UNIT 7 IC ENGINE TESTING

Structure

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 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,

$$bp = \frac{2\pi\ NT}{60} \qquad \ldots (7.1)$$

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 inducting the air and removing the products of combustion from the engine combustion chamber.

Indicated Power

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

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

where, p_m = Mean effective pressure, N/m^2,

L = Length of the stroke, m,

A = Area of the piston, m^2,

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

k = Number of cylinders.

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

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

$$\text{Mechanical efficiency } = \frac{bp}{ip} \qquad \ldots (7.2)$$

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

$$fp = ip - bp \qquad \ldots (7.3)$$

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

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,

$$p_m = \frac{ip \times 60}{LANk} \qquad \ldots (7.5)$$

where, P_m = Mean effective pressure, N/m^2,

I_p = Indicated power, Watt,

L = Length of the stroke, m,

A = Area of the piston, m^2,

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

k = Number of cylinders.

If the mean effective pressure is based on *bp* it is called the brake mean effective pressure (*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,

$$fmap = imep - bmep \qquad \ldots (7.6)$$

The torque is related to mean effective pressure by the relation

$$bp = \frac{2\pi\ NT}{60} \qquad \ldots (7.7)$$

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

By Eq. (5.5),

$$\frac{2\pi\, NT}{60} = \left(bemp\ .\ A\ .\ L\ .\ \frac{Nk}{60} \right)$$

or, $$T = \frac{(bemp\ .\ A\ .\ L\ .\ k)}{2\pi} \qquad \ldots (7.8)$$

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,

$$\text{Specific output} = \frac{bp}{A \times L}$$

$$= \text{Constant} \times bmep \times rpm \qquad \ldots (7.9)$$

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

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

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

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*)

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.

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

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.

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

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.

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

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 Burrette 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.

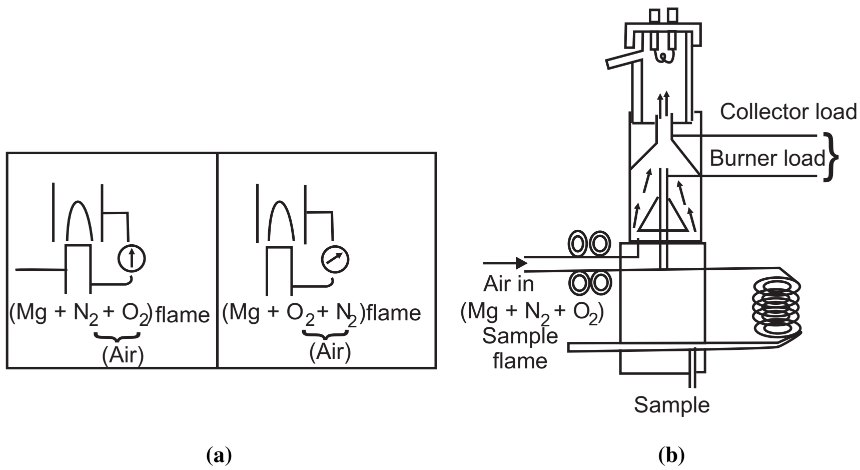

Figure 7.1 : Flame Ionization Detector Burner

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, NOx, 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.

Dispersive 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.

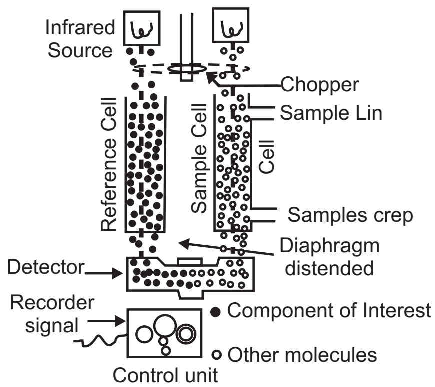

Figure 7.2 : Schematic of Non-dispersive Infra-red Analyzer (NDIR)

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, CO_2 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 .

$$W = 2\,\pi RF$$

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.

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

$\therefore$ Work done/revolution = $2\pi\, SL$

Work done/minute = $2\pi\, SLN$

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

Brake power $P = 2\pi\, NT$

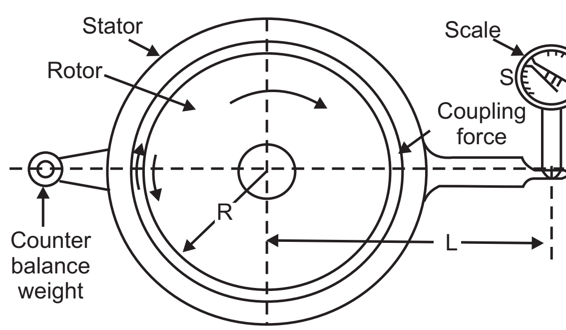

Figure7.3 : Principle of a Dynamometer

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 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 the 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.

