifice in m²;
- \(h_w\) = Head of water in cm causing the flow.
- \(C_d\) = Coefficient of discharge for orifice. ; \(d\) = Diameter of orifice in cm.
- \(\rho_a\) = Density of air in kg/m³ under atmospheric conditions.

Head in terms of meters of air is given by

\[H = \frac{h_w}{100} \cdot \frac{\rho_w}{\rho_a} = \frac{h_w}{100} \cdot \frac{\rho_w}{\rho_a} = 10 h_w \frac{1000}{\rho_a} \ m \ of \ air\]

The velocity of air passing through the orifice is given by

\[v = \sqrt{2gH} \ m/Sec = \sqrt{2g \cdot 10 h_w \frac{1000}{\rho_a}} \ m/Sec\]

The volume of air passing through the orifice is given by
\[ v_a = A_0 v \cdot C_d \cdot \sqrt{\frac{2g \cdot 10 h_w}{\rho_a}} = 14.01 \cdot A_0 \cdot C_d \frac{h_w}{\rho_a} \text{ cu. m/Sec} \]

840.428 \( A_0 \cdot C_d \frac{h_w}{\rho_a} \) m³/ min

The volumetric efficiency of the engine

\[ = \text{Actual volume of air taken in as measure} = 14.01 A_0 \cdot C_d \frac{h_w}{\rho_a} \]

Displacement volume
Where \( N \) is RPM of the engine and \( n \) is number of cylinders. \( D \) & \( L \) are diameter and stroke of each cylinder.

Mass of air passing through the orifice is given by

\[ m_a = V_a \cdot \rho_a = 14.01 \times \frac{\pi d^2}{4 \times 100} \cdot C_d \frac{h_w}{\rho_a} = 11.003 \times 10^{-4} C_d \cdot d^2 \sqrt{\rho_a h_w} \]

\[ = 0.0011 C_d \cdot d^2 \sqrt{h_w \rho_a} \text{ Kg / Sec} = 0.066 \cdot C_d \cdot d^2 \sqrt{h_w \rho_a} \text{ kg / min} \quad \ldots \ldots (13) \]

Where \( d \) is in cm; \( h_w \) is in cm of water and \( Pa \) is in kg/m³

The density of atmospheric air is given by

\[ \rho_a = \frac{P_a \times 10^5}{287 \times T_a} \]

Where \( P_a \) is the atmospheric pressure in bar and \( T_a \) is the atmospheric temperature in K.

Substituting the value of \( \rho_a \) in Eq. (13)

\[ m_a = 0.066 \cdot C_d \cdot d^2 \sqrt{h_w \frac{p_a \times 10^5}{287 \cdot T_a}} \]

\[ = 1.23 \cdot C_d \cdot d^2 \sqrt{\frac{p_a \cdot h_w}{T_a}} \text{ kg / min} \]

Where \( d \) is in cm, \( h_w \) is in cm of water, \( P_a \) is in bar and \( T_a \) is in K.

The measurement of air consumption by the orifice chamber method is used for:
(a) The determination of the actual A : F ratio of the engine at running condition.
(b) The weight of exhaust gases produced, and
(c) The volumetric efficiency of the engine at the running condition.
The mass of air supplied per kg of fuel used can also be calculated by using the following formula if the volumetric analysis of the exhaust gases is known.
\[ m_a / \text{Kg of fuel} = \frac{N \times C}{33(C_1 + C_2)} \]  

Where

- \( N \) = Percentage of nitrogen by volume in exhaust gases.
- \( C_1 \) = Percentage of carbon dioxide by volume in exhaust gases.
- \( C_2 \) = Percentage of carbon monoxide by volume in exhaust gases.
- \( C \) = Percentage of carbon in fuel by weight.

If \( C_2 = 0 \) then; 
\[ m_a = \frac{N \times C}{33 \cdot C_1} \]  

1.8 MEASUREMENT OF FUEL CONSUMPTION

Two glass vessels of 100cc and 200cc capacity are connected in between the engine and main fuel tank through two, three-way cocks. When one is supplying the fuel to the engine, the other is being filled. The time for the consumption of 100 or 200cc fuel is measured with the help of stop watch.

A small glass tube is attached to the main fuel tank as shown in figure. When fuel rate is to be measured, the valve is closed so that fuel is consumed from the burette. The time for a known value of fuel consumption can be measured and fuel consumption rate can be calculated.

Fuel consumption kg/hr = \( \frac{X \times \text{Sp. gravity of fuel}}{1000 \times t} \)

1.9 MEASUREMENT OF HEAT CARRIED AWAY BY COOLING WATER

The heat carried away by cooling water is generally measured by measuring the water flow rate through the cooling jacket and the rise in temperatures of the water during the flow through the engine.