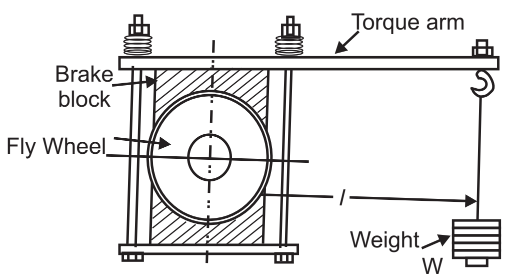

Figure 7.4 : Prony Brake

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.

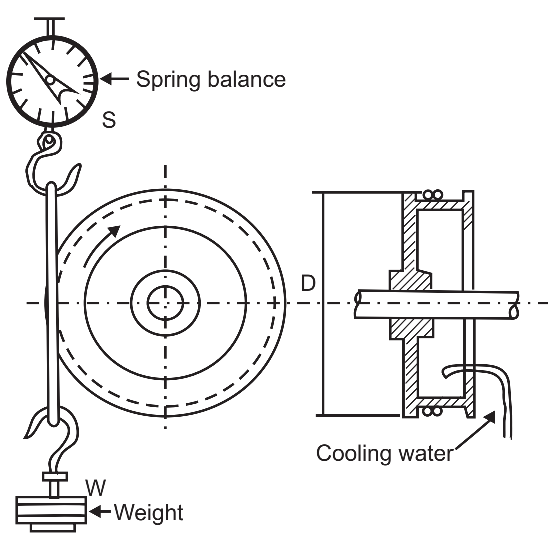

Figure 7.5 : Rope Brake

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.

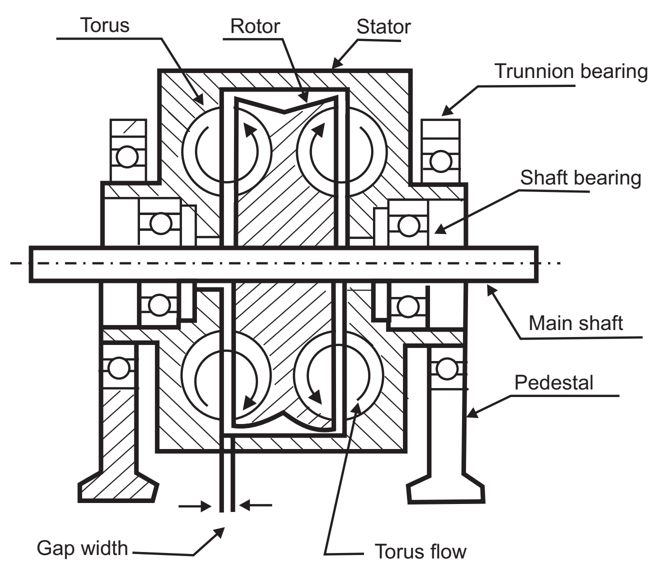

Figure 7.6 : Hydraulic Dynamometer

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.

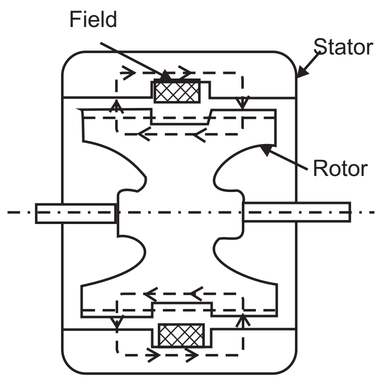

Figure 7.7 : Eddy Current Dynamometer

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 there action torque that the outer case and filed 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.

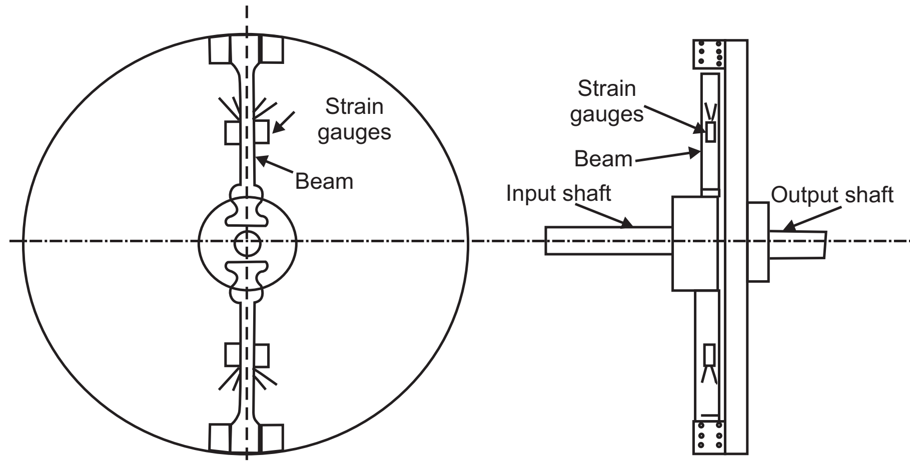

Figure 7.8 : Transmission Dynamometer

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 :

(a) Willan’s line method.

(b) Morse test.

(c) Motoring test.

(d) 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.

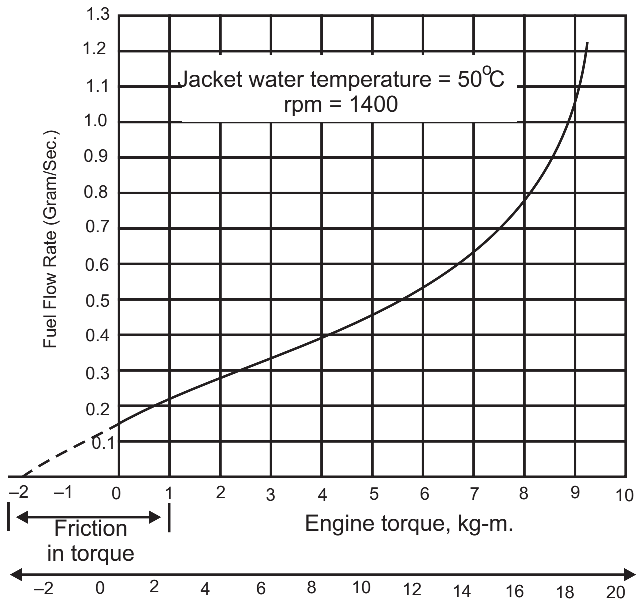

Figure 7.9 : Willan's Line Method

- 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

$$ip_n = bp_n + fp \qquad \ldots (7.17)$$

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

$$ip_{n-1} = bp_{n-1} + fp \qquad \ldots (7.18)$$

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

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

$$(ip)\, n^{th} = bp_n - bp_{n-1} \qquad \ldots (7.19)$$

and the total *ip* of the engine is,

$$hp_n = \Sigma\, (ihp)\, n^{th} \qquad \ldots (7.20)$$

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 sever in air-cooled engines.

(b) The pressure on the bearings and piston rings is lower than the firing pressure. Load on main and connecting road 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' 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.

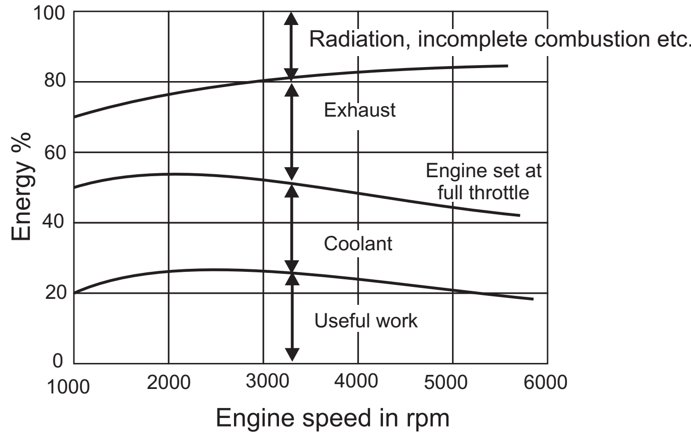

Figure 7.10 : Heat Balance Vs. Speed for a Petrol Engine at Full Throttle

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

Table7.1 : 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.