The inlet and outlet temperatures of the water are measured by the thermometers inserting in the pockets provided at inlet to and outlet from the engine. The quantity of water flowing is measured by collecting the water in a bucket for a specified period or directly with the help of flow meter in case of large engine. The heat carried away by cooling water is given by

\[ Q_w = C_p M_w (T_{wo} - T_{wi}) \text{kJ/min.} \]

Where

- \( Q_w \) = mass of water/min.
- \( T_{wi} \) = Inlet temperature of water, °C
- \( T_{wo} \) = Outlet temperature of water, °C
- \( C_p \) = Specific heat of water.

1.10 MEASUREMENT OF HEAT CARRIED AWAY BY EXHAUST GASES

The mass of air supplied per kg of fuel used can be calculated by using the equation if the exhaust analysis is made

\[ m_a = \frac{N \times C}{33 (C_1 + C_2)} \]
And heat carried away by the exhaust gas per kg of fuel supplied can be calculated as

\[ Q_g = (m_a + 1) C_{pg} (T_{ge} - T_a) \text{ kJ/kg of fuel} \quad \text{...(16)} \]

Where \((m_a + 1)\) = mass of exhaust gases formed per kg of fuel supplied to engine
\(C_{pg}\) = Specific heat of exhaust gases
\(T_{ge}\) = Temperature of exhaust gases coming out from the engine °C.
\(T_a\) = Ambient temperature °C or engine room temperature.

The temperature of the exhaust gases is measured with the help of suitable thermometer or thermocouple.

Another method used for measuring the heat carried away by exhaust gases is to measure the fuel supplied per minute and also to measure the air supplied per minute with the help of air box method. The addition of fuel and air mass will be equal to the mass of exhaust gases.

And exhaust gas calorimeter is commonly used in the laboratory for the measurement of heat carried by exhaust gases.

1.10.1 Exhaust Gas Calorimeter

The exhaust gas calorimeter is a simple heat exchanger in which, part of the heat of the exhaust gases is transferred to the circulating water. This calorimeter helps to determine the mass of exhaust gases coming out of the engine.

The arrangement of the exhaust gas calorimeter is shown in fig. 23.5.

The exhaust gases from the engine exhaust are passed through the exhaust gas calorimeter by closing the valve B and opening the valve A. The hot gases are cooled by the water flow.
rate is adjusted with the help of valve of 'C' to give a measurable temperature rise to water circulated.

If it is assumed that the calorimeter is well insulated, there is no heat loss except by heat transfer from the exhaust gases to the circulating water, then

Heat lost by exhaust gases = Heat gained by circulating water.

Therefore \[ m_g \cdot C_{pg} \left( T_{gi} - T_{go} \right) = m_w \cdot C_{pw} \left( T_{wo} - T_{wi} \right) \]

Where
- \( T_{gi} \) = The temperature of the exhaust gases entering the calorimeter, °C
- \( T_{go} \) = The temperature of the exhaust gases leaving the calorimeter, °C
- \( T_{wi} \) = The temperature of water entering the calorimeter, °C
- \( T_{wo} \) = The temperature of water leaving the calorimeter, °C
- \( m_w \) = Mass of water circulated through the exhaust gas calorimeter, generally measured.
- \( m_g \) = Mass of exhaust gases (unknown)
- \( C_{pg} \) = Specific heat of exhaust gases
- \( C_{pw} \) = Specific heat of water.

\[ \therefore m_g = \frac{C_{pw}}{C_{pg}} \left( \frac{T_{wo} - T_{wi}}{T_{gi} - T_{go}} \right) m_w \]

...(17)

As all the quantities on the RHS are known the gas flow rate can be determined.

Then the heat carried away by the exhaust gases is given by

\[ Q_g = m_g \cdot C_{pg} \cdot (T_{ge} - T_a) \]

...(18)

Where
- \( T_{ge} \) = Temperature of exhaust gases just leaving the engine exhaust valve, °C
- \( T_a \) = Ambient temperature, °C

Usually valve connections are provided as shown in figure so that the exhaust gases are exhausted to the atmosphere during normal operation by closing the valve A and opening the valve B. Only when the apparatus is to be used, the valve A is opened and valve B is closed so that the gases pass through the calorimeter.

The heat carried by the gases is also given by

\[ Q_g = \text{Heat carried by water passing through exhaust gas calorimeter} + \text{Heat in exhaust gases above atmospheric temperature after leaving the exhaust gas calorimeter.} \]

\[ = m_w \cdot C_{pw} \left( T_{wo} - T_{wi} \right) + m_g \cdot C_{pg} \left( T_{go} - T_a \right) \]

...(19)

If sufficient water is circulated to reduce the value of \( T_{go} \) to very near to \( T_a \), then the second term on the RHSs is small and,

\[ Q_g = m_w \cdot C_{pw} \left( T_{wo} - T_{wi} \right) \]

...(20)

## 1.11 HEAT BALANCE SHEET

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

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

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

\[ Q_s = m_f \cdot X \cdot CV \]

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

The various ways in which heat is used up in the system is given by
(a) Heat equivalent of BP = kW = kJ/sec. = 0 kJ/min.
(b) Heat carried away by cooling water
\[ = C_{pw} \times m_w (T_{wo} - T_{wi}) \text{ kJ/min.} \]

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

(c) Heat carried away by exhaust gases
\[ = m_g C_{pg} (T_{ge} - T_a) \text{ (kJ/min.) or (kJ/sec)} \]

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

\[ T_g = \text{Temperature of burnt gases coming out of the engine.} \]
\[ T_a = \text{Ambient Temperature.} \]
\[ C_{pg} = \text{Sp. Heat of exhaust gases in (kJ/kg-K)} \]

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

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

<table>
<thead>
<tr>
<th>Heat input per minute</th>
<th>kcal (kj)</th>
<th>%</th>
<th>Heat expenditure per minute</th>
<th>kcal (kj)</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat supplied by the combustion fuel</td>
<td>( Q_s )</td>
<td>100%</td>
<td>(a) Heat in BP.</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(b) Heat carried by jacket cooling water</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(c) Heat Carried by exhaust gases</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(d) Heat unaccounted for ( = Q_s - (a + b + c) )</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>Total</td>
<td>( Q_s )</td>
<td>100%</td>
<td></td>
<td></td>
<td>100%</td>
</tr>
</tbody>
</table>

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

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

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

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

\[
\text{isfc} = \frac{3600 \text{ m}}{\text{ip}} \text{ kg/kWh} \quad \ldots(13)
\]

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

\[
\text{bsfc} = \frac{3600 \text{ m}}{\text{bp}} \text{ kg/kWh} \quad \ldots(14)
\]

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

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

Indicated Power in kW

Thus \( \eta_{\text{ith}} = \frac{\text{Indicated Power in kW}}{(\text{Mass flow rate of fuel in kg/s}) \times (\text{Calorific value of fuel in kJ/kg})} \)

\[
\eta_{\text{ith}} = \frac{\text{ip}}{\text{m} \times \text{CV}} \quad \ldots(15)
\]

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

\[
\eta_{\text{bth}} = \frac{\text{bp}}{\text{m} \times \text{CV}} \quad \ldots(16)
\]
1.15. Mechanical Efficiency: Mechanical efficiency takes into account the mechanical losses in an engine. The mechanical losses include (i) frictional losses, (ii) power absorbed by engine auxiliaries like fuel pump, lubricating oil pump, water circulating pump, magneto and distributor, electric generator for battery charging, radiator fan etc., and (iii) work required to charge the cylinder with fresh charge and work for discharging the exhaust gases during the exhaust stroke. It is defined as the ratio of brake power to indicated power. Thus

\[
\eta_{\text{mech}} = \frac{\text{bp}}{\text{ip}}
\]

(17)

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

Thus

\[
\eta_v = \frac{V_{\text{actual}}}{V_{\text{th}}}
\]

(18)

where \( n \) is the number of intake strokes per minute and \( V_s \) is the stroke volume of the piston.

2. Illustrative examples:

Example 1:- The following observations have been made from the test of a four cylinder, two – stroke petrol engine. Diameter of the cylinder = 10 cm; stroke = 15 cm; speed = 1600 rpm; Area of indicator diagram = 5.5 cm²; Length of the indicator diagram = 55 mm; spring constant = 3.5 bar/cm; Determine the indicated power of the engine.

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

To find: indicated power, ip.
Solution: Indicated mean effective pressure = \( p_{im} = \frac{a k_s}{I_d} \)

or \( p_{im} = \frac{5.5 \times 3.5}{5.5} \) bar = 3.5 \( x 10^5 \) N / m²

\( p_{im} \) LAnK \( 3.5 \times 10^5 \times 0.15 \times \left( \frac{\pi}{4} \right) \times 0.1^2 \times 1600 \times 4 \)

Indicated power = \( ip = \frac{60,000}{60,000} \)

= 43.98 kW

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

Given: Volume of fuel consumed/hour = \( y/t = 8 \times 10^3 / 3600 \) cc/s;

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

Specific gravity of petrol = \( s = 0.75 \)

To find: \( \eta_{th} \);

Solution: Mass of fuel consumed = \( m = \frac{y \times 8 \times 10^3 \times 0.75}{1000 \times 3600} = 1.67 \times 10^{-3} \) kg/s.

Indicated thermal efficiency = \( \eta_{th} = \frac{ip}{m \times CV} \)

= 0.3402 = 34.02 %.