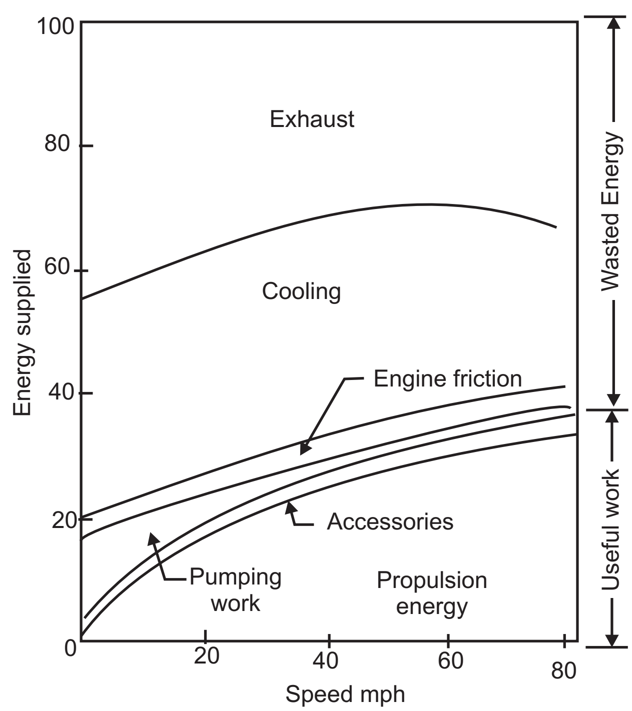

Figure 7.11 : Uncontrolled Otto Engine

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

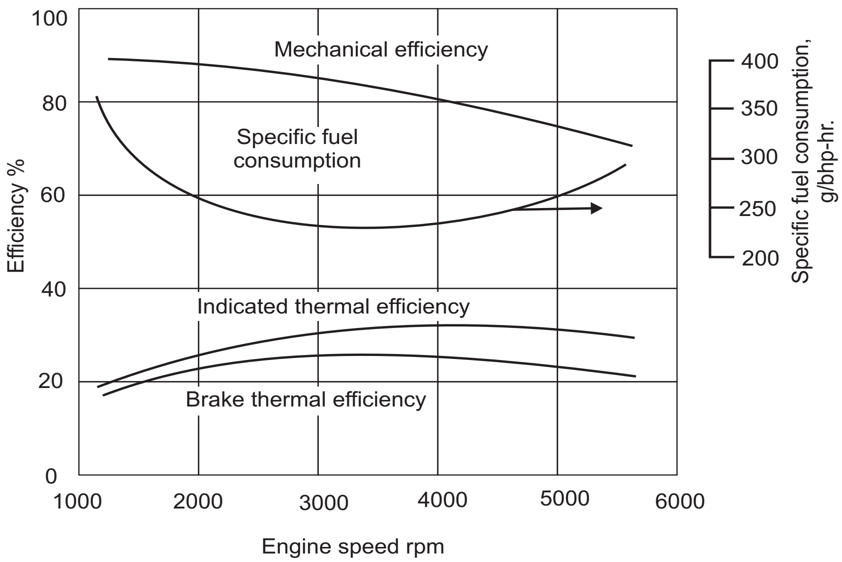

Figure 7.12 : Efficiency and Specific Fuel Consumption Vs.

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

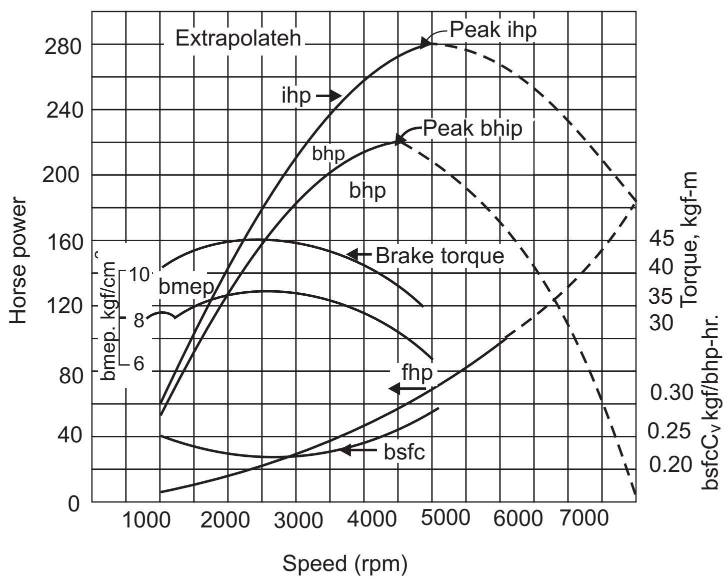

Figure 7.13 : Variable Speed Test of Automotive SI Engine at Full Throttle (*CR* = 9)

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

(a) At full throttle the brake thermal efficiency at various speeds varies from 20 to 27 percent, maximum efficiency being at the middle speed range.

(b) 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.

(c) 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 *bmep*, *bp* and torque directly increase with load, as shown in Figure 7.16. Unlike the SI engine *bhp* and *bmep* 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.