Example 2.3: The bore and stroke of a water cooled, vertical, single-cylinder, four stroke diesel engine are 80 mm and 110 mm respectively. The torque is 23.5 N-m. Calculate the brake mean effective pressure. What would be the mean effective pressure and torque if the engine rating is 4 kW at 1500 rpm?

Given: Diameter = \( d = 80 \times 10^{-3} = 0.008 \) m; stroke = \( L = 0.110 \) m; \( T = 23.5 \) N-m;

To find \( i) \) bmep; \( ii) \) bmep if bp = 4 kW and \( N = 1500 \) rpm.
Solution: (i) Relation between brake power (bp) and brake mean effective pressure (bme) is given by

\[
\frac{2\pi NT}{60,000} = \frac{(bme)LAn}{60,000}
\]

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

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

(ii) when \( bp = 4 \text{ kw} \) and \( N = 1500 \text{ rpm} \), we have

\[
\frac{60,000 \text{ bp}}{LAn} = \frac{60,000 \times 4}{0.110 \times (\pi/4) \times 0.08^2 \times (1500 / 2)} = 5.79 \times 10^5 \text{ N/m}^2 = 5.79 \text{ bar}
\]

Also \( bp = \frac{2\pi NT}{60,000} \) or \( T = \frac{60,000 \text{ bp}}{2 \pi N} = \frac{60,000 \times 4}{2 \times \pi \times 1500} = 25.46 \text{ N} \cdot \text{m} \).

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

Given: \( y = 10 \text{ cc} \); \( t = 20.4 \text{ s} \); \( V_a = 0.1 \text{ m}^3 \); \( t_a = 16.3 \text{ s} \); \( W = 7 \text{ N} \); \( N = 3000 \text{ rpm} \);
\( \rho_a = 1.175 \text{ kg/m}^3 \); \( s = 0.7 \); \( CV = 43 \times 10^3 \text{ kJ/kg} \); Dynamometer constant = \( C = 5000 \).

To find: (i) \( m_a/m_f \); (ii) bsfc; (iii) \( \eta_{bth} \).

Solution: (i) Mass of air consumed = \( m_a = \frac{0.1 \times 1.175}{16.3} \text{ kg/s} \).
Mass of fuel consumed = \( m_f = \frac{10 \times 0.7}{1000 \times 20.4} \text{ kg/s} \).

\[
\frac{y}{s} = 10 \times 0.7
\]

\[
\frac{m_a}{m_f} = \frac{7.21 \times 10^{-3}}{0.343 \times 10^{-3}} = 20.4
\]

\[
\eta_{bth} = \frac{7.21 \times 10^{-3}}{0.343 \times 10^{-3}} = 20.4
\]
\[
\frac{m_a}{m_f} = \frac{7.21 \times 10^{-3}}{0.343 \times 10^{-3}} = 21
\]

Air fuel ratio = \[\frac{m_a}{m_f}\] = 21

\[
7 \times \frac{3000}{5000} = \frac{m_f \times 3600}{0.343 \times 10^{-3} \times 3600}
\]

\[
\text{bsfc} = \frac{0.343 \times 10^{-3} \times 3600}{4.2} = 0.294 \text{ kg/kWh}
\]

\[
\text{bp} = 4.2
\]

\[
\frac{m_f \times 3600}{0.343 \times 10^{-3} \times 43 \times 10^3} = 0.2848 = 28.48\%
\]

Example 2.5:- A six cylinder, gasoline engine operates on the four stroke cycle. The bore of each cylinder is 80 mm and the stroke is 100 mm. The clearance volume in each cylinder is 70 cc. At a speed of 4000 rpm and the fuel consumption is 20 kg/h. The torque developed is 150 N-m. Calculate (i) the brake power, (ii) the brake mean effective pressure, (iii) brake thermal efficiency if the calorific value of the fuel is 43000 kJ/kg and (iv) the relative efficiency if the ideal cycle for the engine is Otto cycle.

Given:- K = 6 ; n = N/2 ; d = 8 cm ; L = 10 cm ; V_c = 70 cc ; N = 4000 rpm ; m_f = 20

kg/h ; T = 150 N-m ; CV = 43000 kJ/kg ;

To find:- (i) bp ; (ii) bmep ; (iii) \(\eta_{\text{th}}\) ; (iv) \(\eta_{\text{Relative}}\).