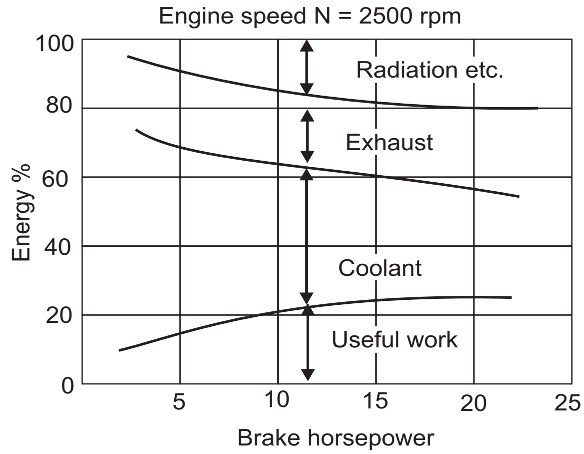

Figure 7.14 : Heat Balance Vs. Load for a Petrol Engine

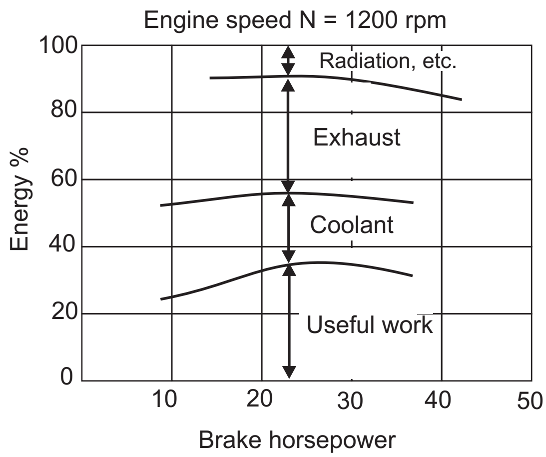

Figure 7.15 : Heat Balance Vs. Load for a CI Engine

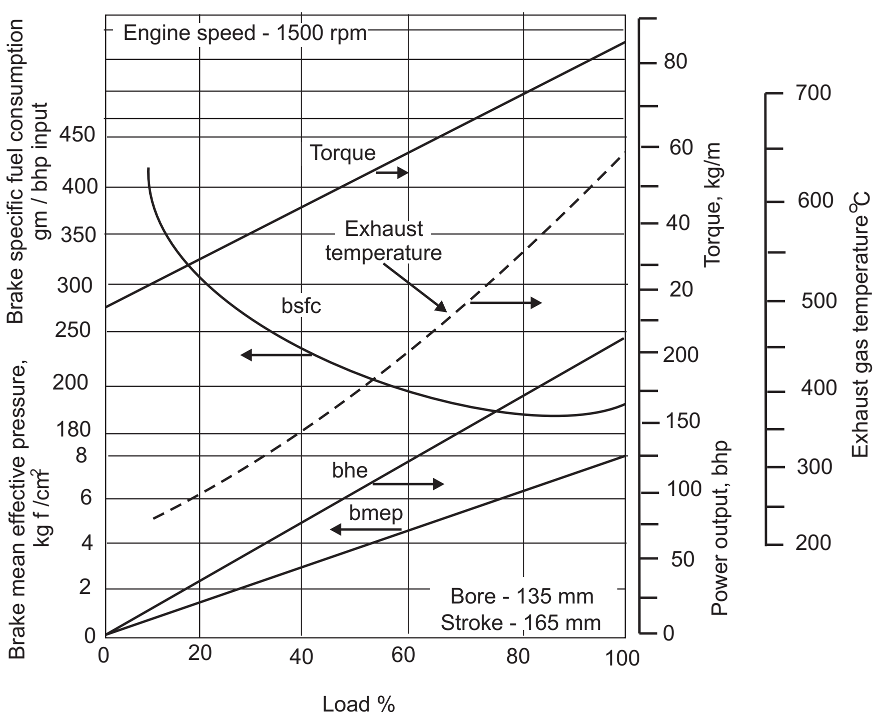

Figure 7.16 : Performance Curves of a Six Cylinder Four-stroke Cycle Automotive Type CI Engine at Constant Speed