Solution:

\[
2\pi \frac{NT2}{\pi} \times 4000 \times 150 = \frac{bp}{60,000} = \frac{60,000}{60,000}
\]

\[= 62.8 \text{ kW}\]

\[
\frac{60,000 \text{ bp}}{0.1 \times (\pi/4) \times 0.08^2 \times (4000/2) \times 6} = \frac{6.25 \times 10^5 \text{ N/m}^2}{6.25 \text{ bar}} = \frac{62.8}{62.8}
\]

\[= 0.263 = 26.3\%\]

\[
\eta_{\text{th}} = \frac{m_f \times CV}{(20/3600) \times 43,000}
\]
(iv) Stroke volume = \( V_s = \left( \frac{\pi}{4} \right) d^2 \) 
\[ L = \left( \frac{\pi}{4} \right) \times 8^2 \times 10 = 502.65 \text{ cc} \]
\[ \frac{V_s + V_c}{V_c} = \frac{502.65 + 70}{70} = 8.18 \]

Compression Ratio of the engine = \( R_c = \frac{V_s + V_c}{V_c} = 8.18 \)

Air standard efficiency of Otto cycle = \( \eta_{Otto} = 1 - \left( \frac{1}{R_c^{\gamma - 1}} \right) \)
\[ = 1 - \frac{1}{8.18^{0.4}} = 0.568 = 56.8 \% \]

Hence Relative efficiency = \( \eta_{Relative} = \frac{\eta_{bth}}{\eta_{Otto}} = \frac{0.263}{0.568} = 0.463 = 46.3 \% \)

**Example 2.6:** An eight cylinder, four stroke engine of 9 cm bore, 8 cm stroke and with a compression ratio of 7 is tested at 4500 rpm on a dynamometer which has 54 cm arm. During a 10 minute test, the dynamometer scale beam reading was 42 kg and the engine consumed 4.4 kg of gasoline having a calorific value of 44,000 kJ/kg. Air at 27 C and 1 bar was supplied to the carburetor at a rate of 6 kg/min. Find (i) the brake power, (ii) the brake mean effective pressure, (iii) the brake specific fuel consumption, (iv) the brake specific air consumption, (v) volumetric efficiency, (vi) the brake thermal efficiency and (vii) the air fuel ratio.

**Given:** \( K = 8 \); Four stroke hence \( n = N/2 \); \( d = 0.09 \text{ m} \); \( L = 0.08 \text{ m} \); \( R_c = 7 \); \( N = 4500 \) rpm; Brake arm = \( R = 0.54 \text{ m} \); \( t = 10 \text{ min} \); Brake load = \( W = (42 \times 9.81) \text{ N} \)
\( m_f = 4.4 \text{ kg} \); \( CV = 44,000 \text{ kJ/kg} \); \( T_a = 27 + 273 = 300 \text{ K} \); \( p_a = 1 \text{ bar} \); \( m_a = 6 \text{ kg/min} \);

**To find:** (i) \( \text{bp} \); (ii) \( \text{bmeq} \); (iii) \( \text{bsfc} \); (iv) \( \text{bsac} \); (v) \( \eta_v \); (vi) \( \eta_{bth} \); (vii) \( \frac{m_a}{m_f} \)

**Solution:**
\[
\begin{array}{ccc}
2\pi \text{ NT} & 2\pi \text{ NWR} & 2 \times \pi \times 4500 \times (42 \times 9.81) \times 0.54 \\
60,000 & 60,000 & 60,000 \\
\end{array}
\]
\[ = 104.8 \text{ kW} \]
\[
\begin{array}{ccc}
60,000 \text{ bp} & 60,000 \times 104.8 \\
L \text{ A n K} & 0.08 \times (\pi/4) \times 0.09^2 \times (4500/2) \times 8 \\
\end{array}
\]
\[ = 6.87 \times 10^{-5} \text{ N/m}^2 = 6.87 \text{ bar} \]
(iii) mass of fuel consumed per unit time = \( m_f = \frac{m_f}{t} = \frac{4.4 \times 60}{10} \) kg/h

\[ = 26.4 \text{ kg/h} \]

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

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

\[ = 3.435 \text{ kg / kWh} \]

(v) \( \eta_{bth} = \frac{104.8}{(26.4 / 3600) \times 44000} = 0.325 = 32.5\% \)

(vi) Stroke volume per unit time = \( \dot{V_s} = \frac{\pi d^2}{4} \) L / min

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

\[ = 9.16 \text{ m}^3 / \text{min.} \]

\[ \text{Volume flow rate of air per minute} = \dot{V_a} = \frac{m_a R_a T_a}{p_a 1 \times 10^5} = \frac{6 \times 286 \times 300}{5.17} \]

\[ = 5.17 \text{ m}^3 / \text{min} \]

\[ \text{Volumetric efficiency} = \eta_v = \frac{\dot{V_a}}{\dot{V_s}} = \frac{5.17}{9.16} = 0.5644 = 56.44\% \]

(vii) Air fuel ratio = \( \frac{m_a}{m_f} = \frac{6 / (4.4 / 10)}{10} = 13.64 \)

**Example 2.7:** A gasoline engine working on four-stroke develops a brake power of 20.9 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 plug 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 brake mean effective pressure when all the cylinders are firing. The bore of the engine is 75 mm and the stroke is 90 mm. The engine is running at 3000 rpm.
Given: - brake power when all cylinders are working = B_t = 20.9 kW;

Brake power when cylinder 1 is inoperative = B_1 = 14.9 kW;
Brake power when cylinder 2 is inoperative = B_2 = 14.3 kW;
Brake power when cylinder 3 is inoperative = B_3 = 14.8 kW;
Brake power when cylinder 4 is inoperative = B_4 = 14.5 kW;

N = 3000 rpm; d = 0.075 m; L = 0.09 m;

To find: - (i) (ip)_{total}; (ii) \eta_{mech}; (iii) bmep;

Solution:

(i) (ip)_{total} = ip_1 + ip_2 + ip_3 + ip_4 = (B_1 - B_1) + (B_1 - B_2) + (B_1 - B_3) + (B_1 - B_4)

= 4B_1 - (B_1 + B_2 + B_3 + B_4) = 4 \times 20.9 - (14.9 + 14.3 + 14.8 + 14.5)

= 25.1 Kw

\begin{align*}
\frac{B_t}{(ip)_{total}} &= \frac{20.9}{25.1} = 0.833 = 83.3\% \\
(\text{ii}) \eta_{mech} &= \frac{\text{B}_t}{(ip)_{total}} = \frac{20.9}{25.1} = 0.833 = 83.3\%
\end{align*}

(iii) bmep = \frac{60,000 \times B_t}{L \times \pi \times (\frac{d}{2})^2 \times \frac{N}{2}} = \frac{60,000 \times 20.9}{0.09 \times \pi \times (0.075)^2 \times \frac{3000}{2} \times 4}

= 5.25 \times 10^5 \text{ N/m}^2 = 5.25 \text{ bar.}

Example 2.8:- The following observations were recorded during a trial of a four-stroke, single cylinder oil engine.

- Duration of trial = 30 min; oil consumed = 4 litres; calorific value of oil = 43 MJ/kg;
- Specific gravity of fuel = 0.8; average area of the indicator diagram = 8.5 cm²; length of the indicator diagram = 8.5 cm; Indicator spring constant = 5.5 bar/cm; brake load = 150 kg; spring balance reading = 20 kg; effective brake wheel diameter = 1.5 m; speed = 200 rpm; cylinder diameter = 30 cm; stroke = 45 cm; jacket cooling water = 10 kg/min; temperature rise of cooling water = 36°C. Calculate (i) indicated power, (ii) brake power, (iii) mechanical efficiency, (iv) brake specific fuel consumption, (v) indicated thermal efficiency, and (vi) heat carried away by cooling water.
Given: \( t = 30 \text{ min} \); \( y = 4000 \text{ cc} \); \( CV = 43 \times 10^3 \text{ kJ/kg} \); \( s = 0.8 \); area of the diagram = \( a = 8.5 \text{ cm}^2 \); length of the diagram = \( l_d = 8.5 \text{ cm} \); indicator spring constant = \( k_s = 5.5 \text{ bar/cm} \); \( W = 150 \times 9.81 \text{ N} \); Brake radius = \( R = 1.5 / 2 = 0.75 \text{ m} \); \( N = 200 \text{ rpm} \); \( d = 0.3 \text{ m} \); \( L = 0.45 \text{ m} \); \( m_w = 10 \text{ kg/min} \); \( \Delta T_w = 36 \text{ C} \); Spring Balance Reading = \( S = 20 \times 9.81 \text{ N} \)

To find: (i) \( ip \); (ii) \( bp \); (iii) \( \eta_{\text{mech}} \); (iv) \( \text{bsfc} \); (v) \( \eta_{\text{th}} \); (vi) \( Q_w \)

Solution:
\[
\begin{align*}
\text{(i) } & \frac{a}{8.5} = \frac{1}{l_d} = 8.5 \quad \frac{p_m}{L A n K} = 5.5 \times 10^5 \times 0.45 \times (\pi / 4) \times 0.3^2 \times (200 / 2) \times 1 \\
& \text{ip} = \frac{60,000}{60,000} = 0.687 = 68.7 \%
\end{align*}
\]

\[
\begin{align*}
\text{(ii) } & \frac{2\pi N(W - S) R}{60,000} = 2 \times \pi \times 200 \times (150 - 20) \times 9.81 \times 0.75 \\
& \text{bp} = \frac{60,000}{60,000} = 0.3195 \text{ kg/kWh}
\end{align*}
\]

\[
\begin{align*}
\text{(iv) Mass of fuel consumed per hour} & = \frac{m_f}{60} = \frac{600 \times 0.8}{60} \\
& = 6.4 \text{ kg/h.}
\end{align*}
\]