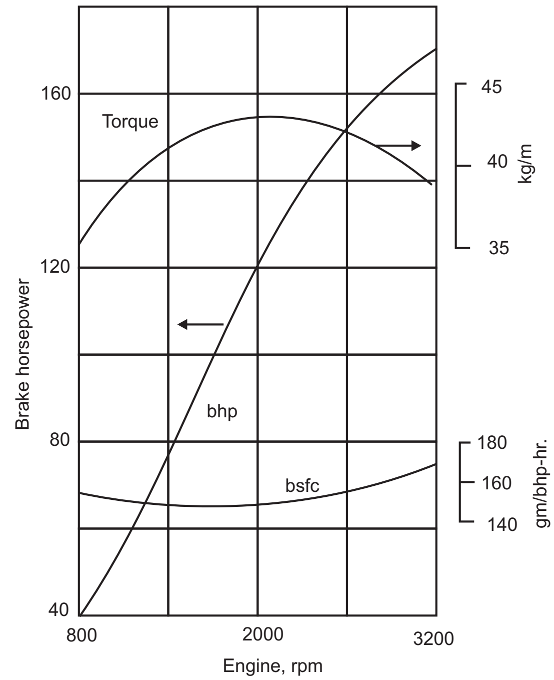

Figure 7.17 : Performance Curves of GM-four Cycle Toro-flow Diesel Engine

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

$$= m\, C_v$$

$$= 8 \times \frac{0.8}{60 \times 60}$$

$$= \frac{6.4}{3600}$$

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

$$\eta_{ith} = \frac{\text{Output power}}{\text{Input power}}$$

$$= \frac{25}{\dfrac{6.4 \times 44000}{3600}}$$

$$= \frac{25 \times 3600}{6.4 \times 44000} = 0.3196$$

or, $= 31.96\%$

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

$$\text{Brake power } = \frac{2\pi\ NT}{60000} = \frac{2\times\pi\times 2000\times 8}{60000}$$

$$= 1.6746 \text{ kW}$$

Friction power = 2.0 − 1.6746

= 0.3253

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

% 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

$$bsec = \frac{\text{kW heat input}}{\text{kW heat output}}$$

$$= \frac{C_v \times m_f}{P} = C_v \times bsfc$$

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

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

C_v = 44 MJ/kg = 44 × 10^3 kJ/kg

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

$isec = bsec \times \eta_n = 3.256 \times 0.85$

$isec = 2.7676.$

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 m^3 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/m^3 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

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

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

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

$$\text{Power output } (P) = \frac{WN}{\text{Dynamometer constant}}$$

$$= \frac{16\times 3000}{5000} = 9.6 \text{ kW}$$

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

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

$bsfc$ = 131.25 g/kWh

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

η_{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

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

$$P_{bm} = \frac{bp \times 6000}{LA\,n\,K}$$

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

$$= 6.66 \times 10^5 \text{ P}_\text{a}$$

Pb_m = 6.66 bar

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

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

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

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,

$$V_s = \frac{\pi}{4} D^2 L n K$$

$$= \frac{\pi}{4} \times (0.1)^2 \times 0.9 \times \frac{4500}{2} \times 9$$

$$= 127.17 \text{ m}^3/\text{min.}$$

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

$$\eta_v = 4.654\%$$

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

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

$$= 502.4 \text{ cc/cylinder}$$

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

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

$$\eta_{bth} = \text{Relative } \eta \times \text{Air-standard } \eta$$

$$= 0.5 \times 0.5914$$

$$= 0.2954$$

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

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

Fuel consumption $= \frac{79.64 \times 3600}{42000} = 6.8264$ 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

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

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

Indicated thermal efficiency = $0.6 \times 0.5647 = 0.3388$

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

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

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

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

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

Fuel consumption = $isfc \times ip = 0.3 \times 67.6$

$ip = 20.28$ 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

Solution

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

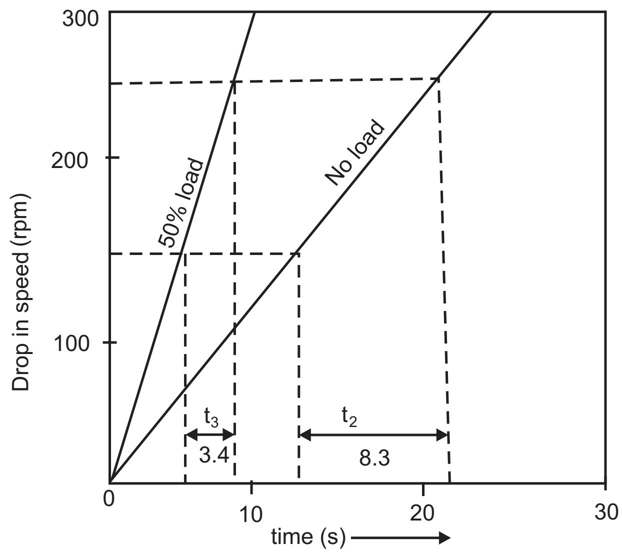

Figure 7.18 : Speed Vs Time

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

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

Torque at half load, $T_{1/2} = 95.5415$ 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.

$$T_f = \frac{t_3}{t_2 - t_3} \times \text{ Torque at 50\% load}$$

$$= \frac{t_3}{t_2 - t_3} \times T_{1/2} = \frac{3.4}{(8.3 - 3.4)} \times 95.5415 = 66.294 \text{ Nm}$$