\[
\begin{align*}
\text{bsfc} & = \frac{m_f / \text{bp}}{20.03} = 0.3195 \text{ kg/kWh}
\end{align*}
\]

\[
\begin{align*}
\text{(v) } & \frac{\text{ip}}{29.16} = \frac{29.16}{20.03} = 0.3814 = 38.14 \%
\end{align*}
\]

\[
\begin{align*}
\text{(vi) } & \dot{Q}_w = m \cdot C_p \Delta T_w = (10 / 60) \times 4.2 \times 36 = 25.2 \text{ kW}
\end{align*}
\]
Example 2.9:- A four stroke gas engine has a cylinder diameter of 25 cm and stroke 45 cm. The effective diameter of the brake is 1.6 m.
The observations made in a test of the engine were as follows.
Duration of test = 40 min; Total number of revolutions = 8080; Total number of explosions = 3230; Net load on the brake = 80 kg; mean effective pressure = 5.8 bar;
Volume of gas used = 7.5 m³; Pressure of gas indicated in meter = 136 mm of water (gauge); Atmospheric temperature = 17 C; Calorific value of gas = 19 MJ/ m³ at NTP;
Temperature rise of cooling water = 45 C; Cooling water supplied = 180 kg.

Draw up a heat balance sheet and find the indicated thermal efficiency and brake thermal efficiency. Assume atmospheric pressure to be 760 mm of mercury.

Given:-
\( d = 0.25 \text{ m} \); \( L = 0.45 \text{ m} \); \( R = 1.6 \div 2 = 0.8 \text{ m} \); \( t = 40 \text{ min} \); \( N_{\text{total}} = 8080 \);

Hence \( N = \frac{8080}{40} = 202 \text{ rpm} \)
\( n_{\text{total}} = 3230 \);

Hence \( n = \frac{3230}{40} = 80.75 \text{ explosions / min} \);
\( W = 80 \times 9.81 \text{ N} \);
\( p_{\text{in}} = 5.8 \text{ bar} \);

\( V_{\text{total}} = 7.5 \text{ m}^3 \);

\( V = \frac{7.5}{40} = 0.1875 \text{ m}^3/\text{min} \);
\( p_{\text{gauge}} = 136 \text{ mm of water (gauge)} \);

\( T_{\text{atm}} = 17 + 273 = 290 \text{ K} \);
\( (CV)_{\text{NTP}} = 19 \times 10^3 \text{ kJ/ m}^3 \);
\( \Delta T_w = 45 \text{ C} \);

\( m_w = \frac{180}{40} = 4.5 \text{ kg/min} \);
\( p_{\text{atm}} = 760 \text{ mm of mercury} \)

To find:- (i) \( \eta_{\text{tith}} \); (ii) \( \eta_{\text{bth}} \); (iii) heat balance sheet

Solution:

\[
p_{\text{in}} = \frac{L A n K}{60,000} = \frac{5.8 \times 10^5 \times (\pi / 4) \times 0.25^2 \times 0.45 \times 80.75}{60,000} = 17.25 \text{ kW}.
\]

\[
bp = \frac{2\pi N \frac{W R}{60,000}}{60,000} = \frac{2 \times \pi \times 202 \times (80 \times 9.81) \times 0.8}{60,000} = 13.28 \text{ kW}
\]

Pressure of gas supplied = \( p = p_{\text{atm}} + p_{\text{gauge}} = 760 + 136 / 13.6 = 770 \text{ mm of mercury} \)

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

\[
= \frac{0.1875 \times 273 \times 770}{290 \times 760} = 0.17875 \text{ m}^3 / \text{min}
\]
Heat supplied by fuel = \( \dot{Q}_f = \dot{V}_{\text{NTP}} \cdot (CV)_{\text{NTP}} = 0.17875 \times 19 \times 10^3 = 3396.25 \text{ kJ/min} \)

Heat equivalent of bp in kJ/min = 13.28 \times 60 = 796.4 \text{ kJ/min} 

Heat lost to cooling water in kJ/min = \( m_w C_p \Delta T_w = 4.5 \times 4.2 \times 45 = 846.5 \text{ kJ/min} \)

Friction power = \( \text{ip} - \text{bp} = 17.25 - 13.28 = 3.97 \text{ kW} \)

Hence heat loss due to friction, pumping etc. = 3.97 \times 60 = 238.2 \text{ kJ/min} 

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

= 1465.15 \text{ kJ/min} 

**Heat Balance Sheet:**

<table>
<thead>
<tr>
<th>Item No.</th>
<th>Heat Energy Input (kJ/min)</th>
<th>Heat Energy Input (percent)</th>
<th>Heat Energy spent (kJ/min)</th>
<th>Heat Energy spent (percent)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Heat supplied by fuel</td>
<td>3396.25</td>
<td>100.0</td>
<td>3396.25</td>
</tr>
<tr>
<td>2</td>
<td>Heat equivalent of bp</td>
<td></td>
<td></td>
<td>896.4</td>
</tr>
<tr>
<td>3</td>
<td>Heat lost to cooling water</td>
<td></td>
<td></td>
<td>796.4</td>
</tr>
<tr>
<td>4</td>
<td>Heat equivalent of fp</td>
<td></td>
<td></td>
<td>238.2</td>
</tr>
<tr>
<td>5</td>
<td>Heat unaccounted (by difference)</td>
<td></td>
<td></td>
<td>1465.15</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td>3396.25</td>
<td>100.0</td>
<td>3396.25</td>
</tr>
</tbody>
</table>

**Example 2.10:** A test on a two-stroke engine gave the following results at full load.

Speed = 350 rpm; Net brake load = 65 kg ; mean effective pressure = 3 bar ; Fuel consumption = 4 kg/h ; Jacket cooling water flow rate = 500 kg/h ; jacket water temperature at inlet = 20 C ; jacket water temperature at outlet = 40 C ; Test room temperature = 20 C ; Temperature of exhaust gases = 400 C; Air used per kg of fuel = 32 kg ; cylinder diameter = 22 cm; stroke = 28 cm; effective brake diameter = 1 m ; 
Calorific value of fuel = 43 MJ/kg ; Mean specific heat of exhaust gases = 1 kJ/kg –K. Find indicated power, brake power and draw up a heat balance for the test in kW and in percentage.
**Given:**- Two stroke engine. Hence \( n = N \); \( N = 350 \text{ rpm} \); \( W = (65 \times 9.81) \text{ N} \);
\[ p_{im} = 3 \text{ bar} \; ; \; m_r = 4 \text{ kg/h} \; ; \; m_w = 500 \text{ kg/h} \; ; \; T_{wi} = 20 \text{ C} \; ; \; T_{wo} = 40 \text{ C} \; ; \; T_{atm} = 20 \text{ C} \; ;
\[ T_{eg} = 400 \text{ C} \; ; \; m_a / m_r = 32 \; ; \; d = 0.22 \text{ m} \; ; \; L = 0.28 \text{ m} \; ; \; \text{Brake radius} = R = \frac{1}{2} \text{ m} \; ;
\[ CV = 43,000 \text{ kJ/kg} \; ; \; (C_p)_{eg} = 1.0 \text{ kJ/(kg-K)} \; ;
\]

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

**Solution:**

(i) **ip** = \( p_{im} \times L \times A \times n \times 10^5 \times 0.28 \times (\pi/4) \times 0.22^2 \times 350 \)
\[ = \frac{18.63}{60,000} \text{ kW}. \]

(ii) **bp** = \( 2\pi NWR \times 2 \times \pi \times 350 \times (65 \times 9.81) \times 0.5 \)
\[ = \frac{11.68}{60,000} \text{ kW}. \]

(iii) Heat supplied in kW = \( m_r \times CV = (4 / 3600) \times 43,000 \)
\[ = 47.8 \text{ kW} \]

Heat lost to cooling water = \( m_w (C_p)_w [T_{wo} - T_{wi}] \)
\[ = (500 / 3600) \times 4.2 \times [40 - 20] \]
\[ = 11.7 \text{ kW}. \]

Heat lost in exhaust gases = \( (m_a + m_r) (C_p)_{eg} [T_{eg} - T_{atm}] \)
\[ = \frac{(32 + 1) \times 4}{3600} \times 1.0 \times [400 - 20] \]
\[ = 13.9 \text{ kW} \]
**Heat balance sheet:**

<table>
<thead>
<tr>
<th>Heat Input</th>
<th>kW</th>
<th>%</th>
<th>Heat Expenditure</th>
<th>kW</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat supplied by fuel</td>
<td>47.8</td>
<td>100</td>
<td>Heat in bp</td>
<td>11.68</td>
<td>24.4</td>
</tr>
<tr>
<td>Heat lost to cooling Water</td>
<td></td>
<td></td>
<td></td>
<td>11.70</td>
<td>24.5</td>
</tr>
<tr>
<td>Heat lost to exhaust Gases</td>
<td></td>
<td></td>
<td></td>
<td>13.90</td>
<td>29.1</td>
</tr>
<tr>
<td>Unaccounted heat (by difference)</td>
<td></td>
<td></td>
<td></td>
<td>10.52</td>
<td>22.0</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>47.80</td>
<td>100</td>
<td><strong>Total</strong></td>
<td>47.80</td>
<td>100.0</td>
</tr>
</tbody>
</table>