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

$D = 100$ mm $= 0.1$ m

$L = 120$ mm $= 0.12$ 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 :

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

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

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

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

(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., $C_v = 42000$ 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,

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

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

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

$P_{imep} = 7.5$ bar . . . (1)

Indicated power $= I.P = \dfrac{P_{imp} \times L \times A \times N}{60}$

$$= \frac{7.5 \times 10^5 \times 0.2 \times \frac{\pi}{4}(0.16)^2 \times 400}{60 \times 2} = 10.05 \text{ kW} \quad . . . (2)$$

Brake Power $= B.P = \dfrac{2\pi\ NT}{60} = \dfrac{2\pi\ N\ (W - S)\ \frac{b}{2}}{60}$

$$B.P = \frac{2\pi \times 400\ (380 - 50)}{60} \times \frac{1.2}{2} = 8.294 \text{ W} \quad . . . (3)$$

(i) Frictional Power = F.P. = I.P. − B.P.

= 10.05 − 8.294

= 1.756 kW

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

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

$$= \frac{2.8}{8.294} = 0.3376 \text{ kg/kW.hr}$$

(iv) Brake Thermal Efficiency (η_{bth})

$$= \frac{B.P}{\text{Heat supplied}} = \frac{8.294}{\frac{2.8}{3600} \times 42000} \times 100 = 25.39\%$$

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 = $n_1 = 6$, $L = 100$ mm = 0.1 m

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

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

isfc = 0.3 kg/kWh

$P_{mi} = 8.5$ bar

$N = 2500$ rpm

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

where, $n = \dfrac{N}{2} = \dfrac{2500}{2} = 1250$ 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, $_f$:

$Isfc =$ i.e. 0.3 =

$f = 20.278$ kg/hr

(ii) Calorific Value (C_v) of fuel :

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

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

$$\eta_i = \eta_r \times \eta_a = 0.55 \times 0.42647 = 0.2346$$

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

$\therefore \quad C_v = 51150.6$ 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) $$B.P. = \frac{2\pi \, NT}{1000} = 2\pi \times \frac{600}{60} \times \frac{120}{1000} = 7.54 \text{ kW}$$

$I.P. = B.P. + F.P. = 7.54 + 15 = 9.04$ kW

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

$$\text{Indicated thermal } \eta = \frac{I.P.}{\text{Heat supplied}}$$

$$= \frac{9.04 \times 3600}{2.5 \times 40.3 \times 10^3} = 0.323 = 32.3\%$$

(b) Heat Balance for the Engine :

(i) Energy supplied $= \frac{2.5 \times 40300}{60} = 1680$ kJ/min

(ii) Energy Distributed

(a) Heat in $B.P.$ = 7.54 × 60 = 452.4 kJ/min.

(b) Heat in cooling water $= \frac{818}{60} \times 4.2 \times 10 = 570.8$ kJ/min

(c) Heat in exhaust gases = m_g Cpg (ΔT)

$$= \frac{33 \times 2.5}{60} \times 1.05\ (345 - 25) = 462 \text{ kJ/min}$$

(d) Heat unaccounted (by difference)

= 1680 − (452.4 + 570.8 + 462) = **197.8 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, $i.p. = b.p. + f.p.$

$f.p. = 10$ 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, $f.p. = 10$ kW

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

$= 60.01 + 10$

$\therefore$ $i.p. = 70.01$ kW

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

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

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

(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 = $m_w\, C_{PW}\, \Delta T$

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

$T_o = 65.8 + 273 = 338.8$ K

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

Power lost to cooling water = 50.28 kW

(iv) Power to exhaust = $m_E\, C_{PE}\, \Delta T$

Here, mass of exhaust gases

$m_E = m_a + m_f$

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

Power lost to exhaust = $0.0833 \times 1.25 \times (400 - 20.8)$

$= 39.48$ kW

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 180 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

$$\text{Brake power } = B.P. = \frac{2\pi NT}{60} = \frac{2\pi \times 1800 \times 8}{60}$$

$$= 1507.96 \text{ W} = 1.50796 \text{ kW}$$

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

$= 1.8 - 1.50796$

$= 0.29204$ kW

Loss due to friction power as the percentage of brake power

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

= 19.37% of brake power.

SAQ 1

(a) 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.

(b) 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.

(c) 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/m^3 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 cm^2; 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/m^3 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 mm^2. 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 *bmep* 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

(a) $$P = \frac{2\pi\, NT}{60000} = \frac{P_{bm}\; LA_n}{60000}$$

$$P_{bm} = \frac{2\pi\, NT}{LA_n} = \frac{2\pi\, NT}{L \times \frac{\pi}{4} \times D^2\; \frac{N}{2}} = \frac{16T}{D^2\, L}$$

$$= \frac{16 \times 22.5}{(0.08)^2 \times 0.1} = 5.875 \times 10^5 \text{ Pa}$$

$$= 5.875 \text{ bar}$$

$$P_{bm} = \frac{P \times 6 \times 10^4}{\frac{\pi}{4}\, D^2\; L\, \frac{N}{2}} = 10^{-5} \text{ bar}$$

$$= \frac{4 \times 60000}{\frac{\pi}{4} \times (0.08)^2 \times 0.1 \times \frac{1500}{2}} \times 10^{-5}$$

$$= \frac{24 \times 10^4 \times 4 \times 2 \times 10^{-5}}{\pi \times (0.08)^2 \times 0.1 \times 1500} = 6.369 \text{ bar}$$

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

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

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

$$isfc = bsfc \times \eta_m$$

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

(c) 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 = $8000 \times 0.25133 = 2010.64$ kJ/min

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

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

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

(d) Net area of diagram = $5.75 - 0.25$

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

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

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

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

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

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

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

(e) Average bp for 3 cylinders $= \frac{2\pi\, NT}{60000}$

$$= \frac{2\pi \times 1250 \times 110}{60000}$$

$= 14.39$ kW

Average ip with 1 cylinder $= 21 - 14.39$

$= 6.608$ kW

Total input $= 4 \times 6.608 = 26.433$ kW

$$isfc = bsfc \times \frac{bp}{ip} = 360 \times \frac{21}{26.433}$$

$= 286.006 \approx 286$ g/kWh

Fuel combustion $= \frac{isfc \times ip}{3600 \times 1000}$

$$= \frac{286 \times 26.433}{3600 \times 1000}$$

$= 2.099 \times 10^{-3}$ kg/sec.

$$\eta_{\text{ith}} = \frac{ip}{m_f \times C_v}$$

$$= \frac{26.433}{2.009 \times 10^{-3} \times 43000} \times 100$$

$\eta_{\text{ith}} = 29.29\%$

SAQ 2

(a) $bp = \frac{2\pi\, NT}{60000} = \frac{2\pi \times 4500 \times 48 \times 0.54 \times 9.81}{60000} = 119.82$ kW

$$b_{mep} = \frac{bp \times 6000}{LA\,n\,K} = \frac{119.82 \times 60000}{0.09 \times \frac{\pi}{4}\,(0.1)^2 \times \frac{4500}{2} \times 8} = 5.653 \times 10^5 \text{ Pa}$$

$b_{mep} = 5.653$ bar

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

$bfsc = 0.2203$ kg/kWh

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

(b) $bp = \frac{2\pi\, NT}{60000} = \frac{2\pi NWR}{60000} = \frac{WN\, 2\pi R}{60000}$

$$= \frac{40 \times 9.81 \times 1800 \times 3}{60000} = 35.316 \text{ kW}$$

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

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

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

$imep = 6.94$ bar

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

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

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

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

Heat added = $1493.4 \times 10^{-6} \times 12 \times 10^3$

= 17.92 kJ/cycle

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

$$\eta_{ith} = \frac{ip}{\text{Heat added (in kW)}} \times 100$$

$$= \frac{11.44}{17.92 \times \frac{250}{2 \times 60}} \times 100 = 30.66\%$$

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

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

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

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

Indicated power, $ip = \frac{P_{im} \, LAn}{60000}$

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

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

(e) $ip_1 = bp_{1234} - bp_{234}$

$= 22 - 14.9 = 7.1$ kW

$ip_2 = bp_{1224} - bp_{134}$

$= 22 - 14.3 = 7.7$ kW

$ip_3 = bp_{1234} - bp_{124}$

$= 22 - 14.8 = 7.2$ kW

$ip_4 = bp_{1234} - bp_{123}$

$= 22 - 14.5 = 7.5$ kW

$ip_1 + ip_2 + ip_3 + ip_4 = ip_{1234} = 7.1 + 7.7 + 7.2 + 7.5$

$= 29.5$ kW

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

$$P_{bm} = \frac{bp \times 60000}{\angle \, AnK}$$

$$= \frac{22 \times 60000}{0.09 \times \frac{\pi}{4} \times (0.08)^2 \times \frac{3000}{2} \times 4} = 4.8665 \times 10^5 \text{ Pa}$$

$P_{bm} = 4.8655$ bar.